Electric Railway Handbook

Second Edition

A Reference Book of Practice, Data, Formulas and Tables for the Use of Operators, Engineers and Students

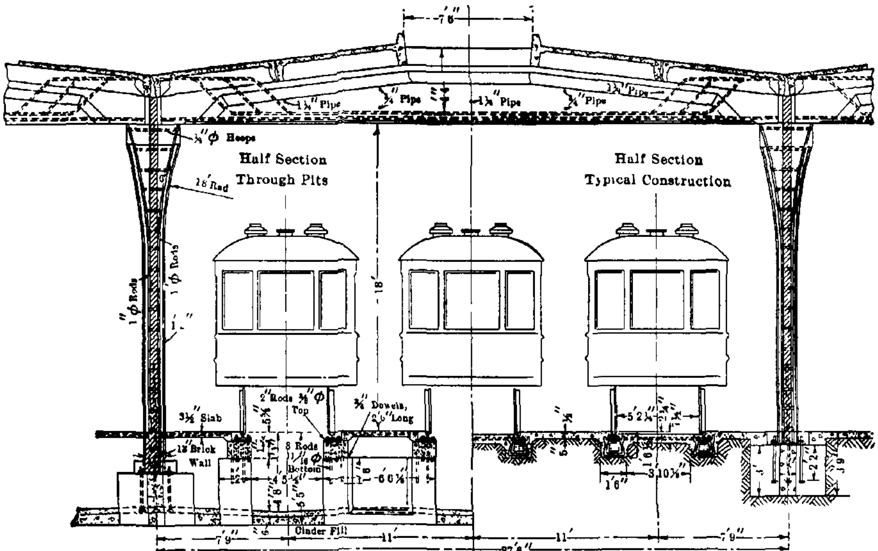

FIG. 19.—Luzerne car house, Philadelphia. Pit section and flush section of inspection bays.

Albert S. Richey

ELECTRIC RAILWAY HANDBOOK

A REFERENCE BOOK OF PRACTICE DATA, FORMULAS AND TABLES FOR THE USE OF OPERATORS, ENGINEERS AND STUDENTS

BY

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SECOND EDITION

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PREFACE TO SECOND EDITION

During the decade since the first edition of this Handbook was presented, there have been many changes in electric railway practice Urgent necessity for the most economical operation has resulted in more rapid development than might have been expected ten years ago. The most striking single feature of this development is the light weight car; this has been accompanied by more scientific design of the car body, both as to structural features and arrangement of entrances and exits, better design of trucks, especially of the single truck, lighter and more economical motors, and more efficient control, brakes and safety appliances. A much more considerable use of machinery in track construction and maintenance is another outcome of the past several years.

Such changes in the art impelled the present revision, and necessitated a radical rewriting of several sections. Advantage was taken of the opportunity to revise and add much valuable material in other sections. In those on Train Movement and Motors, for instance, there is given a very greatly improved treatment of the general subjects pertaining to the selection of a proper motor equipment to meet a required service. It is felt that the present edition is a much better handbook as it relates to the industry today than was the first in its presentation of the material available ten years ago.

Costs and prices have had wide fluctuations during this period, and have not yet reached any stable level. However, where such figures are given herein, they are accompanied either by the basic unit costs or the dates at which they were obtained, so that the user of the Handbook may convert them to the proper figures as of the date of use

ALBERT S. RICHEY.

WORCESTER, MASSACHUSETTS,
August, 1924.

FOREWARD

In 1915, Albert S. Richey, E.E., Consulting Engineer, Professor of Electric Railway Engineering, Worcester Polytechnic Institute, and Fellow of the American Institute of Electrical Engineers published his treatise on the current technology of the electric railway industry. Into this 800 page reference he attempted to present a complete description of all facets of electric railway practice from the engineer's point of view. Updated in 1924, the Electric Railway Handbook remains one of the most complete compendiums of electric railway engineering and practice.

Richey is still a useful reference for today's museum member. Topics covered include roadbed and track, motors, trucks and control. Also such useful engineering data as brake rigging design, car body construction, electric transmission and distribution, signals and communication. Every chapter contains information valuable for accurate and safe restoration of antique electric railway equipment. Each subject is covered in detail including charts and diagrams.

The Association of Railway Museums and the Parts Committee are proud to present this reprint of the Electric Railway Handbook to the museum community. Originally reprinted in 1978 by the Illinois Railway Museum, the ARM is deeply indebted to the Illinois Railway Museum and especially to Mr. James Johnson for preserving the offset printing plates for use in this edition as well as arranging for publication. This edition retains the double page format of the 1978 reprint while incorporating double sided printing and a sewed hardcover binding. We expect that each volume will be a useful addition for bookshelf and shop.

The Electric Railway Handbook is part of a continuing series of publications by the ARM of information for the railway museum community. We are always on the lookout for new material to publish and solicit contributions from our readers.

Rod Fishburn
Chairman, ARM Parts Committee
3 August 1989

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PREFACE TO FIRST EDITION

In the collection and presentation of the material for the Electric Railway Handbook, the aim has been to get together in compact and usable form a large amount of the information which the electric railway engineer frequently requires, but often finds only after an extended search through mechanical, electrical or civil engineers' handbooks or the files of technical journals or proceedings of engineering societies. The field of the electric railway engineer is so broad that such information has necessarily been very much scattered, and progress has been so rapid that a great deal of valuable material has been available only in periodical publications, where, unless held by a good index system, it was soon lost to the operating engineer. The idea of a reference book for the practical electric railway man has been kept in mind, and no attempt has been made to produce a text book, although it is believed that the book may be a useful aid to the student.

It has not seemed wise to include detailed material relating to power plants as such, as this is a field in itself, well covered by existing books, and requiring much more space than is available here. Such matters as the influence of amount and character of load on the capacity and location of power plants and substations are, of course, given due consideration in connection with such subjects as power requirements, energy consumption, feeder layout, etc. In the section on Buildings, the field of the architect has been entered practically only so far as electric railway building problems may differ from others. Other instances may be cited of intentional omissions or very brief consideration of certain matters, such as the elementary treatment of the laying out of track curves, the design of transmission lines, and signaling systems. In the comparatively rare cases where the matters in hand require more extended formulas and data than are here included, the problems are so special and the engineer who is competent to handle them is so well supplied with information that it seems unwise to burden this volume with the material. In short, no attempt has been made to cover the entire field of electric railway engineering, ramifying, as it does, into the special provinces of the civil, mechanical, electrical, chemical engineer and architect. Rather, it has been the aim to present data on the subjects which come up in everyday electric railway practice for constant use by the operating, constructing or designing engineer; a book which may be used by the non-technical manager or operator as well as by the engineer; and a convenient reference book on electric railway practice for those who may be specializing in other or allied lines.

Much material has been gleaned from standard books and publications. Due credit has been given in the text, but special mention should be made of such sources as the *Manual* and the *Proceedings* of the American Electric Railway Engineering Association, the *Transactions* of the American Institute of Electrical Engineers, and of other engineering societies, the *Handbook on Overhead Line Construction* of the National Electric Light Association, the files of the *Electric Railway Journal*, *Electric Journal*, and *Aera*.

A special acknowledgment is also due to the following who have so kindly assisted in furnishing needed information: A. H. Armstrong, M. V. Ayres, J. H. Barnard, E. J. Blair, M. H. Bronsdon, C. L. Cadle, Charles M. Clark, C. L. Crabbs, R. E. Danforth, H. P. Davis, S. R. Dunbar, A. W. French, S. L. Foster, W. G. Gove, Charles Rufus Harte, W. J. Harvie, H. C. Ives, A. St. George Joyce, G. H. Kelsay, Norman Litchfield, Bruce Loomis, W. H. McAloney, Thomas B. McMath, W. S. Murray, George W. Palmer, Jr., F. R. Phillips, Clarence Renshaw, F. L. Rhodes, Ralph H. Rice, Martin Schreiber, Harold B. Smith, W. C. Sparks, Thomas Sproule, C. W. Squier, R. B. Stearns, H. M. Steward, Robert I. Todd. Mr. Walter Jackson has read both the original manuscript and the final proof sheets, and has offered many valuable suggestions.

The book is now presented to its user with the request that he make any criticisms or suggestions for additions which may be helpful in making future editions more valuable in the field. While great care has been exercised in proof reading, it is inconceivable that no errors exist, and notification of any that may be found will be greatly appreciated.

ALBERT S. RICHEY.

WORCESTER, MASSACHUSETTS,
February, 1915.

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ELECTRIC RAILWAY HANDBOOK

SECTION I

ROADBED AND TRACK

Railway Location. The following extracts from the section on "Economics of Railway Location" of the Manual of the Amer. Ry. Eng. Assn. apply to electric as well as to steam railways.

Locating a railway means designing an economical plant for handling a given traffic. The economical plant for a given quantity and class of traffic may not be the economical plant for a greater or less quantity of traffic or for traffic of a different class. It is considered good practice to discount the future within reasonable limits, provided the necessary funds are available.

The most general formula for the economic value of a location is:

$$\frac{R - E}{C} = p$$

Where R = Annual revenue (receipts from operation);
E = Annual expense of operation, including depreciation and taxes;
C = Capital invested (cost of construction);
p = Percentage of income on investment.

The following equation may be used in certain cases, especially where the annual revenue, known or unknown, is constant:

$$R - (E + I) = P$$

Where I = Amount of interest on cost of construction;
P = Amount of profit (net corporate income).

When the revenue is constant the condition of the second equation is that the sum of operating expenses, plus interest on cost of construction, shall be a minimum, and is convenient in many cases, but does not indicate the proportion of profit to investment. Care should be taken not to use too low a rate of interest. The ratio of profit to investment should be considered.

In order to make an economical location of a railway, the engineer must know or make a reasonable assumption of the amount, direction and class of traffic that the railway will be called upon to handle, class of power and the approximate efficiency and cost of fuel that will be used, the rate of wages that will be paid to employes, the cost of maintenance materials, and the rate of interest considered a proper return for additional expenditures involved in the improvement of the line for the reduction of operating expenses.

In the construction of a line where the contemplated immediate traffic is small and the future traffic large, sharp curvature and steep temporary gradients, so situated as to be possible of reduction when justified by the traffic, may be advantageously introduced; such a line will provide for immediate requirements and can be improved for future requirements at a reasonable expense. Before deciding upon such temporary expedients, compare the cost of the work ultimately to be abandoned with the interest saved on the extra cost that would have been necessary to construct a line on the final location during that period in which the more expensive construction would appear uneconomical. In the construction of temporary lines, due consideration must be given to the location of stations, and these should not be located on portions of the line where revisions are contemplated, owing to the fact that when a station is once established, opposition from the public may prevent its removal.

The location of terminal points and ruling gradient having been decided upon, the effect of the minor details of location upon operating expenses may be determined approximately by comparing distance, curvature and line resistance; curvature divided into sharp curvature, necessitating a reduction of speed for trains, and ordinary curvature, which will again be subdivided into that increasing line resistance in both directions and that increasing line resistance in one direction only; and line resistance which is the sum of the rolling resistance (or friction resistance), plus the resistance of gravity overcoming difference in elevation on upgrades, plus the resistance due to curvature, minus the energy of gravity on trains on descending grades, from which has been subtracted the loss of energy (or velocity head) due to the application of brakes. The above does not take into account the resistance due to accelerating trains. This may or may not be a considerable part of the total resistance, depending on the rate of grades and the distance between stops.

Right of Way. Crandall and Barnes in "Railroad Construction' (1913) state that prices at which land can be purchased for railroad purposes will be from one to two times the market value in cities, and from one to four times in the country, it being generally recognized that higher prices are justifiable, partly on account of the greater value of the land for the special purpose and partly on account of the lower value of contiguous lands in consequence of their proximity to the road. The latter reason is not so likely to apply in the case of an electric railway.

Clearing and Grubbing are sometimes lumped together with grading in a contract. Specifications for clearing should require the removal from right of way of all trees, brush and other obstruction to grading, except as reserved; brush and other refuse should be burned or otherwise disposed of, but timber should be saved in the form of saw logs, cut into cord wood, or made into ties, as specified; tops of stumps under embankments should be at least 2½ ft. below grade; the removal of stumps and roots (grubbing) is usually required over all places where excavation occurs and between the slope stakes of embankments less than 2½ ft. in height. Payments for clearing and grubbing are made usually by the acre or square of

100 ft. on a side, or fraction thereof, actually cleared or grubbed, but the removal of isolated trees or buildings is often separately contracted for.

Classification of Grading. Too minute classification gives opportunity for dispute and litigation as the percentages of the different classes are usually estimated, not being susceptible to exact measurement. The American Railway Engineering Association (Manual, 1911) gives three classes (solid rock, loose rock and common excavation) for ordinary use, but recognizes the necessity of special classes, clearly defined, in some localities. The Manual defines the three classes mentioned, as follows: "Solid rock shall comprise rock in solid beds or masses in its original position which may be best removed by blasting, and boulders or detached rock measuring 1 cu. yd. or over. Loose rock shall comprise all detached masses of rock or stone of more than 1 cu. ft. and less than 1 cu. yd., and all other rock which can be properly removed by pick and bar and without blasting, although steam shovel and blasting may be resorted to on favorable occasions in order to facilitate the work. Common excavation shall comprise all other materials of whatsoever nature that do not come under the classification of solid rock, loose rock or such other classification as may be established before the award of the contract." Common excavation may be subdivided into loam, strong, heavy soils and stiff clay or cemented gravel in estimating the cost of excavation with men and teams, but a steam shovel will handle all of these and even loose rock at about the same cost. Hardpan, or earth which cannot be plowed with a four-horse team, is now generally omitted from classification on account of difficulty of separating it from common excavation.

Shrinkage must be allowed for both in figuring on the distribution of the earth and in making the embankments, and the American Railway Engineering Association (Manual, 1911) recommends the following: "For green embankments, shrinkage allowance should be made for both height and width, as follows:

For black dirt, trestle filling	15 per cent
For black dirt, raising under traffic	5 per cent
For clay, trestle filling	10 per cent
For clay, raising under traffic	5 per cent
For sand, trestle filling	6 per cent
For sand, raising under traffic	5 per cent

Solid rock will swell about 70 per cent from cut to fill.

On account of the uncertainty in shrinkage percentages, it is customary to measure earthwork in excavation, but the method of measurement should be clearly specified.

Pay Quantities. Where the material taken from the cuts is used in making the fills, the price paid for excavation includes hauling and placing in the embankment, and this should be done as far as is practicable. The cost increases, however, until the economic lead is reached, where the cost is such that it is cheaper to waste at the cut and borrow at the fill. Beyond the economic lead, the yardage will then become the sum of the remaining cuts and fills (plus shrinkage).

Overhaul. When a limit is set for "free haul" or the maximum haul at the contract yardage rate, a definite additional price per cubic yard per station for overhaul is allowed. The definition for overhaul adopted by the American Railway Engineering Association requires the limit of free haul to be fixed so as to include the grade point and balance the cut on the one side with the fill, plus proper shrinkage allowance, on the other. All material within this free-haul limit is omitted from further consideration. The distance from the center of gravity of the remaining excavation to the center of gravity of the resulting embankment, less the free haul, is the overhaul. When the haul is over runways laid out by the engineer from borrow pits or to waste, the overhaul is one-half the round trip distance, less the free haul. To calculate the overhaul, find the free-haul limit by adding the yardage each way from the grade point, allowing shrinkage for the fills and keeping the quantities balanced until the free-haul limit is reached. This will require using plusses with the corresponding yardages for the final result. Beyond this free point in the cut, multiply the yardage of each volume by the distance of its center of gravity from the near end of the free-haul limit, add the products and divide by the corresponding yardage for the overhaul at the cut end. Find how far this material will extend in the fill and determine the distance of its center of gravity from the near end of the free haul limit as above. The sum at the two ends will give the total overhaul. For heavy work, this method is preferable to a graphic one as being more accurate, unless an inconveniently large scale is used.

Mass Diagram. A simple method of determining the most economical distribution of the material is by means of the mass diagram (or Bruckner's curve). As described by Crandall and Barnes in "Railroad Construction," the mass diagram is constructed by starting at the left or zero end of the profile or other convenient point, say one past which no material will be transported, and adding the yardage algebraically, station by station, allowing for shrinkage on fills. The totals are plotted at the corresponding stations, or plusses for grade points, and a smooth curve drawn through the points thus found as in Fig. 1. Straight lines joining the points would mean that the yardage increments were assumed proportional to the distance

This method of construction gives to the curve the following properties:

(*a*) An upward inclination of the curve from left to right indicates excavation, a downward inclination, embankment, and maximum and minimum points, grade points. Thus (Fig. 1), $A'B'$ indicates excavation, $B'G'$, embankment and B' and G', grade points.

(*b*) Cuts and fills balance between the points of intersection of any horizontal line with the curve, leads being to the right where the curve is above the line and to the left where it is below. Thus, the cut from D to B will make the fill from B to E, and that from F to G, the fill from G to E.

(*c*) Since the cut BD makes the fill to E, the last of the cut at D is moved to E, a distance $D'E'$, while the last of the cut from F is moved a distance $F'E'$. Equal distances will give the minimum average lead for the fill, as this allows taking the material for each part of the fill from the nearest point. Hence in making a fill from adjacent cuts, or in utilizing the material from a cut in adjacent fills, the minimum average lead is found by raising or

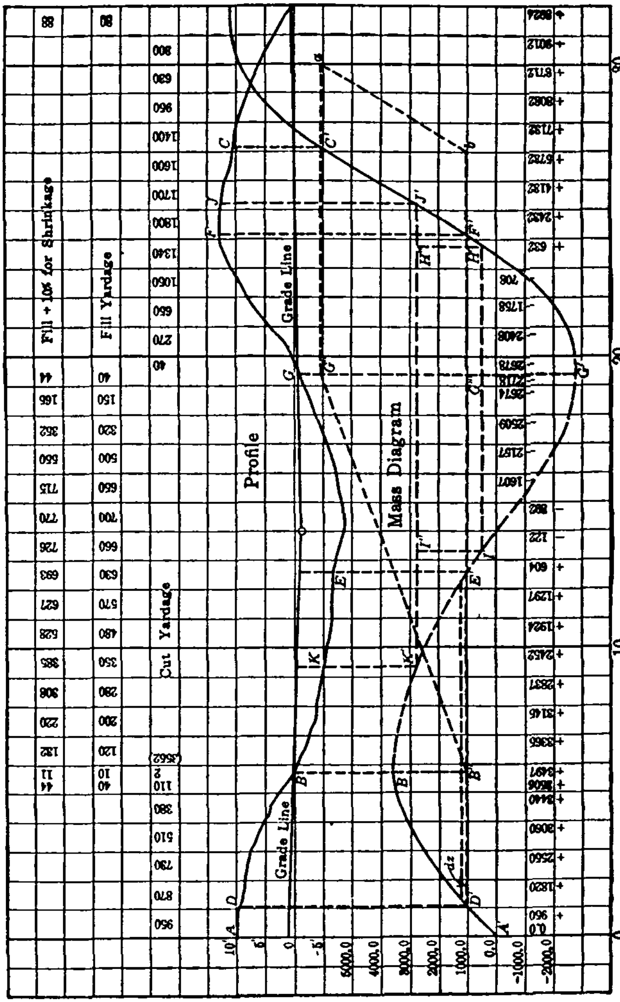


FIG. 1.—Mass diagram.

lowering the horizontal closing line until the adjacent intercepts are equal.

(*d*) The area between the curve and an intersecting horizontal line will be the product, lead times yardage, for the material between the two points of intersection. For since by (*c*) the material in the cut, *e.g.*, from *B* to *D*, will just make the fill from *B* to *E*, the material represented by the increment dz to the ordinate at *D* must be moved from *D* to *E*, a distance $D'E'$, giving for lead times yardage the area $dz \times D'E'$. Similarly for the other increments giving the total area, lead times yardage as stated above. If area be divided by yardage, the quotient will be the average lead for the portion taken. The yardage for any cut or fill (not containing partial sections) will be the difference of the ordinates at the ends. Hence to find the average lead where the material from a cut is utilized in making an adjacent fill, or *vice versa*, find the area between the closing line and the curve by planimeter or by Simpson's rule and divide by the grade point ordinate or total yardage. In the case of side-hill work, *i.e.*, both cut and fill between adjacent sections and no grade point, only the difference of quantities is used. The balanced quantities in each length are thus ignored. The lead for this material will depend upon the longitudinal and transverse slopes; it will generally be small and can be neglected without serious error. To find the average lead for all the material, including the side-hill work, add to the area found from the mass diagram the product of the balanced yardage or side-hill work by its lead and divide by the total yardage.

The total yardage for the cut *AB* (Fig. 1) is 3552, while the maximum or grade point ordinate at *B* is only 3506 less by 46 cu. yd. If *D* be taken at *sta.* 1 + 08, $D'E'$ will be 1020 above the zero line and the area $D'E'B'$ will be 1,868,700 cu. yd. ft. Adding 46 × 10 for the side-hill product, 10 being assumed for the lead,

Total area = 1,869,160
Total yardage = 3552 − 1020 = 2532
Dividing, 1,869,160/2532 = 738 ft.

If the side-hill work be omitted,

1,868,700/2486 = 752 ft., average lead.

If the side-hill lead be omitted,

1,868,700/2532 = 738 ft.

The maximum lead, $D'E'$, = stas. [12 + 50 − (1 + 08)] = 1142 ft.

If the material from *C* to *F* is to be wasted uniformly along the fill, *GB*, draw the horizontal $C'G''$ and join $G''B''$. The area $C'G''B''F'$ divided by the ordinate $G''G'''$ will give the average lead. If this material is to be wasted at the grade point, *G*, divide the area $C'G''G'''F'$ by the yardage $G''G'''$ for the average lead. If wasted on the side at a constant lead, aC', draw the horizontals, aC' and bF' and the curve *ab* for the yardage lead area.

To find the overhaul from the mass diagram, draw the horizontals giving the greatest lead or haul and the free haul; the distance between them will give the yardage for overhaul; draw verticals through the ends of the free-haul line to meet the maximum haul line. The sum of the areas between these verticals and the mass diagram curve will give the overhaul product, lead times yardage. If the overhaul is required, divide the overhaul product by the overhaul yardage. Thus in Fig. 1, for the cut GJ which would make the fill to K, $J'K'$ gives the greatest haul, $H'I'$ the free haul, $H'H''$ or $I'I''$ the overhaul yardage and the sum of the areas, $H'H''J'$ and $I'I''K'$, the overhaul product.

Handling Earthwork. The following figures on cost of handling earthwork, except where otherwise credited, are taken from Crandall and Barnes' "Railroad Construction" (1913).

Loosening. All materials excepting possibly sand require loosening for shovels or scrapers and it will often be economical to loosen even sand, especially for shoveling. Loosening is commonly done with picks or plows, but explosives may be used to advantage under some conditions. A man with a pick will loosen about 15 cu. yd. of stiff clay, or cemented gravel, 20 of strong, heavy soil, or 30 of common loam, per 10-hour day. With wages at 15 cents per hour, these quantities give the following costs for labor, except foremen, for loosening with picks:

Stiff clay or cemented gravel........	10 0 cents per cu. yd.
Strong, heavy soils..................	7.5 cents per cu. yd.
Loam.............................	5.0 cents per cu. yd.

A two-horse team with plow and driver and an extra man to hold will loosen about 250 cu. yd. of strong, heavy soil per 10-hour day or about 400 of ordinary soil. With very hard material requiring a pick-pointed plow with two teams and an extra man to ride the beam, about 180 cu. yd. can be loosened. With wages at $3.50 per day for team and driver and $1.50 for man, these data would give the following costs for labor, except foreman, for loosening with plows:

Stiff clay or hardpan	5.6 cents per cu. yd.
Strong, heavy soils..................	2.0 cents per cu. yd.
Loam.............................	1.25 cents per cu. yd.

Shoveling. Earth may be cast short distances with shovels, the cost being about the same for limits of 5 to 10 ft. horizontally or 4 to 7 ft. vertically. For somewhat greater distances it may be recast, the unit cost being about 80 per cent if a platform or other suitable bed is provided from which to shovel. This is frequently done in taking material from a pit too deep for one cast. Platforms are also often used in mine tunneling and might well be used in cuts when the material is broken down from a face. Shoveling is also required in loading wheelbarrows, carts, wagons or cars for transporting longer distances. The material should be thoroughly loosened and often a second plowing will be more than repaid in the reduced cost of shoveling. The quantity loaded per man will depend upon the material, the extent to which it is loosened, the height to which it must be raised and upon so proportioning the gang that the shovelers will not have to wait for either material or vehicles. For loading into an ordinary wagon or cart, 24 cu. yd. per 10-hour day is about the upper limit for light material or material well loosened, 18 for material at the face of a cut which has been broken down onto a good surface for shoveling, and 15 for rather heavy plowed earth. At $1.50 per day these would give the following costs for labor, except foremen, of shoveling:

Light or well loosened material.......	6.25 cents per cu. yd.
Face broken onto good surface.. . ..	8.25 cents per cu. yd.
Heavy plowed earth..	10.00 cents per cu. yd.

In *Engineering-Contracting*, 1909, it is stated that experiments on a number of pieces of work involving the handling of thousands of yards of excavation have shown the long-handled shovel much superior to the short for handling earth. Men can do more work with less effort and can stand more nearly erect. It is stated that the long handle is used in Europe and the West, and the short handle in the East. The round-pointed shovel enters the earth easier than the square and is generally used except in shoveling from a platform when a square-pointed scoop should be used unless the material has to be cast a considerable distance.

Spreading and Dressing. A bankman will spread in 6-in. layers about 75 cu. yd. of average earth which has been dumped from carts or wagons, the cost therefore being about 2 cents a cubic yard with wages at $1.50 per day. For a railroad bank, it is usually only necessary to keep the surface smooth enough to drive over until grade is reached, in which case half a cent per cubic yard should be sufficient if the work is so planned that the bankman will be kept busy, *i.e.*, at least 300 cu. yd. must be delivered at the bank each day. If the earth is hauled in carts and dumped over the edge of the bank, about one-quarter of a cent per cubic yard should still be allowed for keeping the dumping place in order. Earth in large quantities can be spread by a team and road machine or Shuart grader for from one-half to three-quarters of a cent per cubic yard. In dressing railroad earth slopes from 125 to 150 sq. yd. per man per day is a fair average on construction work. This would make the cost about 1 cent per square yard. The cost per cubic yard would, of course, depend on the lightness of the work, the amount of material handled not being a proper criterion on account of the time spent in smoothing and skimming.

Superintendence, Timekeeping and Water Carrying. These items vary with local conditions, but half a cent per cubic yard will usually cover all three.

Wheelbarrows. The wheelbarrow is used for excavating small quantities of material and in cases where it is impracticable to use carts or scrapers on account of the cramped situation or deep mud in which horses could not work. It is also valuable for moving stony soil over short distances. Run planks or "gangways" should be provided and usually each man should load his own barrow. The load is about 1/14 cu. yd. for wheeling up slopes, as is generally necessary, but it may reach 1/10 for level ground. A study of the available data would indicate that the time for loading a cubic yard may be taken at 0.6 hour for the heavy soils and 0.5 for loam, while the time for wheeling, including dumping, adjusting the plank and other unavoidable delays, may be taken at

⅓ hour per cubic yard per 100 ft. for wheeling up slopes and ¼ hour for level ground. Adding the cost of picking gives the following costs for labor except foremen, for picking and loading, with wages at 15 cents per hour:

Stiff clay or cemented gravel	19.0 cents per cu. yd.
Strong, heavy soil	16 5 cents per cu. yd.
Loam	12.5 cents per cu. yd.

to which should be added 3¾ to 5 cents for each 100 ft. of lead, depending upon the slope.

Drag Scrapers. The ordinary No. 2 drag scraper is a steel scoop weighing about 100 lb., which will hold from ¼ to ⅙ cu. yd., place measurement. It is drawn by a two-horse team which will travel at the rate of 2.5 miles per hour, or cover 100 ft. of haul per minute including loading and dumping, the haul making allowance for the extra distance required for turning at the pit and dump and being greater than the lead or straight line distance. An extra man is required to hold in loading while the driver usually dumps his own scraper. As a man can load for two or more teams, depending upon the lead, it is advisable to work the scrapers in gangs. The material should be well loosened so that the scoop will fill readily. J. W. Brown, *Engineering Record*, gives the following data based on the average cost of scraper work in Iowa. In making low embankments from side ditches with 6-ft. berms he makes the following assumptions: Distance, center of ditch to center of bank, 33 ft. Seven to ten trips per cubic yard. Sixty cubic yards per 10-hour day good average work. Plow team generally required to loosen the material, one plow to six scrapers. Two horses per plow required in light soil, four or more necessary in compact soil, average three at $6 per day with driver and man to hold. Field expenses, except management, and maintenance for loosening, loading and dumping, 360 cu. yd.:

Three-horse team with driver and plowman, loosening	$6.00
Three men holding scrapers, loading	4.50
One man dumping	1.75
One foreman	2.50
Maintenance, plows and scrapers	0.90
Total	$15.65

This gives 4.35 cents per cubic yard. The cost of hauling at $3.50 per day for team and driver is 5.83 cents, giving a total cost of 10.18 cents per cubic yard for the lead of 33 ft. For double the lead, or 66 ft., the yardage would be reduced to 40 per scraper per day, requiring 9 teams or $31.50 for hauling, while for a lead of 100 ft. the number of teams must be increased to 12 or the cost to $42, the other expenses remaining the same except that maintenance would be increased to $1.08 and $1.44, respectively, giving 8.75 cents and 11.66 cents for hauling and an increase of 0.05 cent and 0.15 cent for maintenance. For the above prices and material these data give the following rule for field expenses, except management, and maintenance for the drag scraper: To a fixed cost of 7.2 cents per cubic yard add 9 cents per 100 ft. of lead with 20 ft. as a minimum value to allow for turning. For stiff clay 25 to 30 per cent should be added to the above.

Fresno Scrapers. The Fresno scraper is an improved form of drag scraper invented in California and used mainly in the West. The four-horse scoop is 5 ft. long laterally and only about 18 in. wide from cutting edge to rear, the former giving large capacity and the latter easy filling. Two- and three-horse sizes are also made. The horses are driven abreast by one driver. The four-horse scraper weighs from 275 to 310 lb. Many Fresno loads at the dump have been compacted into a box with a rammer and found to run from 12 to 16 cu. ft. in average earth where the lead was not so great that much material was lost in transit. The editors of *Engineering-Contracting* give the following rules for estimating cost, based on a study of a large number of data: When the daily wage of a driver is $2 and that of each of the four horses is $1 a total of $6 per Fresno per 10-hour day, the average cost, not including plowing, trimming or superintendence, will be as below: To a fixed cost of 4 cents per cubic yard add 2 cents per 100 ft. of lead. The fixed cost includes traveling the extra distance and the slower speed in loading, the shifting to newly plowed ground, etc. The hauling cost is based upon a traveling speed of 200 ft. per minute when not delayed by loading, dumping, etc., and upon an average load of ⅓ cu. yd., with 50 ft. as the minimum lead. If the soil is not of a kind that heaps up and drifts well in front of the scraper, the average load will probably not exceed ¼ cu. yd., particularly on long hauls. This would change the rule to: To a fixed cost of 4 cents per cubic yard add 3 cents for each 100 ft. of lead. The editors believe that the horses can be crowded so as to do about the same amount of work in an 8-hour day as in a 10-hour day, or that the cost per cubic yard would be but slightly affected, but experience in the East with 8-hour days would hardly warrant this belief. They state that the cost of plowing ordinarily ranges from ¾ cent to 1.5 cents per cubic yard, foreman's wages, from ½ to 1 cent, and dressing roadbed and slopes about ½ cent per square yard of surface trimmed. If the cost of plowing, dumping, maintenance and superintendence be added on the basis of $10.25 for 300 cu. yd. with ¼ cent per cubic yard for maintenance, as previously given, the fixed cost would be increased by 3.1 cents giving: To a fixed cost of 7.1 cents per cubic yard add 2 cents per 100 ft. of lead, or to a fixed cost of 7.1 cents per cubic yard add 3 cents per 100 ft. of lead, according as the material heaps up and drifts in front of the scraper. This allows the driver $2 per day and requires him to load his own scraper, thus increasing the cost about ½ cent per cubic yard per 100 ft. of lead, and reducing it 1¼ cents independent of lead, as compared with the drag scraper figures previously given.

Wheel Scrapers. With the wheel scraper, the steel scoop is hung between two wheels with broad tires and it can be lowered and filled, raised or dumped, while the team is in motion. The capacity, place measurement, ranges from about 6½ cu. ft. for the No. 1 to 12½ for the No. 3, while the loads actually carried are about 0.2, 0.25, 0.4 cu. yd., respectively, for the three sizes. These loads can be increased for long leads by finishing with shovels when the material does not fill readily. The dead load varies from

350 to 800 lb. according to size. A snatch team is generally used in loading all but the No. 1, even then shovelers are necessary if the box is to be filled in tough clay. In scraper, as in other work, the details must be carefully studied and given attention if economical results are to be secured. Thus the plow should be set to cut 10 to 12 in. deep or to such a depth that the scoop will be heaping full after traveling but a few feet. The rear portion of the pan will not fill well with shallow plowing. The furrows should be close together, and if the soil is heavy it should be plowed twice. The bottom of the cut should be kept level so that the scoop will lie flat and not tilted. J. W. Brown, *Engineering Record,* as the result of experience in Iowa, gives the same costs, not including scraper teams and drivers, for handling 360 cu. yd. of earth with the No. 1 wheeler as were given for the drag scraper on page 9, except that the 90 cents for maintenance is increased to $1.40. He assumes four loads per cubic yard and about 100 ft. of lead per minute while traveling, or 60, 40 and 30 cu. yd. per scraper per day for leads of 100, 200 and 300 ft., respectively, requiring 6, 9 and 12 wheelers for transporting the 360 cu. yd. over the respective distances. He also increases maintenance slightly for the longer leads. These values would give for the No. 1 wheeler at $3.50 per 10-hour day, for average soil: To a fixed cost of 7½ cents add 3 cents for each 100 ft. of lead, with 75 ft. as a minimum. For leads over 300 ft. Mr. Brown uses No. 3 wheelers with two men to hold a scraper (requiring one extra holder) and a two-horse snatch team to aid in loading. To move the 360 cu. yd. per day he used 8 wheelers for a lead of 400 ft., 10 for 500, 12 for 600, 14 for 700 and 16 for 800, the limit to which he considers it advisable to go with wheel scrapers. Adding $3.50 for the snatch team, $1.50 for the extra man to hold and $0.90 as lead increases for extra wear to the fixed cost, will give a total of $21.15 per day, or 5.88 cents per cubic yard for the fixed charge. Adding to this $3.50 per scraper for the different leads and dividing by 360, the number of cubic yards moved, will give the cost for hauling. The cost per cubic yard is given quite closely by the following: To a fixed cost of 5½ cents add 2 cents for each 100 ft. of lead within the limits of 300 and 800 ft. For a No. 2 the cost would be given approximately by the following: To a fixed cost of 6½ cents add 2½ cents for each of 100 ft. of lead within the limits of 200 to 500 ft. Good roads are essential to economy, especially with the No. 3 wheeler. These are for average conditions. For light material $4.50 per day could be saved for the No. 1 wheeler by having the drivers hold their own scrapers, while for heavy clay, a three- or four-horse snatch team with an extra man to hook and unhook would be needed for the No. 3, instead of the 2-horse team. Four horses might also be needed for the plow team. Scrapers are used to some extent for loading cars and wagons through a platform, over which the teams are driven and the material dumped through an opening.

Carts. The one-horse cart, although not used so much as formerly, is economical for short leads when shovel loading is employed and is convenient in turning and in dumping over the end of an embankment. Five is a suitable number of shovelers for loading, two on each side and one in the rear. They can load a cart with 1/3 cu. yd in 2 1/2 minutes, while about 1 minute is required for turning and dumping, making a total of, say, 4 minutes per trip, allowing for turning into place for loading. A driver can attend two carts, by dumping one while the other is being loaded, for leads up to 300 ft. For greater leads he can attend to two by taking them both together to the dump. At $1 per day for a horse and $1 50 for a driver this would give for 100 ft. of lead per minute and 3 trips per cubic yard:

5/8 cent per 100 ft. of lead, driver to 2 carts.

1 1/4 cents per 100 ft. of lead, driver to 1 cart.

For the fixed cost,

Four minutes per trip	3 1/2 cents
Shoveling, average material	9 cents
Plowing	2 cents
Dump	1/2 cent
Foreman and maintenance	1 cent
Total per cubic yard	16 cents

This gives for field expenses, except management and maintenance:

To a fixed cost of 16 cents, add 5/8 cent for each 100 ft. of lead.

With one driver per cart, the 4 minutes will cost 5 cents, giving:

To a fixed cost of 17 1/2 cents, add 1 1/4 cents for each 100 ft. of lead.

This is on the supposition that the number of carts is so proportioned to that of the shovelers that both can be kept busy, otherwise the cost may be much greater.

Wagons. These may have an advantage over carts for long leads on account of the larger load, but they are at a disadvantage in turning The slat-bottom box used for grading with an ordinary wagon is 3 by 9 by 1 ft giving a capacity of 1 cu. yd. of loose earth, or about 0 8 cu yd place measurement. This is a full load for temporary roads over soft earth and up steep pitches, as in most railroad work. For long hauls over hard roads, as in road improvement and city work, additional side boards are much used, increasing the load to 1 1/4 to 1 1/2 cu. yd , place measurement. The slat-bottom box has about 3 by 4-in slats for the bottom and requires a man at the bank to aid the driver in dumping, which takes about 1 1/2 minutes for the 0 8-cu. yd or 3 minutes for the 1 1/2-cu yd load. Dump wagons which can be dumped and the bottom closed again without stopping the team are rapidly coming into use. Enough men should be put in the pit to load a cubic yard in about 5 minutes. This would give at 35 cents per hour for team and driver, 3 cents for loading time, or about 4 cents total for loading and dumping time with slat-bottom wagons. If inconvenient to use so many shovelers an extra wagon may be loaded while the team is going to the dump so that by shifting the lost time can be kept about the same. If the earth is plowed and shoveled, with foreman and maintenance increased to 1 1/2 cents this would give for field expenses, except management, and maintenance for a speed of 2.5 miles per hour, or 220 ft. per minute when traveling:

To a fixed cost of 17 cents, add ½ cent for each 100 ft. of lead, for loads of 1 cu. yd., place measurement.

For different loads the amount to add for each 100 ft. of lead would be as follows:

Load of 0.8 cu. yd. add 0.66 cent per 100 ft.
Load of 1 cu. yd. add 0.53 cent per 100 ft.
Load of 1.5 cu. yd. add 0.35 cent per 100 ft.
Load of 2 cu. yd. add 0.26 cent per 100 ft.

Frequently wagons can be loaded with scrapers dumping through a platform cheaper than with shovels. The following data are taken from *Engineering-Contracting*, 1907, for the cost of excavating a street of a western city to a depth of about 2 ft., using a plow and drag scrapers. A 10 by 12-ft. platform was built with a floor of 2-in. plank on 6 by 6-in. stringers high enough to give a clearance of about 7.5 ft. An opening 2 ft. square was left in the center through which the material was dumped automatically by the front end of the scraper catching on a cleat nailed in front of the hole. This aided but did not do away with the dumpman. There were two inclined runways. The approach was steep and soon banked with earth; the run off was on a 15 per cent gradient. The street gradient was 6 per cent and the material was hauled down grade an average distance of 120 ft. in direct line. The platform had to be moved from time to time. The wagon loads averaged 2 cu. yd. in place and a wagon was filled by scraper loads in less than 6 minutes, or at the rate of more than 20 cu. yd. per hour. The labor cost, except foreman, of loading per hour was as follows:

One plow team	$0.40
One man holding plow	0.20
One man holding scraper	0.20
One man at dump	0.20
Five scraper teams	2.00
Total for 20 cu. yd.	$3.00

or 15 cents per cubic yard. The No. 1 wheel scraper should give better results than the drag scraper for this lead.

Power Shovels. The standard machine for loading large quantities of material is the power shovel. It is usually operated by steam, but sometimes by electricity. It will handle earth, loose rock, and even cemented material without loosening. Solid rock must, of course, be blasted and it will often prove economical to loosen cemented material and loose rock. The ordinary shovel is usually mounted on standard gage trucks and provided with propelling chains. The boom swings through an angle somewhat over 180 deg. Jacks outside the tracks are used at the front corners to give a broader base for stability while at work. Some of the smaller shovels are balanced on a car and can swing through a full circle. Some of the makers mount these on traction wheels for highway and street work, cellar excavation, etc. Shovels with short booms are also made for tunnel and mining work. The dipper dumps through a door at the bottom and for light material its capacity is sometimes increased by providing an extension or lip in front. For hard material dippers are fitted with steel teeth. Heavy machines with smaller dippers than for earth are usual for hard digging and for handling large boulders. Steam shovels are usually operated by three men, the engine man or runner, the cranesman and the fireman The engineman controls the raising and lowering of the dipper and the swinging of the boom, while the cranesman regulates the depth of cut or "bite," releases the dipper from the bank when full and dumps the load. Pitmen, usually four to six, prepare for and move the short sections of track forward, operate the jacks and chocks in moving, etc An excellent analysis of the cost of steam shovel work and discussion of the factors affecting the same are given in a Handbook of Steam Shovel Work published by the Bucyrus Company, South Milwaukee, Wis, it being a report by the Construction Service Company, based on records and time studies given in full for 45 Bucyrus shovels working under different conditions as to material, depth of cutting, size of cars, number of cars in a train, management, etc A formula is given for cost of loading cars, in cents per cubic yard, place measurement, for shovel work only, including plant expenses, and labor (except superintendence) and materials of field expenses, in which

d = time in minutes to load 1 cu. ft, place measurement
c = capacity of one car in cubic feet, place measurement
f = time shovel is interrupted to spot one car
e = time shovel is interrupted to change trains
g = time required to move shovel
L = distance of one move of shovel in feet
M = minutes per shift less loss for accidental delays
A = area of excavated section in square feet
R = cost per cubic yard on cars
n = number of cars per train
C = shovel expense in cents per shift

From these,

$$R = \frac{27C}{M}\left(d + \frac{f}{c} + \frac{e}{nc} + \frac{g}{LA}\right)$$

Using estimated values of C and A and the average values given below (or estimated ones) for the other terms except M and d, a plate of cost curves may be plotted for any value of LA showing the relation between R and d for various values of M. To estimate C, a $14,000 shovel is assumed, with the following data.

	Cost per year
Depreciation, 4⅔ per cent	$653 34
Interest, 6 per cent	840 00
Repairs, when working one shift	2000 00
	$3493 34

Per year of 150 working days, or $23 29 per working day	$23 29
Shovel runner	5 00
Cranesman	3 60
Fireman	2 40
One-half watchman at $50 per month	1 00
Six pitmen at $1 50	9 00
One team hauling coal, water, etc, half day, say	2 50
Two and a half tons coal at $3.50	8 75
Oil, waste, etc, say	1 50
ost per day, C/100	$57.04

The depreciation is found by distributing the difference between first cost, assumed at $150 per ton, and scrap value, $10 per ton, over the life of the shovel, assumed as 20 years. It is stated that "The cost of repairs should be apportioned to the work turned out rather than considered as a function of the age of the shovel. It will be higher for rock than for earth-work and higher for badly broken rock than for well-blasted material." In assuming 150 working days per year, allowance has been made for bad weather, lack of continuous work, transportation of plant, etc. The actual number will be greatly affected by local conditions. The fuel consumption assumed for the heavy shovel used checks fairly well with some data given by Gillette, Handbook of Cost Data. His values for coal and water per 10-hour day vary from ¾ ton and 1500 gal. for a 35-ton shovel with a 1¼-cu. yd. dipper to 2¼ tons and 4500 gal. for a 90-ton shovel with a 3-cu yd dipper. The average shovel move, L, was 6 ft. A varies with the depth and width of cut. For example, values of 250, 500 and 1000 sq. ft. are used in the Handbook in computing cost curves. The larger the volume, LA, per shovel move the less the cost per cubic yard. The width of cut and L are fixed by the reach of the shovel. In increasing the depth to increase A, the danger of landslides should be considered as also the height of the loading track if cars are handled in trains alongside the shovel. On the other hand, if the depth reaches a certain minimum, varying with the material, such that the dipper will not fill in one raise, the cost will also be increased by increasing d, the time required to load a cubic foot. The time d, required to load 1 cu. ft depends upon the material, the depth of cutting, the shovel and the capacity of its dipper, as well as upon the skill and cooperation of the engineman and cranesman. These men must work together perfectly or costs will be seriously affected. The results of the tests taken show that the average time to load 1 cu. yd, place measurement, is about 10½ seconds for iron ore, 12 for sand, 18⅓ for clay and earth and 31½ for rock. These were with average dipper capacities, place measurement of 1 cu. yd. for rock, 1¼ for earth and sand, 1½ for clay and 2⅓ for iron ore. The average ratios of place measurement to water measurement for the dippers were 0.94 for iron ore, 0.56 for sand, 0.61 for clay, 0.53 for earth and 0.43 for rock. The time, f, for spotting cars is usually zero, as it is done while the shovel is turning and digging. This is where the train is alongside and moved a car length at a time without switching. The capacity, c, is taken as 4 cu. yd, water measurement, or 2.5, place measurement, for ordinary contracting work where 10 car trains of side dumping cars are most common, or $n = 10$. The time, e, the shovel was interrupted to change trains averaged 4 minutes. The average time, g, required to move the shovel averaged 8 minutes. It depends upon the skill and cooperation of the crew and pitmen, and should be done according to a definite schedule. The time lost by accidental delays, to be subtracted in finding M, averaged about 7¼ per cent for brick clay, 8½ for sand, gravel and iron ore, 17½ for earth, clay and loam from railroad borrow pits and crushed stone from quarries, and 20 for rock cuts on railroad work; with a maximum of about 40 per cent for borrow pits of earth and 56 for rock cuts. These delays may be due to the condition of the material or to breakdowns or to accidents. Thus, wet clay often clogs the dipper teeth and sticks in the bucket. Delays are most frequent, however, in handling rock, especially if it has not been properly broken. Large pieces have to be "chained out" or broken by mud capping or block-holing and blasting, thus seriously delaying the shovel. Small air hammer drills can be used to advantage in block-holing as the hole can often be drilled on the side of the stone away from the shovel and the stone broken by light charges. Every effort should be made, however, to properly break the rock with the original blasts. Holes a few inches too shallow or with the bottoms not properly loaded often leave ridges which must be drilled and blasted before the track for the shovel can be laid. Delays due to breakdowns should be minimized by keeping duplicate parts liable to breakage on the job and training the men to make repairs quickly. The shovel should be carefully inspected each night and parts liable to break the next day replaced. It should be noted that the use of average values of the various factors will give only what may be considered standard cost curves. But estimated values, or actual ones from time studies, are easily used, thus making it possible to vary conditions and determine what plan of operations gives the best results. The Bucyrus formula shows clearly that delays, either e, f and g, or those which reduce the value of M, increase R, the cost of loading, and they may also increase the cost of transportation. The best efforts of the management should therefore be directed toward reducing these, both by the use of proper general design and plan of operation, and by careful supervision during the progress of the work.

Steam Shovels for Light Work. For light work or for loading into wagons or small cars a small shovel is often preferable. It is estimated, for example, that the Thew 13-ton full swing shovel with traction wheels and ⅝-cu. yd. dipper can be operated for $13.50 per 10-hour day, as follows:

Engineman	$5.00
Fireman	2.00
Two pitmen	3.00
Fuel at $4 per ton	2.00
Supplies and repairs	1.50
Total	$13.50

It is claimed that it will excavate some 35 cu. yd. per hour in ordinary soil, while if the engineman does his own firing and but one pitman is used the cost can be reduced to about $7 and the capacity will be about 25 cu. yd. per hour. Allowing 1 minute delay per load in getting the 1-cu. yd. wagons into place and a little extra time for moving, this would give 20 cu. yd. per hour, or 200 per day for the full crew. Dividing $13.50 by 200 would give 6¾ cents per cubic yard for loosening and loading. This does not include delays, depreciation or interest.

Electric Shovels. When electric power is available, a considerable saving can be made by the use of the electric shovel, in which electric motors displace the steam engine and boiler of the steam shovel. Many electric roads are making good use of the electric shovel, not only in gravel pits and grading, but in construction and reconstruction work in city streets, in the latter case usually in connection with pavement plow, pneumatic tools or the "skull cracker" for breaking up the pavement. During construction, the Brantford & Hamilton (Ont.) Electric Railway used an electric shovel of 1 to 1¼ cu. yd capacity in gravel pit work, the depth of cut being about 14 ft., gravel loaded on flat cars of 14 cu. yd. capacity each, frequently hauling away 100 loaded cars daily.

The Rockford & Interurban Railway reports use of a 1¼ cu. yd. electric shovel in 15-ft. gravel bank, with three work train motors and thirteen 12-yd. center dump ballast cars; four per cent grade out of pit, average daily mileage of work trains, 120; shovel operated by three men, foreman's time divided between pit and surfacing gang; work trains cleared passenger cars, shovel not operated to capacity because of inability to keep cars spotted for loading; total mileage per month, 6630. The same shovel, operating in 12-ft. bank of earth with hardpan at bottom, one work train of three 12-yd. side dump cars, average 50 miles daily with work train, cleared passenger cars, shovel operated by two and three men, 9240 cu. yds. moved per month.

The Milwaukee Electric Railway & Light Company, using an electric shovel to excavate track trench in city streets, reports a season's output of 25,000 cu. yds. Part of this work was on new construction, but on that part which was reconstruction excavation was done at night, mainly during the time of owl car operation. The excavated material was loaded into dump cars on the adjacent track, the work trains clearing the passenger cars, except at the outer ends of lines stub service was operated through the early hours of the night, and passengers transferred around the shovel operation.

The use of the electric shovel has become quite general, and these instances are only given as examples. One more might be cited, however, that of The Indianapolis Traction & Terminal Company, which uses the electric shovel to excavate, grade and subgrade in city streets. The ditch is 9 ft. wide by 24 in. deep, and an average day's work is 340 lin. ft., maximum 410 lin. ft. Work less than 300 ft. is due to obstructions such as gas and water pipes, conduits, manholes, etc., which the shovel must work over or around. The shovel crew consists of one foreman, one shovel operator and eight laborers, moving an average of 226 cu. yds. daily. To do an average day's work requires a move every six or seven minutes, each move being four feet. With deeper digging and less frequent moves a greater number of cubic yards can be handled. It makes no material difference as to the character of the excavation, that is, clay, gravel, old paving concrete and old ties. The old rail is pulled out with a crane car and then the shovel is put in after removing any of the old paving that has value. No current charge is made, but it will average about 35 amperes on 550 volts. On clay digging in 4-ft. bank the machine averaged 420 lin. ft. ditch 9 ft wide, with same crew In all cases material was loaded into 6-yd standard gage Western dump cars.

Hauling with Cars and Dinkey Locomotives. On new construction this is the standard method of hauling material excavated with steam shovel. The usual gage is 3 ft. The cars are side dump of 3 or 4 cu yd. capacity the latter weighing about 6000 lb. The dinkey weighs from 8 to about 30 tons; all on drivers. This light weight allows the use of rails of from 16 to 40 lb. per yard. With 5-ft ties about 6 by 6 in. in section, it makes a light track which can be easily shifted at cut or dump as required. The following rental rates were quoted by an equipment company, July, 1911.

Four-cu yd 3-ft. gage Western cars,	
First month	$10 50
Second month	10 20
Third to fifth month inclusive	9 60
Sixth to eighth month inclusive	9 00
Each month thereafter	8 40
Twelve-ton dinkey,	
First month	$141 00
Second month	132 00
Third to fifth month inclusive	117 00
Sixth to eighth month inclusive	101 00
Each month thereafter	93 00

The rolling friction on this light track is from 20 to 30 lb. per ton and probably more in starting on dirty track. The dinkey can exert a pulling force of about one-fourth of its weight. The speed is about 5 miles per hour when loaded and 8 to 9 when empty or on down gradients with smooth track. One dinkey is often used with a 1½-cu. yd. shovel for leads up to 1000 ft. With six cars and three or four dumpmen the train can be dumped in about 2 minutes so that good results can be obtained. For longer leads, a second engine would be required for spotting cars, with cars enough for two trains, one dumping while the other is loading. The length of train and weight of engine should increase with the lead when the work is heavy enough to warrant it

Hauling with Cars and Horses. Two-foot gage 1-cu. yd. capacity cars weigh about 1000 lb each and 1¼-cu. yd. cars about 1350, so that one horse can draw three loaded cars if favored slightly by the gradient for the heavier cars. Fifteen to 20-lb rails are heavy enough, with plank or round timber ties A side track is put in at the cut and two trains are used the same as for dinkey engines. For hand loading both tracks should extend into the cut and be used alternately to save work in switching. Allowing the driver 6 minutes for dumping and 1 minute at the cut, the fixed cost, at $1 50 for the driver and $1 00 for the horse, would be 1 cent per cubic yard for the heavier cars, 1 cu. yd place measurement, while the cost per 100 ft of lead would be 0.14 cent, giving for the cost of hauling per cubic yard· To a fixed cost of 1 cent add 0 14 cent for each 100 ft of lead. For level or slightly rising track, requiring the 0.8-cu. yd., place measurement, cars, this would become: To a fixed cost of 1¼ cents, add 0 18 cent for each 100 ft. of lead. This assumes a dumping trestle and no trackwork or depreciation. The trackwork would cost much less than for the shovel and dinkey, especially at the cut. One cent plus 0.2 cent per 100 ft. of lead should cover ordinary conditions. With no dumping trestle 1½ cents per cubic yard should be added for spreading and about 1 cent for extra shifting of track. This would give for the cost of transportation: To a fixed cost of 4 5 cents add 0 34 cent for each 100 ft. of lead, or to a fixed cost of 4¾ cents add 0 4 cent for each 100 ft. of lead, according as 3 or 2 4 cu. yd. are hauled per train. *To this must be added the cost of loosening and loading.*

Power Shovel and Standard Equipment. Standard gage flat cars and ballast cars are used on maintenance work in widening cuts and fills, filling old trestles, reducing gradients, distributing ballast, etc. The flat cars are unloaded by a plow drawn by cable. The ballast or dump cars are dumped through doors operated by air from the brake system. For the heavy cars, inclined floors rather than tilting bodies make them self cleaning. The capacities of the flat and ballast cars range from 10 to about 30 cu. yd. The cost of loading with power shovel and hauling on good track with large cars is less than with the small cars and poor track used on new work, provided the forces can be so adjusted as to keep all busy. Usually, however, the work must be done subject to interruption from traffic so that a careful study of conditions must be made in order to estimate costs. W. Beahan, First Asst Engr., L. S. & M S. Ry., in a letter dated October, 1911, places the cost of grading for third and fourth track, using standard equipment and a haul not exceeding 5 miles including loading, unloading and leveling ready to lay the ties, about as follows per cubic yard:

Borrow pits or cuts with 15-ft. face	$0 11
Earth cuts, 3 to 10 ft. deep . .	0 15
Shale cuts, all blasted	0 21
Other rock cuts all blasted and requiring breaking up by blockholing	0 25

This includes labor and supplies as follows.

Foreman per month	$75 00
Laborers per hour, 10-hour days	0 15
Steam shovel crew, 8 men per 10-hour day	25 00
Train service, labor and supplies per day	28 00

Interest, depreciation, explosives and overhead charges are not included The repairs for shovel probably are included as the labor was performed by the shovel crew. Mr. Beahan states that in moving a short distance where overhead obstructions will allow, they sometimes let the dipper rest on a flat car, remove the jack arms and haul the shovel with the work train; but that usually it is best to take the shovel down even for moving a short distance, and they estimate that it will cost $100 to take a shovel out of one cut and put it in another, although they can occasionally do it for $50. A. J. Himes, Engr. of Grade Elim., N. Y. C. & St. L. R. R. Co., gives the cost of moving a shovel about 3½ miles through the City of Cleveland in August, 1911, as follows.

LABOR		
Taking down		$23 26
Moving		49 11
Setting up		23 27
		$95 64
WORK TRAIN SERVICE:		
Placing cars to load parts and helping to take down	$10 20	
Moving shovel and bunk cars	8 50	18 70
		$114 34

The cost of shipping the above shovel from Ashtabula to Cleveland and setting up is given as follows·

Freight shovel and three cars at $19 50	78 00
Boarding car and tool car at $22 50	45 00
Lost time shovel crew	8 84
Setting up shovel	31 47
	$163 31

On new work, the cost of moving from the railroad to the site would be in addition to that of setting up, or taking down and setting up, as above. This may be over highways or across country. A track is required with force sufficient to take up and move forward ahead of the shovel. The shovel can be moved with its own power if the gradients are not too steep. If too steep for adhesion, one end of a rope can be anchored ahead, the other end wound around the driving axle and the running gear started. No general estimate of cost of moving can be given on account of the variation in conditions. The Bucyrus Handbook of Steam Shovel Work states that a 70-ton shovel was moved 1600 ft in 8 hours by the shovel crew, 16 men, foreman and one team at a total cost of $34 or 2 12 cents per foot

Culvert Openings. The following run-off formulas for culvert openings are taken from the report of Committee on Drainage of the Illinois Society of Engineers and Surveyors, 1913 In these formulas, M = area of watershed in square miles; A = area of watershed in acres; Q = maximum discharge entire watershed in cubic feet per second; q = maximum discharge per square mile in cubic feet per econd, c = coefficient; a = area of opening required in square feet:

TALBOTT $a = c\sqrt[4]{A^3}$. $c = 0\ 6$ for flat land; $c = 0\ 85$ for moderate slope, $c = 1.1$ for steep slope

MYERS $a = c\sqrt{A}$. $c = 1$ for flat land; $c = 1.6$ for hilly land; $c = 4$ for mountains

PECK· $a = \frac{A}{c}$· $c = 4$ to 6 (Missouri Pacific Ry)

WENTWORTH· $a = A^{2/3}$. Applicable to conditions on Norfolk & Western R. R.

C. B &Q $Q = \dfrac{3000M}{3 + 2\sqrt{M}}$

COOLEY. (1) $Q = 200M^{2/3}$, (2) $Q = 180M^{2/3}$
TIDEWATER RY CO: $a = 0.62A^{7/10}$

EL PASO & SOUTHWESTERN RY $Q = 12\sqrt{\frac{8000}{M}}$

O'CONNELL: $Q = \frac{1421M}{0.311\sqrt{M}}$

KUICHLING For occasional floods: $q = \frac{44,000}{M + 170} + 20$;

for rare floods: $q = \frac{127,000}{M + 370} + 7.4$

MURPHY. $q = \frac{46,790}{M + 320} + 15$

GRAY. $q = 5.89M^{4/5}$

FANNING $Q = cM^{5/6}$ c = 130 to 200 for New England and Appalachian watersheds; c = 60 to 100 for eastern middle states watersheds, c = 12 to 50 for western tributaries of Mississippi River north of Missouri River.

Wooden Trestles. The reasons for the common use of wooden trestles on new work are summarized in an editorial in the *Engineering News* as follows

1 A well-built timber trestle, while it lasts, is a very solid and safe structure, and it lasts normally in good condition for from 5 to 10 years while much hastily built masonry gives out in 1 or 2 years

2. There is more time to determine accurately the size of opening needed and thus avoid needless washouts, besides, well-built timber structures are less likely to wash out suddenly

3. The time of construction is shortened materially, often an important consideration

4 The masonry, when at last built, is almost certain to be better built and of better stone. Haul then is of less importance and there will be more time to secure good materials. The roads are few on which any large proportion of the original masonry is in good condition after 10 years. This is especially true of the smaller structures, such as cattle guards and open culverts which are often so poor as to shake to pieces in a few months. The lesson that the smaller the structure, the larger and better dressed must be the stones composing it, if it is to be durable, is one which engineers are slow to learn

5 It is easier to introduce long and high fills afterward to be filled by train, or replaced by masonry or iron, and thus to secure a better alinement and avoid rock cutting or other objectionable work.

6. A very large part of the total cost of the line in its permanent form is postponed for 6 to 8 years past the trying years of early operation, thus not only saving the interest on the cost of the permanent work but going far to protect the company from the danger of early insolvency, which has proved so deadly to many overconfident companies

7. The only necessary disadvantages are the liability to decay and fire. To guard against the former is a mere question of inspection. The danger from fire is a real one and every year has its record of accidents resulting therefrom, but if the danger is real it is small. There are few such accidents and those mostly from gross carelessness. In proportion to their number, accidents from iron structures have been vastly more numerous and more fatal, and the same is true in substance of small masonry structures where the great liability to washouts is a serious matter.

The reasons given above will apply to-day in sections where timber is cheap, the country and traffic undeveloped, and the company with scarcely sufficient means to put the road in operation. In improvements in alinement and gradients, or in building extensions and branch lines in a fairly well-developed country by a prosperous, well-established company, the conditions are different and the tendency is toward masonry structures with solid floors so as to give a continuous ballasted roadbed. In improvements under heavy traffic, as in track elevation or depression, grade reduction, etc., timber trestles are usually necessary to carry the track until the permanent roadbed is completed.

Quantities of Material and Cost of Wooden Trestles. The editors of *Engineering-Contracting* from a carefully prepared and tabulated bill of materials for the Northern Pacific Railway standard wooden trestle deduced the following formulas for preliminary estimates:

$$\begin{aligned} M &= 220 + 6H \text{ for } H \text{ between 20 and 25} \\ &= 240 + 8H \text{ for } H \text{ between 25 and 50} \\ &= 240 + 9H \text{ for } H \text{ between 50 and 75} \\ &= 240 + 10H \text{ for } H \text{ between 75 and 125} \end{aligned}$$

where M = feet B.M. in trestle, including deck, per lineal foot.
H = average height from ground to a point $3\frac{1}{2}$ ft. below base of rail.

The division into groups is due to the construction of high trestles in stories, each story being about 25 ft. The bents are 15 ft. 9 in. centers. Each has four 12 by 12-in. posts, the outside posts having a batter of 3 in. per foot and the inside posts a batter of 1 in. per foot. The deck consists of six 9 by 18-in. stringers, with 8 by 8-in. cross ties $13\frac{1}{2}$-in. centers, and 5 by 8-in. guard rails, a total of 164 ft. B.M. per lineal foot of trestle. For the deck there are 40 lb. of wrought iron, 25 lb. of cast iron, and 25 lb. of galvanized iron, a total of 90 lb. per 1000 ft. B.M. of timber, or 15 lb. per lineal foot of trestle. For the bents and braces there are about 35 lb. of wrought iron and a little less than 15 lb. of cast iron per 1000 ft. B.M., a total of 50 lb. per 1000 or 0.05 lb. per foot B.M. of timber. This would give in pounds:

$$\text{Iron per foot of trestle} = 15 + 0.05\,(M\text{-}164).$$

If piles are used under the sills as for the Sante Fe, five would be required for the heights up to 25 ft. and six above that height. The average penetration will be from 12 to 18 ft., depending on the soil, requiring about 20-ft. piles.

For a pile trestle, four piles per bent, bents 16-ft. centers, with 20 ft. per pile allowed for penetration and cut off, lineal feet of piles per foot of trestle = $(20 + H)/4$, where H is the average height in feet to a point 3½ ft. below base of rail.

The sawed lumber per foot of trestle = 185 ft. B.M. for trestles under 15 ft. high.
= 200 ft. B.M. for trestles 15 to 25 ft. high.

The iron weighs 16 lb. per foot of trestle; 40 per cent is wrought, 30 cast and 30 galvanized.

With bridge carpenters at $2.50 per day it is stated that the cost for framing and erection, including the handling of the iron, should rarely exceed $10 per 1000 ft. B.M., while the cost of driving the piles is placed at 7 cents per lineal foot of pile (not per lineal foot of penetration). Freight or freight and cartage must be added to the cost of the material if delivery was not included in the purchase price.

Steel Trestles. The general type of construction is with spans alternately about 30 and 60 ft. without much regard to height of trestle; longitudinal bracing is placed under the 30-ft. spans joining the vertical bents in pairs forming towers capable of resisting the longitudinal forces due to starting and stopping trains on the track. The open deck is carried by plate girders spaced to support the ties on the upper flanges; the bents are in vertical planes and the posts batter about 2 in. per foot. Each post rests on a masonry pier some 4 to 5 ft. square at the top and large enough at the base to reduce the unit pressure on the foundation to a safe value. Anchor bolts are set to templet before the pier is built and masonry thus adds to stability in the case of wind pressure strong enough to produce tension in the post.

The following formulas have been given for weight per foot of steel in terms of height of trestle:

1. C. P. Howard, *Eng. News*, 1906, for Cooper's E 40 loading.

Weight per foot = 520 lb. for height of 20 ft.,
= 1200 lb. for height of 60 ft.,
= 1530 lb. for height of 90 ft.

2. The above values with 20 per cent added for Cooper's E 50 loading.

3. Editors, *Eng.-Cont.*, 1907, for two 116-ton engines followed by 3000 lb. per foot, spans 30 and 60 ft. alternating.

Weight per foot = 600 + 12 times height.

It is claimed that this has been used in estimating the weight of many viaducts of different heights and has been found to give very close results except for heights as low as 20 to 25 ft.

4. H. G. Tyrell, *Eng. News*, 1900, for two engines weighing 100 tons each followed by 4000 lb. per foot. Unit stresses 10,000 and 12,000 lb. per square inch.

Weight per foot:

Deck plate girder = 100 + 9 times span,
= 550 for spans of 30 and 60 ft.
Bents and bracing = 9 times height.

5. Electric railway trestles for 25-ton cars, or 2000 lb. per lineal foot.

Weight per foot:

Deck plate girder = 30 + 5 times span,
= 260 for spans of 30 and 60 ft.
Bents and bracing = 6 times height.

Combination Highway and Railway Bridges. The following specification of the Massachusetts Public Service Commission is widely used. It is definitely stated as for purposes of design of new bridges or strengthening of old ones, and that it may require modifications in considering old structures and the desirability of continuing them longer in service.

For the track load these specifications use a 50-ton car with wheel spacing of 5 ft., 15 ft., 5 ft. and a total length of 40 ft. over all. For roadway and sidewalks loads of 100 lb. per square foot are used for city bridges and 80 lb. per square foot for country bridges 100 ft. or less in length. These uniform loads are assumed to cover the full area of the roadway and sidewalks except a width of 9 ft. at each track. For longer spans these uniform loads are reduced 1 lb. per square foot for every 5 ft. additional length up to 200 ft., and for all greater lengths 80 and 60 lb. per square foot respectively are used. For suburban bridges the floor is designed for the same loads as the city bridges, while trusses and girders are designed as for country loads. For highway bridges in city, town or country the specifications require provision for an alternative roadway load of a single 20-ton auto truck on two axles 12 ft. on centers and wheels at 6-ft. gage; the weight assumed to be distributed 6 tons on one and 14 tons on the other axle; the truck assumed to occupy a floor space of 32 ft. long and 10 ft. wide, the overhang being equal at front and back and at the sides. With track and uniform roadway and sidewalk loads impact of 25 per cent is added for floor beams and stringers, while for girders and members of main trusses the impact used varies from 25 per cent to 10 per cent according to the loaded length producing maximum live-load stresses, except that 40 per cent is used for counters and floor-beam hangers. With the auto-truck load 50 per cent impact is used for steel members which receive their full load from one panel point only and no impact is used for *wood floor or stringers. Tension stress allowed by these specifi*cations is 16,000 lb. per square inch of "structural steel." Other allowed unit stresses in general correspond with those given in other specifications using the same tensile stress, except that in direct compression these specifications allow only 12,000 lb. per square inch of steel, reduced by the Gordon formula.

Ballast. R. C. Cram lists the materials used for ballast in order of desirability as follows: (1) Broken stone; (2) coarse slag; (3) screened and washed gravel; (4) chats, granulated slag and disintegrated granite; (5) burnt clay; (6) bank-run gravel; (7) cinders; (8) chert and cementing gravel; (9) sand; (10) shells, and (11) earth. Gravel is the material most commonly used both on steam and electric railways, but in city tracks crushed stone is used to a considerably greater extent than gravel. In selecting ballast the first consideration should be to obtain a material as free from clay and loam as possible, in order to afford an opportunity for water to drain off rapidly. Crushed stone possesses most of the qualities of an ideal ballast, while screened and washed gravel is a fairly close second, and is quite extensively used in city track work. However, bank-run gravel is in general use, due to its availability and comparatively low first cost.

Crushed stone should not be smaller in size than will pass through a ¾-in. ring nor larger than will pass through a 2½-in. ring. Larger sizes have too many voids, while the smaller sizes wear the ties less, are less noisy, more easily tamped and give a better surface with less labor.

Gravel varies greatly in quality in different localities, and bank-run gravel will vary in clay, sand and gravel as follows: Dust and clay, 0 to 20 per cent; sand, 5 to 60 per cent, and gravel, 35 to 90 per cent. Good gravel ballast should not contain more than 10 per cent clay and 20 per cent sand, as greater proportions of these materials seriously interfere with drainage. It has been stated that ties will last from two to three years longer in washed gravel ballast as compared with bank-run; this is attributed to the lesser amount of water retained.

Slag, being available only in the vicinity of blast furnaces, has not been used as ballast by electric railways generally. Coarse slag, however, is said to be almost as durable as crushed stone and to equal it in many ways. In certain cases it is stated that dry rot attacks the ties more rapidly in slag than in stone ballast. Granulated slag is much inferior to the coarse variety, but is still superior in drainage qualities to cementing gravel, cinders or sand.

Chats, disintegrated granite, burnt clay, sand and shells are not much in use for ballast on electric railways, but cinders are in quite general use, largely through their availability as a by-product from power stations. There is a wide divergence of opinion as to their value, due to the claim that sulphur in them tends to decrease the life of the ties. Cinders stand eighth in the order of desirability for ballast materials, but when of good quality they serve a good purpose as sub-ballast and for ballast in yards and sidings or upon main lines having comparatively light traffic.

The depth of ballast depends upon the nature of the soil in the subgrade, the size and spacing of ties, the strength of the rail as a beam, the weight of the wheel load and the number of loads. The recommendations in the A.R.E.A. Manual for 1915 are as follows: Minimum depths of ballast for Class A traffic, 12 in.; for Class B traffic, 9 in.; for Class C traffic, 6 in. The recommended minimum depths given in the 1916 A.E.R.E.A. way committee report are as follows:

Ballast material	Main tracks	Side tracks and yards
Broken stone	8 in.	6 in.
Washed gravel	8 in.	6 in.
Bank run gravel	12 in.	8 in.
Cinders	12 in.	8 in.

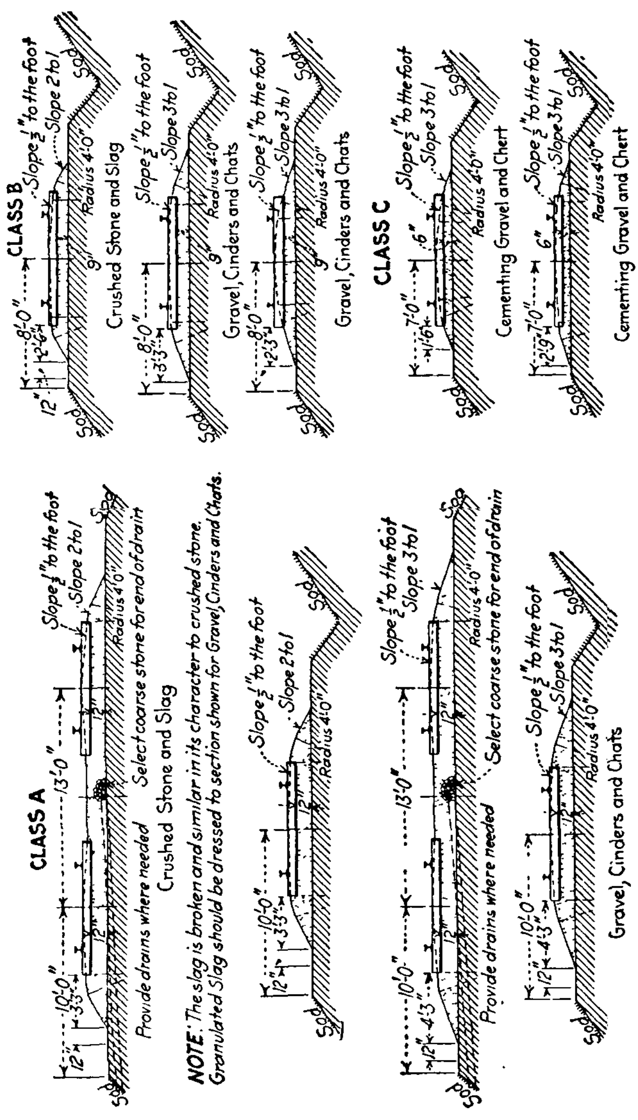


FIG. 2.—Ballast sections. (From *Manual of Am. Ry. Eng. Assn.*)

Ties. The following table, from information by the Government Forest Service, shows the proportions of different species of woods purchased for ties by electric railways in 1915:

Kind of wood	Per cent
White oak	26.7
Chestnut	21.8
Cedar	11.6
Southern pine	10.3
Red oak	10.3
Douglas fir	7.4
Redwood	3.3
Western yellow pine	2.5
Cypress	1.2
Eastern tamarack	1.0
All others	3.9
Total	100.0

White-oak timber is by far the best and most largely used. It is but seldom treated, as it has been found that it is hardly economical to do so, owing to its very high resistance to decay. The average life of white oak under heavy steam-road traffic is about nine years, while this is extended to from twelve to fifteen years for moderate traffic. Bur oak, rock oak and chestnut oak will last from six to eight years. Other species of oak, such as black and red oak, pin or swamp oak and water oak are inferior woods and have a life of from four to five years if untreated. The several species of pine are used on steam roads in quantities second only to oak, but chestnut has taken second place on electric lines and the pines take third place. While it is a soft wood as compared with oak, pine is quite slow in decaying and long-leaf heart pine will average seven years and has been known to last twelve years. Some pines, if high in pitch, will check badly, but long-leaf yellow pine is much to be preferred for bridge timbers and bridge ties, since it does not warp as much as oak. Chestnut is not used much for bridge timber because of its tendencies to split and check, but for ties it is nearly as durable as oak, having a life untreated averaging seven years. Cedar is a durable species of soft wood and will resist decay for from twelve to fifteen years, but it is apt to fail from spike driving and nail cutting; it has an average life of ten years. Hemlock is a soft wood and is very short-lived when untreated, averaging not over four years; its use continues to a considerable degree because of its cheapness. Tamarack and spruce have characteristics quite similar to hemlock and cost about the same but have an average life of from five to six years. Red and black cypress are soft woods, largely used in the south, and they decay rather slowly. Cypress has an average life of nine years. In California, redwood is used to a large extent. It is classed as a soft wood which resists decay quite well, lasting five years untreated and without tie plates and twelve years when used with tie plates and treated. The foregoing information on life of ties is based on steam road conditions.

The manner in which the tie is cut out of the tree is generally the basis for defining its kind (Fig. 3). A tie cut from a tree from which not more than one tie can be produced from a section is called a "pole" tie and it is hewed or sawed on two parallel faces. When made from a tree of a size that two or more ties can be made from a section by splitting, the tie is called a "split" tie. An inferior tie, named a "slab" tie, is sometimes made from the first or outside cut of a log. A sawed tie has the two sides and two faces sawed. The upper or lower plane surface is called the "face." A "quartered" tie is one made from a tree of a size to yield four ties per section. A "slabbed" tie is one sawed on only two faces. If the two faces are of equal width, a slabbed tie is also a "pole" tie, but should the lower face be wider than the upper, it is called "half-round" tie. A "hewed" tie must be hewed on at least two surfaces other than the ends. Tie specifications always limit the amount of sapwood, and if the section shows more than the specified amount the tie is called a "sap" tie. If the specified amount of sapwood is exceeded on only one or two corners, but does not measure more than one inch on either corner measured diagonally across the tie, it is classed as a "heart" tie. An "all-heart" or "strict-heart" tie has no sapwood. A "wane" tie is made from a tree too small to make a pole tie, by allowing the original surface of the tree to show on one or more corners. When a tie has been made from a tree from which the resin or turpentine has been extracted before felling, it is called a "tapped" tie. Ties which do not conform to the specifications are "cull" ties.

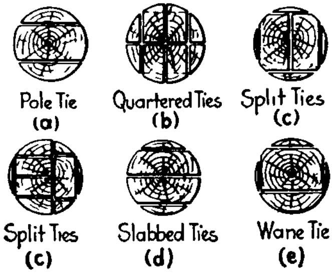


Fig. 3.—Types of ties.

The thickness of ties varies from 6 to 8 in.; the width from 6 to 12 in., and the length from 8 to 10 ft., the greater length being for bridge ties and tracks over marsh land. Electric railways commonly use a size of 6 in. × 8 in. × 8 ft., while 7 in. × 9 in. or 7 in. × 8 in. × 8 ft. to 8 ft. 6 in. long are sizes being used more and more by steam roads. Spacing should be considered as of more importance than size. Two advantages are obtained by decreasing the spacing: the unit pressure on all track material is decreased, and the carrying capacity of the roadbed is increased correspondingly. The minimum spacing should not be less than the width of track shovels used. The usual spacing varies from sixteen to eighteen for a 30-ft. rail or eighteen to twenty for a 33-ft. rail, indicating a variation of from 2640 to 3200 per mile. Bridge ties are usually 8 in × 8 in. × 10 ft. spaced from 12 to 16 in. centers.

Treatment of ties for wood preservation is without doubt an economical measure for increasing life and thus reducing maintenance charges, inasmuch as the labor cost is now so great a proportion of the total cost of tie renewal.

Fences. The American Railway Engineering Association Manual gives specifications for three classes of smooth wire fences

4½ ft. high with wooden posts. Preference is given to smooth wire, but if barbed wire is used, a heavy smooth wire, or a plank at the top of the fence is recommended. For the three classes of smooth wire fence, galvanized No. 9 gage is used throughout except for the top and bottom longitudinal wires of Class 1 which are No. 7 gage. The longitudinal wires are all coiled; the spacing, commencing at the bottom, is Class 1: 3, 4, 5, 6, 7, 8, 9 and 9 in.; Class 2: 5, 6½, 7½, 9, 10 and 10 in.; Class 3: 14, 14 and 14 in. The bottom wire is to be placed above the ground 3, 6 and 12 in., respectively, for the three classes. The stay wires are spaced 12, 22 and 22 in., respectively. Intermediate posts are to be 8 ft. long and not less than 4 in. in diameter at the small end, and end posts 9 ft. long and 8 in. in diameter; round posts are preferred. The posts are to be set with the large end down, the end posts 4 ft. deep and the intermediate ones 3 ft., with spacing from 16½ to 33 ft., depending upon the nature of the ground and the service required. Gates are necessary at farm or private crossings.

In Bulletin No. 144 of the Railway Engineering Association, it is stated that the tendency to use reinforced concrete posts is increasing and that the figures prevailing for the most popular form now on the market are from 18 to 22 cents. The prevailing cost for wood posts of the most durable kinds of timber native to the road is from 12 to 15 cents. Several forms of metal posts are being made, and it is claimed by a large manufacturer that they will have a life of at least 30 years and can be delivered at reasonable distances for 23 cents f.o.b. line of road.

Camp, Track, estimates that under average conditions the labor of building a mile of barbed wire fence four strands high, posts 16 ft. apart, is about 13 days work (10-hr. day); with posts 12 ft. apart, 16 days; with top board and four wires, posts 12 ft. apart, 18 days. For a fence with a different number of wires allow about 8 hours labor for each wire. Experienced fence men working by contract will build about 50 per cent more fence per day than the same number of ordinary track laborers engaged on the work only a short time each season. The average cost for labor in erecting 22 miles of Page woven wire fence, posts 17 ft. apart and set 3 to 3½ ft. in the ground, was 17.2 cents per rod as shown by the report of the fence gang of a certain railroad. The surface was generally rough and uneven and a great many anchor posts had to be used. The cost stated covered the labor of loading and unloading new material, removing the old fence and piling or burning it, and the time used in moving the fence gang from point to point.

Snow Fences. Where much trouble is experienced from drifting snow, snow fences have been extensively used to protect cuts and other places where snow accumulates. These snow fences may be installed permanently or may be made of the portable type. The standard portable snow fence of the New York State Railways is shown in Fig. 4, from the *Electric Railway Journal*, 1910.

Street Railway Roadbed Construction. The construction of roadbed in highways must necessarily differ materially from construction on private right of way, and as the conditions as to subsoil, highway traffic, pavement, etc., vary so widely, it is not possible to standardize such construction to so great an extent as has been done in the case of the private right-of-way roadbed. A careful study is necessary with respect to the bearing value of the soil in the

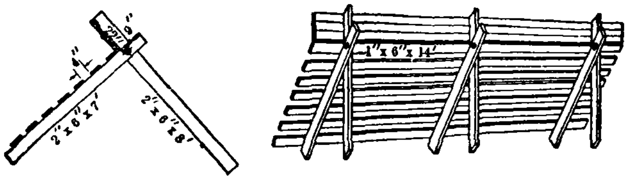


FIG. 4.—N. Y. State Railway's portable snow fence.

street, and upon this will depend not only the character and depth of ballast, but whether or not some form of concrete foundation is desirable. The subgrade and ballast should be rolled, especially where the street is known to be on made ground or where there has been much disturbance of subgrade due to foreign subsurface con-

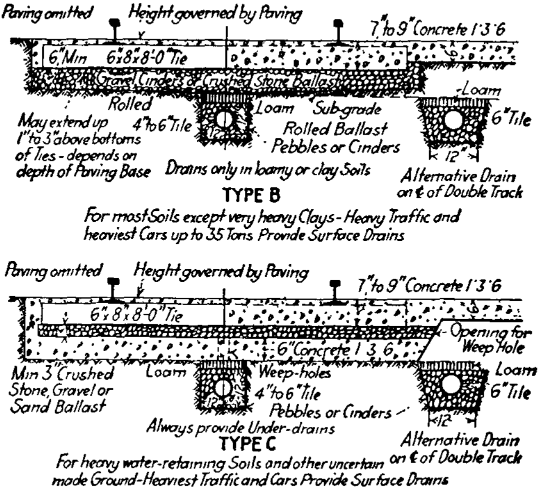


FIG. 5.—Types of track foundations for city streets.

structions and cross trenches. When it is impractical to roll, the subgrade should at least be thoroughly rammed, soft spots eliminated and ballast placed under tamping which should be continued until there is no movement observed under load. When a concrete paving base is to be constructed, the ties should again be tamped just prior to the placing of the concrete. The installation of surface drains and good pavements is necessary in order to prevent the infiltration of surface waters which eventually causes disintegration of all forms of foundation.

Two of the types of foundation construction submitted by the 1914 Committee on Way Matters of the A.E.R.E.A. are shown in Fig. 5. Type B has been adopted as a "Recommended Design." Some city track is being laid on ties, either wood or steel, embedded wholly in concrete.

Data relative to the standard track construction on some of the principal street railways, as compiled by the San Francisco-Oakland Terminal Railways in 1921, are shown in the accompanying tables, pages 32 to 35, inc.

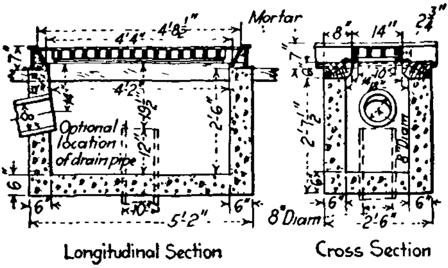


FIG. 6.—Track drain, N. Y. State Rys.

Track Drains. The importance of caring for surface water makes advisable the frequent use of track surface drains in connection with street railway tracks. The type used by the New York State Railways (Fig. 6) is inexpensive to handle and install, takes large amounts of water from along the rail, does not clog easily either on the surface or under ground, and may be cleaned with a shovel.

Selection of Rail. For open track, the selection of rail for electric railways will follow the same general principles as for steam railways, the principal factors being weight of car, axle spacing, tie spacing, kind of ballast, frequency and character of service. However, the loading conditions resulting from the typical steam locomotive driving wheel arrangement are not found with electric locomotives or electric motor cars, the latter having less average wheel loads and better distribution of weight, although the center of gravity may be lower, thus producing greater lateral thrust, especially on curves. The stresses in the rail and the ability of the steel to resist them must be taken into account, as well as the proportions of the rail which must be such as to properly distribute the load to the ties. The load due to movement must be considered as well as the weight of the car at rest; the former is known as the dynamic augment, and is usually taken as 0.7 times the static load for wheel loads less than 15,000 lb., or as 10,000 lb. for wheel loads of 15,000 lb. or more. The Baldwin Locomotive Works' rule for determining weight of rail is: "Each 10 lb. per

Data Covering Standard Track Construction on Some of the Principal Railways Throughout the United States

(Compiled by San Francisco-Oakland Terminal Railways, April, 1921)

City	Type of rail	Type of joint	Tie rod	Tie plate
Albany	122 lb.—7" gir., 95 lb., 7" T.	Continuous	Flat	None.
Atlanta	103 lb.—7" gir., 80, 70 lb. ASCE, 80, 70 lb., 7" T.	Elec. weld, Cont. plates, Thermit.	3/8"×2"	Flat.
Baltimore	122, 105-lb.—7" gir., 100 lb. ARA.	Weld fish and base plates to rail.	None	6"×10" shoulder.
Birmingham	105 lb.—7" gir., 80 lb. ASCE.	Bolt plates	Brace plates	Brace plates.
Boston	132 lb.—9" gir., 122 lb., 7" gir., 100 lb.—6" T.	Lorain, Thermit, arc weld	2¼"×1 13/32"	Flat.
Brooklyn	122 lb.—7" gir.	Cast weld	3/8"×2"	None.
Buffalo	124 lb.—9" gir.	Continuous	Flat	Flat.
Chicago	129 lb.—9" gir., 122 lb., 7" gir., 91 lb., 7" T, 85 lb. ASCE.	Lincoln, Lorain elec. weld	2"×5/16"	½" shoulder.
Cincinnati	140 lb.—9" gir.	Arc welded channels	Rail brace	3/8" flat.
Cleveland	95 lb.—7" T.	Rivet and weld	None	None.
Columbus	122 lb.—7" gir.	Columbus joint, arc weld	Only in macadam	½"×6½" under spec. wk.
Conn. Co.	9" gir., 95 lb.—7" T, 80 lb.—5" T, 100 lb. ARA.	Continuous, arc weld	3/8"×1½"	Lundie, tilted type.
Dallas Ry	103 lb.—7" gir., 80 lb.—7" T.	Apex, Indpls.	None	R. R. S. Co.
Denver	80, 65 lb. ASCE	Cont. arc weld in paved streets.	None	Shoulder.
Detroit	91 lb.—7" T.	Cast weld	3/8"×1¾"	None.
Indianapolis	95 lb.—8 29/32" gir., 91 lb.—7" T.	Thermit	3/8"×2"	None.
Kansas City	91 lb.—7" T.	Gailor, Thermit, Lorain, Indpls.	3/8"×2"	None.
Los Angeles	132 lb.—7" gir., 116 lb., 7" gir.	Continuous	None	Brace plates.

Memphis	105 lb.—7″ gir.	Bar with base plate welded to rail.	⅞″ round	⅜″×5½″.
Milwaukee	100 lb. ARA., 95 lb., 7″ T.	Cast weld, thermit, elec. weld.	5/16″×2″	Shoulder.
Minneapolis & St. Paul	91 lb.—7″ T.L.S. Co., 93-507.	Cast weld, thermit, arc weld.	⅜″×1½″	Flat.
N. J. Public Service	116 lb.—7″ gir., 101 lb.—7″ gir.	Lorain elec. weld	⅜″×2″	Brace plates.
New Orleans	105 lb.—7″ gir., 100, 80 lb. ASCE.	Channels on gir., angles on T.	⅞″ round	Brace plates.
Oakland	141, 106 lb.—9″ gir.	Continuous	⅜″×2½″	Shoulder.
Omaha	97 lb.—7″ gir.	Cont., thermit, arc weld	None	Brace plates.
Philadelphia	141 lb.—9″ gir.	Nichols zinc & elec. weld	None	None.
Pittsburgh	134 lb.—9″ gir.	Thermit	⅜″×2″	Pl. flat sh.
Portland, Ore	80 lb.—7″ T, 72 lb.—6″ T.	Continuous	None	None.
St. Louis	132 lb.—9″ gir., 103 lb., 7″ gir., 100 lb. ARA.	Nichols	Round and flat	Flat.
Washington (Cap. Trac.)	80 lb.—5″ T.	Bolted	⅜″×1½″	None.

Data Covering Standard Track Construction on Some of the Principal Railways Throughout the United States—(*Concluded*)

(Compiled by San Francisco-Oakland Terminal Railways, April, 1921)

City	Cross ties	Ballast	Drain tile	Type of paving
Albany	Yellow pine	Cr stone & concrete	None	Granite block
Atlanta	6"×8" sap pine creosoted	Crushed granite, 6" deep	6" tile	Granite asphalt wood or brick on concrete
Baltimore	6"×8"–8' untreated wood	6" cr stone under ties	6" & 8" tile	Sheet asphalt gran blkliners Gran blk paving
Birmingham	Int twin steel Creo yellow pine	6" cr stone under ties open trk 6" slag conc under ties paved trk	None	Concrete Brick on concrete with asph filler
Boston	So pine tr Roeping proc Int twin steel	Broken stone 4"–6" under tie	6"–8" tile.	5" gran blk & 3½"×4" wood blk laid on concrete
Brooklyn	Rough sawed yel pine heart	Natural soil in trench	None	Granite blks cement joints
Buffalo	Y pine 90% hrt pvd sts, oak in open trk Some Carnegie M-24	8" concrete sub-base, 2' broken stone	4" tile	Sandstone blk. on 6" concrete base
Chicago	Long leaf y pine, $45 Some Carnegie M-25	Concrete 6" under tie Spec wk 8" cr stone under tie	None	Granite blk.
Cincinnati	Oak 2' centers Carnegie M-25 3' centers	Concrete 7" below tie	4" tile	Granite blk.
Cleveland	Cut steel Carn	12" concrete under rail	6" tile	Granite
Columbus	Carnegie M-25, wh. oak spec wk	Concrete and crushed stone	6" tile	Granite blk. Brick.
Conn Co	6"×8"–8' chestnut	6" gravel or cr stone under tie	Some 6" tile	Hassam block, brick and bitumen
Dallas Ry	Long leaf yellow pine	Gravel	4" tile	Bitulithic and brick
Denver	Int twin steel, Ore long leaf	Grav & broken conc pvg base 12"–15" deep	None	Std concrete slab
Detroit	6"×10"–6' 8" # 1 white oak	Concrete slab 8" under ties	6" tile	Gran nose blk adj to gage of rails brick between.

Indianapolis	6″×8″–8′ white oak	6″–8″ conc slab 3″ dry mix conc ballast	Some 4″ tile	Gran block, brick
Kansas City	Int twin steel untreated wh oak	Cr stone 6″–8′ deep. Solid conc 6″ deep	6″ tile	Spec dressed gran blks. Brick gran flwys & conc
Los Angeles	6″×8″–6′ redwood	6″ cr stone under tie	4″ tile	Asphalt on concrete
Memphis	Creo pine white oak	6″–8″ cr stone	6″ tile	Asphalt brick & wood blk
Milwaukee	Long leaf y pine, wh oak, steel	Cr stone 8″ deep	6″ tile	Various city requirements
Minneapolis & St Paul	6″×8″–8′ sawed	6″ cr stone under ties	None	Granite blk preferred
N J Public Service	Long leaf y pine creo, oak	6″ stone	Some 4″ & 6″ tile	Gran blk with cement grout
New Orleans	Long leaf y pine creo Int twin steel	Conc with gir rail Cr stone with T-rail	Small amt	Gran & wood (creo) blk
Oakland	6″×8″–8′ redwood	9″ cr stone under tie	6″ tile	1½″ asphalt on conc
Omaha	6″×8″–7′ red oak tr with creo	6″ cr stone Conc slab where necessary	Small amt 6″ tile	Vit brk blk, gran blk, conc base
Philadelphia	5″×9″–8′ yellow pine	None in city work	None	Gran blk on concrete
Pittsburgh	6″×8″–8′ wh oak	8″ cr stone under tie	6″ tile	Block stone.
Portland Ore	Fir	6″ conc slab with 2″ of 1″ stone	4″ tile	Concrete.
St Louis	Oak. Steel ties in spec const	Solid conc in pavemt Stone in unpvd sts.	6″ tile.	Concrete, brick. Wood blk under protest
Washington (Cap. Trac)	Oak and chestnut	Stone	6″ tile	Asphalt and macadam

yard of ordinary rail steel, properly supported by not less than 14 ties per 30-ft. rail, is capable of supporting a safe load per wheel of 2240 lb." R. C. Cram has suggested that in the application of this rule to an electric railway, the dynamic augment should be added to the static load.

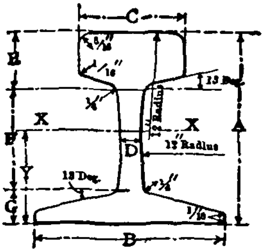


FIG. 7.—A.S.C.E. rail section.

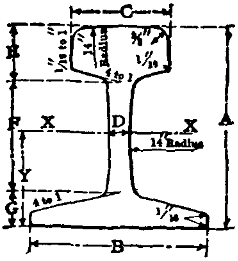


FIG. 8.—Am. Ry. Eng. Assn. rail section.

For use in public streets, a 7-inch girder rail usually will be necessary to care for paving requirements satisfactorily. With modern joints and paving blocks not over four to five inches deep, the 9-inch girder rarely will be required, as the 7-inch rail with proper ties spacing will carry very heavy static and dynamic

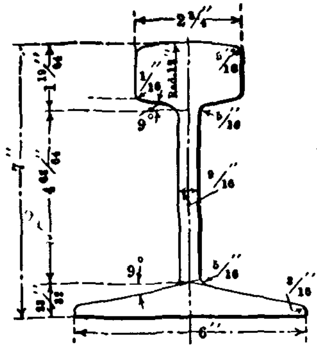


FIG. 9.—Am. El. Ry. Eng. Assn. 91-lb., 7-in. T-rail.

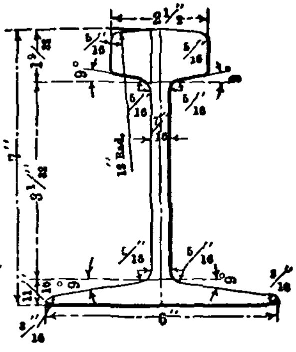
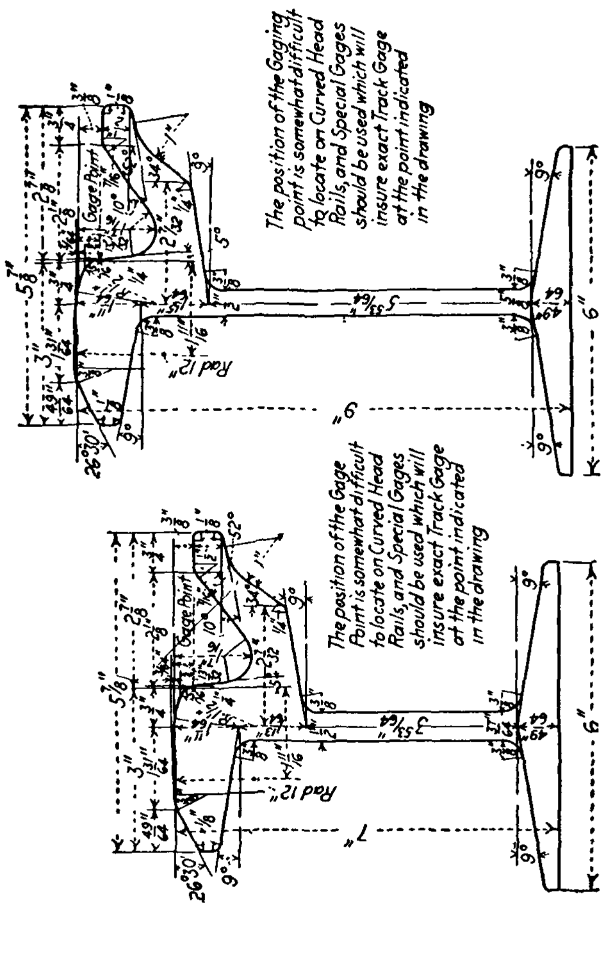


FIG. 10.—Am. El. Ry. Eng. Assn. 80-lb., 7-in. T-rail.

loads. Where shallow pavement is used, the standard T-rail may be used with economy. The recommendations of the Am. El. Ry. Eng. Assn. are as follows:

The selection of plain girder rails for use in paved streets requires the most careful consideration of the type of pavement to be installed and the vehicular traffic to be sustained. Particular care should be taken with respect to the use of standard section rails in pavements, as it will often be found that plain girder rails of equal weight are much better adapted to a greater range in types of pavement which may be selected for use therewith. For use where the type of pavement will permit, as with macadam or other shallow pavements in wide streets having moderate vehicular traffic, three standard rails are recommended. These are shown in the table on page 39, and weigh 80, 90 and 100 lb. per yard, respectively.



FIGS. 11 and 12.—Am. El. Ry. Assn. Standard girder grooved rails.

The first is the A.S.C.E. standard 80-lb. rail, while the others correspond to the Am. Ry. Assn. old standard Series A 90 and 100-lb. rails. The three types are recommended in order to provide for varying degrees of rail service as may be desired. For use in light service with deep pavement, a 7-in. plain girder rail, weighing 80 lb. per yard, as shown in drawing in Fig. 10, has been adopted.

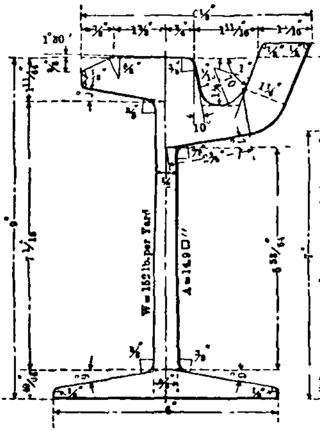


FIG. 13.—Am. El. Ry. Eng. Assn. 9-in girder guard rail. American Standard E7–1923.

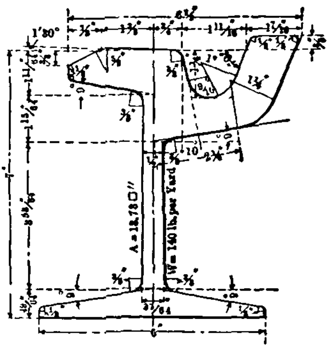


FIG. 14.—Am. El. Ry. Eng. Assn. 7-in girder guard rail. American Standard E6–1923.

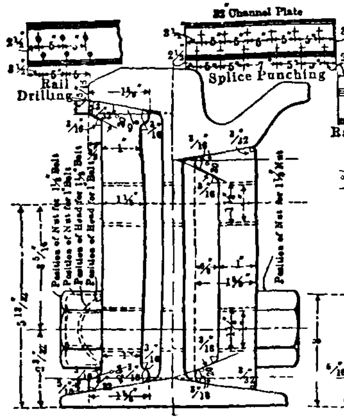


FIG. 15—Joint plates for Am. El. Ry. Eng Assn. 9-in. girder rails. American Standard E3–1923.

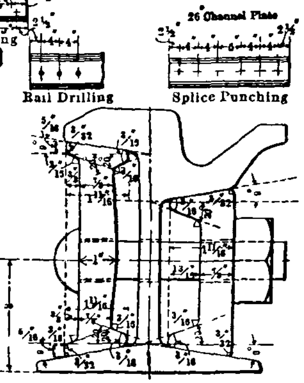


FIG. 16—Joint plates for Am. El. Ry. Eng. Assn. 7-in girder rails. American Standard E2–1923.

This section is identical with L.S. Co. Section No 80-335 and B.S.Co. Section No. 277. For use in heavy service, in connection with deep block pavement, a plain girder rail 7 in. in height, weighing 91 lb per yard (Fig. 9) is recommended. This section is identical with L.S. Co Section No 91-375 and B.S. Co. Section No. 282. For use in heavy service in connection with deep block pavements in the congested sections of narrow city streets where the vehicular traffic is largely confined to the pavement area to be maintained by the railway, which conditions exist, as a rule, only in cities of the largest class, the committee recommends the use of the Association Standard 7 and 9-in. girder grooved rails, as shown in Figs 11 and 12. The corresponding girder guard rails are as shown in Figs. 13 and 14, and the girder rail joint plates in Figs. 15 and 16.

Standard T Rails

	A	B	C	D	E	F	G	Percentage of metal			Moment of inertia about X − X	Y
Pounds per yard	Height of rail	Width of base	Width of head	Thickness of web	Depth of head	Height of web	Height of flange	Head	Web	Base		
American Society of Civil Engineers (Fig. 7)												
100	5 3/4	5 3/4	2 3/4	9/16	1 45/64	3 5/64	31/32	42	21	37	43 8	2 8
95	5 9/16	5 9/16	2 11/16	9/16	1 41/64	2 63/64	15/16	42	21	37	38 6	2 7
90	5 3/8	5 3/8	2 5/8	9/16	1 19/32	2 55/64	59/64	42	21	37	34 0	2 5
85	5 3/16	5 3/16	2 9/16	9/16	1 35/64	2 3/4	57/64	42	21	37	30 0	2 5
80	5	5	2 1/2	35/64	1 1/2	2 5/8	7/8	42	21	37	26 2	2 4
75	4 13/16	4 13/16	2 15/32	17/32	1 27/64	2 35/64	27/32	42	21	37	22 9	2 4
70	4 5/8	4 5/8	2 7/16	33/64	1 11/32	2 15/32	13/16	42	21	37	19 6	2 2
65	4 7/16	4 7/16	2 13/32	1/2	1 9/32	2 3/8	25/32	42	21	37	16 9	2.2
60	4 1/4	4 1/4	2 3/8	31/64	1 7/32	2 17/64	49/64	42	21	37	14 5	2.1
55	4 1/16	4 1/16	2 1/4	15/32	1 11/64	2 11/64	23/32	42	21	37	11 9	2 0
50	3 7/8	3 7/8	2 1/8	7/16	1 1/8	2 1/16	11/16	42	21	37	9 8	1.9
45	3 11/16	3 11/16	2	27/64	1 1/16	1 31/32	21/32	42	21	37	8 0	1 8
40	3 1/2	3 1/2	1 7/8	25/64	1 1/64	1 55/64	5/8	42	21	37	6 6	1 7
American Railway Engineering Association (Fig 8)												
90	5 5/8	5 1/8	2 9/16	9/16	1 15/32	3 5/32	1	36 2	24 0	39 8	38 7	2 35/64
100	6	5 3/8	2 11/16	9/16	1 21/32	3 9/32	1 1/16	38 2	22 6	39 2	49 0	2 3/4
110	6 1/4	5 1/2	2 25/32	19/32	1 23/32	3 13/32	1 1/8	37 4	23 0	39 6	57 0	2 53/64
120	6 1/2	5 3/4	2 7/8	5/8	1 25/32	3 17/32	1 3/16	37 1	22 7	40 2	67 6	2 59/64
130	6 3/4	6	2 15/16	21/32	1 27/32	3 11/16	1 7/32	36 4	23 8	39 8	77 4	3 1/32
140	7	6 1/4	3	11/16	1 29/32	3 27/32	1 1/4	36 3	24 1	39 6	89 2	3 9/64
American Electric Railway Engineering Association												
100*	6	5 1/2	2 3/4	9/16	1 9/16	3 3/8	1 1/16	36 9	23 4	39 7	48 9	2 3/4
90*	5 5/8	5 1/8	2 9/16	9/16	1 15/32	3 5/32	1	36 2	24 0	39.8	38 7	2 35/64
80†	5	5	2 1/2	35/64	1 1/2	2 5/8	7/8	42 0	21 0	37 0	26 7	2 3/8

* Fig. 8. † Fig. 7.

As to the relative merits of the plain girder rail for use in paved streets, either can be used satisfactorily as far as car operation goes, and pavement can be installed with plain girder rails in such a manner as to be unobjectionable to teamsters. Less wheel flange wear obtains with plain girder rails but in some cases this may be offset by the less paving maintenance costs with groove girder rails. A groove girder rail of proper design may weigh 122 lb. per yard where a plain girder of ample capacity for the same loads will not weigh over 91 lb.; the price per ton being the same for both, the plain girder will cost less per foot of track. Based on wear of pavement alone, the groove girder rail is preferable for use in narrow streets of large cities under heavy steel-tired wagon traffic which is largely confined to the railroad pavement area. When the plain girder (high-T) rail can be used in small cities and towns where wide streets

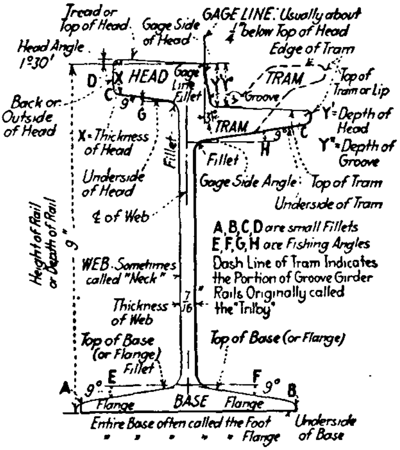


FIG. 17.—Rail nomenclature.

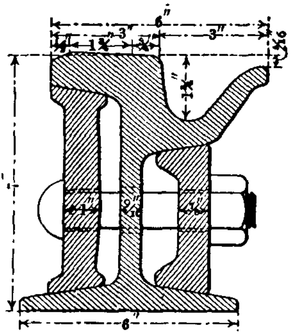


FIG. 18.—Pacific Electric grooved rail for M.C.B. flanges.

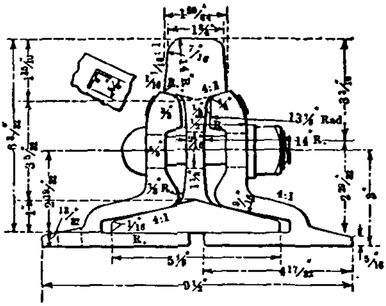


FIG. 19.—Special "frictionless" rail for inside of sharp curves; So. Pac. Ry.

permit the wagon traffic to keep away from the tracks, there is an increasing tendency toward the use of standard section (low-T) rails weighing 100 lb. per yard and of a depth of about 6 in. These rails can be purchased at prevailing prices for standard section rails which are much less than the prices for plain girders (high-T). Incidentally the 6 in.-100 lb. low-T rail will permit the use of many types of pavement which are suitable for moderate traffic.

There seldom should be occasion for the adoption of a section other than one of the A.R.E.A. or A.E.R.E.A. standards, except possibly in the need for a groove girder rail to carry M.C.B. wheels, as the A.E.R.E.A. groove girder rail provides little head wear when used with M.C.B. wheel flanges. Fig. 18 shows a groove rail designed by the Pacific Elec. Ry. for use with M.C.B. flanges.

Special Rail Head Section for Curves. Special rails for curves have been designed mainly to give additional metal available for wear. The Manning rail, tried some years ago on the Baltimore & Ohio R. R., had $\frac{1}{32}$ in. of additional metal on the inner side of the head and $\frac{1}{32}$ in. less on the outer side. The special 110-lb. rail of the Lehigh Valley R. R. has the head slightly wider and considerably deeper than that of the standard rail. The special feature of the so-called "frictionless" rail (Fig. 19) is a very narrow head, and its purpose is to reduce the slip of the inside wheel, which takes place in compensating for the greater length of travel of the wheel on the outside rail. It is claimed that there is a diminution of friction and resultant wear to both the outer and inner (frictionless) rails and the wheel flanges, while the reduced friction gives a freer and smoother passage of the wheels, with a reduction in power required to handle a given tonnage.

Rail Renewals. There is little or no standard practice regarding the limits of wear for the various types of girder and high T-rail. Allowable rail wear has been fixed by various engineers at from 35 to 50 per cent reduction in area of the head. Many engineers believe that with the grooved-girder and tram girder rails the limit of wear has been reached when the wheel flanges ride on the floor of the groove or tram, but others favor the opposite extreme in permitting wheel flanges to shear the tram completely off. The large investment in track compared with that in wheels might under some conditions dictate a temporary change in the wheel flange contour to prolong the life of track, especially where rolled steel wheels are used, but safety of operation must be a controlling factor. Limits of wear fixed for T-rails are usually based on steam-road practice. Conditions governing the life of T-rail on open track with high-speed trains, however, are not analogous to those in paved streets where speeds are comparatively slow, and the unmodified application of steam-road practice to electric railway tracks in paved streets is in utter disregard of the economics of the problem at hand and results in most extravagant track maintenance methods. Generally speaking, street railways cannot afford to use relay rails, except in temporary work, on account of the expensive construction employed and the difficulty experienced in making repairs, although some relay rails have been taken from heavy trunk lines and laid in extensions of light traffic lines, but usually such practice is economical only if undertaken at the time of pavement renewals. In some cases the limiting factor of the life of the rail has been railway traffic, in others vehicular traffic, in still others, corrosion. Provision has been made in modern rail sections for these and other factors which might influence serviceability. The general adoption of the girder-grooved rail as a substitute for the tram rail with the horizontal wagon-wheel tread is certain, except under extraordinary conditions, to eliminate vehicular traffic as a life-determining factor. The shape of the groove also makes flange riding less hazardous, and the depth of the groove insures a liberal wear value in the head of the rail.

The wear of electric railway rails in streets is dependent upon operating speed, wheel load of equipment, density of car and vehicular traffic, use of brakes, frequency of stops, grades, general alinement with respect to curvature, design of rails, design of wheels, upkeep of wheels, use of sand, cleanliness of streets and manufacture and composition of rails. When pavement renewals become necessary, or the foundations, ties or joints have failed, the question often arises whether it is economical to use the old rails in the new work. Before the economy may be determined, it is necessary to fix some limit of wear, but with a limit fixed, the remaining life of the rail may be estimated by obtaining the average head reduction for the period the rail has been in service. If corrosion indicates that it may limit rail life in advance of wear, the rate may be determined in a similar manner. The condition of the foundation, ties and joints also affects the economy of renewal. When the problem can be decided solely on the basis of economics, it resolves itself into one of balancing interest, depreciation and maintenance of the new rail against the old.

The report of the Am. Ry. Eng. Assn. committee for 1919 states that the average rail wear on a 6-deg. curve is about 100 per cent more than that for a straight rail, and 25 per cent for a 3-deg. curve, the variation being approximately as the square of the degree of curve.

Tilted Rail. For the purpose of obtaining a full line contact between the wheel and rail, some companies have tilted the axis of the rail inward at a slope of 1:25, with good results. The tilting is accomplished by the use of a tapered tie plate, or, where steel ties are used, by bending the channels at the proper points depending upon the gage and type of rails.

Rail Length. The standard length is 33 ft., but 60-ft. rails are used to a considerable extent in paved streets, the increased cost of the latter being offset by the first cost and maintenance of the reduced number of joints. Specifications provide for the acceptance of about 10 per cent of rails of lengths shorter than the standard by whole feet down to about 25 ft. because the cropping of the top of the ingot may prevent the remaining portion from cutting into full rail lengths.

Composition of Rail Metal. The quality of the metal in the finished rail will depend upon the chemical composition, the temperature of rolling and the work put upon the metal during rolling. The chemical composition, to be determined from drillings taken from the ladle test ingot, is to be as follows in the Am. Ry. Eng. Assn. specification for carbon steel rails, 1915:

Elements, per cent	Bessemer process 70–84 lb.	Bessemer process 85–100 lb.	Open-hearth process 70–84 lb.	Open-hearth process 85–100 lb.
Carbon	0 40 to 0 50	0 45 to 0 55	0 53 to 0 66	0 62 to 0 75
Phosphorus, not to exceed	0 10	0 10	0 04	0 04
Manganese	0 80 to 1 10	0 80 to 1 10	0 60 to 0 90	0 60 to 0 90
Silicon, not less than	0 10	0 10	0 10	0 10

The Am. El. Ry. Assn. specifications for carbon steel rails, 1922, contain the following requirements, with the statement that it is desired that the percentage of carbon in an entire order of rails shall average as high as the mean between the limits specified.

Elements, per cent	50–69 lb.	70–84 lb	85–100 lb.	101–120 lb.
		Bessemer steel		
Carbon	0 37 to 0 47	0 40 to 0 50	0 45 to 0 55	0 45 to 0 55
Manganese	0 80 to 1 10	0 80 to 1.10	0 80 to 1.10	0.80 to 1 10
Phosphorus.	0.10	0 10	0 10	0 10
Silicon, maximum .	0 20	0 20	0 20	0 20
		Open-hearth steel		
Carbon	0 50 to 0 63	0 53 to 0 66	0 62 to 0 75	0.62 to 0 75
Manganese	0 60 to 0 90	0 60 to 0 90	0.60 to 0 90	0 60 to 0 90
Phosphorus.	0 04	0 04	0 04	0 04
Silicon	0 20	0 20	0 20	0 20

Carbon increases hardness and tensile strength and decreases ductility. Manganese tends to prevent the coarse crystallization due to phosphorus and sulphur, and raises the critical temperature to which it is safe to heat the steel. The effect of silicon is small although in manufacture it acts like manganese as a flux and tends to prevent injury by oxidation. Phosphorus tends to produce coarse crystallization and hence lowers the finishing temperature. Its effect when cold, up to about 0.12 per cent, is to increase strength and hardness, but it renders the steel brittle under shock and should be kept at the lowest practicable limit. In the Bessemer process an acid lining is used in the converter and this prevents burning out either phosphorus or sulphur. In the open-hearth method a basic lining is used and this permits the conversion of the phosphorus into a slag with lime, and the sulphur with lime and manganese ore. The basic open-hearth method thus allows the use of cheaper ores and the reduction of phosphorus and sulphur to low limits. It also furnishes a more uniform product, as the melt can be sampled and the proportions corrected if found desirable before pouring.

Alloyed Steel Rails. The consensus of opinion seems to be that the use of special alloy steels is not generally warranted for ordinary street service when cost differences are taken into account. In certain special cases, as in subways, on elevated roads or in other locations where the radius of the curves is very short and where the renewal work is expensive, some alloy steel may be warranted. In general, special alloy steel can only be justified when its life is three or more times that of ordinary open-hearth steel. Girder and high T, as well as standard sections, can be easily rolled from the following special steels possessing the general characteristics mentioned.

Kind	Steel containing	General properties anticipated
Titanium........	0.1 per cent metallic titanium.	Less segregation, cleaner metal, hence, longer life.
Nickel..........	3.5 per cent nickel.....	Increased life.
Nickel chrome...	Containing varying percentages—nickel and chromium.	Increased life by being tough and hard.
Manganese... ..	About 12 per cent manganese.	Very tough and hard, cannot be easily cut or drilled, wears slowly.
Electric.........	Made in electric furnace.	Very clean steel, free from impurities, thus adding life.
High silicon.....	About 0.35 per cent silicon.	Increased life. Much used in England.

Ferro-titanium Steel. The addition of the alloy ferro-titanium to either Bessemer or open-hearth steel is to-day the most common method of seeking to prolong the life of steel rails without materially increasing their cost. The alloy is, as the name indicates, composed chiefly of iron and titanium, the latter being a chemical element found in various ores and conspicuous for having great affinity for oxygen. The manufacture of the alloy renders obtainable in it various proportions of titanium, so that a 15 per cent alloy means that nominally there is that amount of titanium present as against 85 per cent of iron, but these figures are not fixed and allowance must be made for the presence of other ingredients, as carbon, aluminum, etc. The theory on which the use of ferro-titanium is advocated is very simple, hinging upon the affinity that titanium has for oxygen, or largely upon the effect of chemical reactions that occur from its addition to the molten steel, resulting in a cleansing, so that the name "scavenger" has often been applied to the alloy. There are two brands of ferro-titanium alloy available for use. The one known as the Rossi process is most frequently used, and differs principally from the other or Goldschmidt alloy in being practically free from aluminum. The latter brand contains from 4 to 6 per cent of aluminum, which is a deoxidizing element often used in casting steel to reduce piping and blow holes. The amount of titanium alloy to be used is a matter of some argument, but in short it may be said that enough should be added to thoroughly saturate the molten metal with titanium. Theoretically, then, a trace of titanium in the finished steel may be regarded as proof that enough has been added to the molten steel to effect complete deoxidation. Recent practice advises the addition of one-tenth of 1 per cent metallic titanium to either Bessemer or open-hearth steel, and while this figure may be taken as a safe minimum, there is abundant reason to think that a larger amount would be more satisfactory, especially under some conditions. The use of the alloy requires close attention to detail in the steel works, and should be attended with such supervision as will insure a strict adherence to the recognized principles and specifications governing its use. Success due to the use of titanium has resulted from a denser, more uniform and homogeneous metal, as might be expected from the nature of its duty as a scavenger. Granted that the metal is clarified by using titanium, the possibility of increasing the carbon content without a material loss of ductility occurs, so that of late many tons of rails have been made containing more carbon than formerly whose wearing qualities are regarded as greatly increased, and with no loss of shock-resisting qualities. Recommendations as to the carbon content as well as to the other usual elements had best be left to the individual case until more definite information has been obtained. When used under proper metallurgical conditions, titanium alloy increases the wearing qualities of rails, sufficient evidence having been produced as a result of careful measurement to justify such conviction. Whether the increased wearing quality is economically profitable to the purchaser, is a different question, depending upon the first cost of the rails, and experience with grooved girder and high T-rails for street and interurban uses, has not been sufficient to make accurate figures obtainable.

Manganese steel is a high-carbon open-hearth steel containing from 12 to 15 per cent of manganese. Steel having such a proportion of manganese present, when quenched in water from a red heat becomes exceedingly hard and tough, but remains sufficiently ductile to resist impact. The manufacture of rails of such composition has attained considerable headway in the last few years, and it is now possible to produce any section if ordered in fair quantity. The process is quite simple, involving principally the addition of a large amount of ferro-manganese to the open-hearth metal. The treatment of the ingots in the soaking pits must be attended with great care, and finally after rolling and sawing to length, the rails must be immediately immersed in a tank of water. Naturally the long lengths become very crooked when thus cooling, so that a greater amount of cold straightening often becomes necessary. A very tough and ductile steel results, the most objectionable feature of which is the impossibility of sawing or drilling it, and therefore, careful ordering to length and punching of all holes is necessary. While the price of manganese steel rails quite precludes their common adoption for straight track, still for curves and in special or hard service track the benefits derived are such as to warrant their careful consideration.

Chrome-nickel steel does not flow under compressive stresses as does manganese steel; the flange-bearing portions of the special trackwork do not cut out as fast and after they are cut out can be replaced by welding. This feature alone adds considerably to the life of special work, for in manganese work after the flange bearing is cut out, pounding begins which loosens the special pieces and ultimately destroys the foundation. Joint plates do not wear into chrome-nickel steel castings as they do into manganese steel, and switch tongues stand up better at the heel because this metal does not flow under the action of the wheels. Joints can be thermit welded and thus do away with this troublesome feature of special trackwork and save the cost of machining to exact dimensions necessary for proper fit of joint plates. Repairs can be made either by electric welding or thermit welding. Chrome-nickel steel can be machined with the ordinary machine shop equipment, and no special grinding machine, etc., are necessary in producing special work of this material. As successfully used for special work in Milwaukee, the following is the specified analysis (*El. Ry. Jour.*, 1923):

Carbon	0.45 to 0.55	Sulphur	0.05 max.
Chromium	0.60 to 0.80	Phosphorus	0.05 max.
Nickel	2.50 to 3.00	Manganese	0.60 to 0.80
Silicon	0.25 to 0.35		

High Silicon Steel. It may be well here to mention the possibilities in using an increased amount of silicon in common open-hearth steel for rails. Steel with as high as four-tenths of 1 per cent of silicon, or double the high limit used in this country, is reported to have given excellent wearing results on English roads.

Rail Joints. The common angle bar splice, Fig. 20, comes in contact with the rail along the fishing surfaces under the rail head

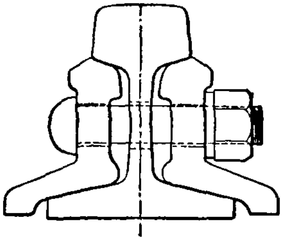


FIG. 20.—Angle bar splice.

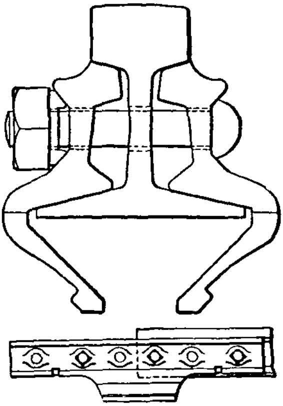


FIG. 21.—100 per cent type joint.

and on the rail base. By tightening the track bolts the bars are wedged in so that shear and bending moment due to wheel loads will be transmitted across the joint giving somewhat the effect of a continuous rail. To increase the strength and stiffness, the lower flanges may be widened opposite the joint and extended down below the rail base as in the "100 per cent" type, Fig. 21, the Bonzano, Duquesne, etc. In other forms a plate is placed under the joint as an extension of the lower flange of one of the angle bars, or as a separate plate locked to the lower flanges of both bars. Figs. 22 to 26, inclusive, illustrate the special forms of joints known as the Webber, Continuous, Wolhaupter, and Atlas.

A number of rail joints were tested at the Watertown Arsenal under the direction of the Committee on Rail of the Am. Ry. Eng. Assn. and the results published in Bulletin 123, 1910. The span was 30 in. Two joints of a kind were tested, one with a center load of 32,000 lb. on the base, the other with an equal load on the head. The rails were then inverted and the load increased to failure or to the capacity of the testing machine. Below are given the

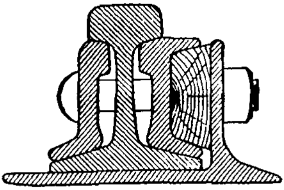


FIG. 22.—Webber rail joint.

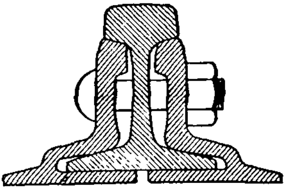


FIG. 23.—Continuous rail joint.

data for the strongest and stiffest joint tested for the 100-lb. rail and for the strongest and stiffest angle bar joint for the 100-lb. and for the 80-lb. rail.

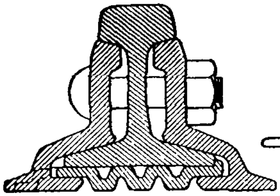


FIG. 24.—Wolhaupter rail joint.

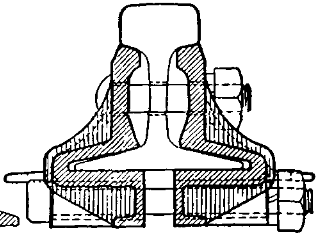


FIG. 25.—Atlas rail joint—section A–A.

	Rail			Joint		
	100-lb.		80-lb.			
	a	*b*	*c*	*a*	*b*	*c*
Area full section	9.92	10.02	7.84	13.6	6.74	6.74
Moment inertia	43 80	49.00	27.78	47.2	9.36	9.33
Section modulus, normal	14.85	15.69	10.58	15.0	4.39	4.36
Section modulus, inverted	15.64	17.00	11.11	12.24	5.14	5.12
Elastic limit, normal	119,000	125,000	85,000	72,500	56,000	55,500
Elastic limit, inverted	125,000	136,000	89,000	41,000	......	......
Max. deflection, normal	0.013	0.012	0.020	0.048	0.065	0.093
Max. deflection, inverted	0.013	0.012	0.020	0.050	0.052	0.082
Ultimate, normal	......	......	......	117,500	95,500	88,000
Ultimate, inverted	......	......	......	87,500	......	......

The 100-lb. rail, *a*, was an American Society section, the 100-lb., *b*, and the 80-lb., *c*, Dudley New York Central sections. The section modulus normal is for the head and the inverted for the base of the rail. The elastic limit and the maximum deflection are inserted from computation for comparison with the joints, with an assumed elastic limit of 60,000 lb. and a modulus of elasticity of 30,000,000 lb. It may be noted that the joints compare more favorably with the rail in section modulus than in stiffness.

The bolt holes for the splice bars and the notches for the spikes to prevent rail creeping are usually punched, and this is one of the reasons why soft steel, about 0.1 per cent carbon, is common, but one of the pairs of bars tested contained 0.63 per cent carbon. The length varies from about 2 ft. with 4 bolts to 3 ft. with 6 bolts, although a length of 3½ ft. has been used. The short bar

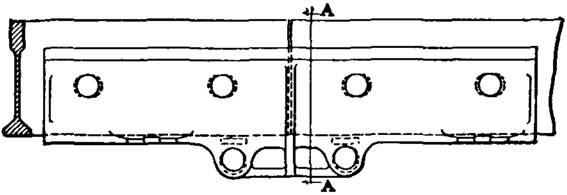


FIG. 26.—Atlas rail joint—suspended type.

is used with a suspended joint midway between two ties, the bar reaching from tie to tie. The long bar is used with a three-tie joint, the ends resting on the outer ties with the joint on the center one. The suspended joint is advocated as doing away with pounding action due to a solid support under the rail and it is required for the splice bars which extend below the rail base at the joint; the three-tie joint is advocated as giving better support for the joint (which is the first part of the rail to go down under traffic) than the two-tie support and it is used on a number of the heavy traffic trunk lines. With long bars a little wear of the fishing

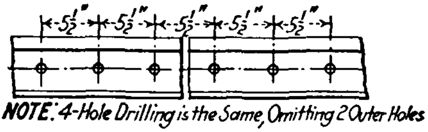


FIG. 27.—Am. El. Ry. Assn. 6-hole drilling for standard section rails.

surfaces or looseness of the bolts has less effect in allowing angular motion, a consideration often overlooked. The track bolts are usually ¾ to ⅞ in. with round heads and elliptical section under the heads to prevent turning in the elongated holes of the splice bars. Various methods are used to prevent the nuts from rattling loose, among the most effective being the spring washer and the Harvey grip thread. The holes in the rails are made large enough to allow for temperature changes, the bolts acting only in tension to hold the fishing surfaces in contact with sufficient force to transmit bending moment.

Rail joints may be laid opposite or alternate. The former is common in the West and in Europe. It is advocated on the ground that since the tendency of the track is toward low joints, if they are put opposite no side lurch is given and the train is easier on track and passengers. The motion, however, is unpleasant and it is hard on draft rigging and on track, the blow if both sides go down being heavier than for only one. On curves alternate joints hold alinement much better as there is a solid rail opposite each joint to prevent the track from kinking due to springing the track somewhat to fit the curvature. Again, it is more expensive on curves due to having to cut and redrill each inner rail, rather than let the inner joints run ahead until a rail 1 ft. or more shorter than the standard length can be used.

If a concealed type of rail bond is to be used, care must be taken in the design of the rail joint that sufficient space is allowed for the bond, even after wear has taken place on the joint.

The Am. El. Ry. Eng. Assn. recommends joints for the 7 and 9-in. girder rails as shown by Figs. 15 and 16. The Committee on Way Matters, 1910, states that failures due to poor design of plate may be attributed largely to insufficient fishing surface or insufficiency of metal in the web, either of which faults may cause a line contact, or nearly so, at the fishing surfaces, thereby materially decreasing the life of the joint. The life of this style joint is materially increased, at small additional cost to the completed track structure, when a plate is designed of such strength and stiffness that it cannot be bent under any load applied when tightening the bolts, particularly when supplied with channel contact surfaces of such width as the head of the rail will permit.

Welded Joints. By the use of welded joints a continuous rail, which fulfills the three required elements of a perfect joint, is obtained, and if a permanent weld could be made, thus eliminating the joint, the life of the rail at the joint would be equal to that of the rail at any other point. The early attempts to weld rails were made by means of electricity. Later the cast weld was introduced and became very popular, and after another period an improved electric weld was placed on the market. A late development in the cast weld line is the thermit weld, which is essentially a modification of the general principles of cast welding. This has been followed, still more recently, by the arc welded joint.

Cast Weld. This is an early type of weld, sometimes called the Falk joint, produced by pouring molten cast iron into a mold around the ends of the abutting rails, the latter having been cleaned by a sand blast or some other means. In many cases it is not a weld at all owing to the difficulty in preheating the cast iron sufficiently to melt the steel of the rails. It was a good mechanical joint, however, and frequently served admirably the functions of both bond and joint plates. The many failures that occurred in this and other types of welds have been attributed to numerous causes. Whether the heating of the rail was sufficient to change its properties, and thereby result in excessive wear and breaks, has occasioned a great deal of discussion and still is a mooted question. Expansion and contraction has undoubtedly been responsible for failures in some cases, particularly where rails were welded in hot weather or laid in pavements having poor binding qualities. While the cast weld joint has been superseded in most cities by various forms of improved electric or thermit welded joints, it is interesting to note that in a few cities the results obtained were such as to justify their continuous use down to this date, among these cities being Minneapolis, Detroit, Brooklyn and Milwaukee. In Milwaukee there are many cast weld joints which are 20 years old, and which are in such good condition that their location in the track can hardly be distinguished on inspection. A new type of cast weld joint has been developed in Milwaukee which promises to give much better results than the old rectangular form of weld formerly used and in which many failures developed. In the new joint by keeping the weld well below the head of the rail and taking precautions to keep the head of the rail cool by means of a water-filled strong back while pouring the weld, the running portion of the rail retains its original properties and does not have so great a tendency to cup under continuous traffic. The cast weld in this altered form is considered as the standard in Milwaukee, although a considerable percentage of repair joints are now being made by other methods.

Thermit Weld. This is essentially a cast weld, but as the temperature attained by the reaction of aluminum and iron oxide is far in excess of that produced in the ordinary cast weld, the ends of the steel rails are melted down and an obliteration of the joint actually results. In making the improved form, known as the thermit full section weld, the rails are spaced three-fourths of an inch apart, in the case of new rail, and a gap of that width is provided by cutting or sawing in the case of old rail. Into this space and between the heads of the rails only is placed an insert cut from a rolled section of similar analysis to the rail itself. (See Fig. 28.) The web and base are thoroughly melted and amalgamated with the thermit steel, but the entire head of the rail is not, only the outside of the head and the lip of the rail being melted by the thermit steel. A weld of the entire rail section is obtained, however, when the thermit steel begins to cool and contract. The proximity of this steel to the insert has melted up the lower part of the insert and heated the upper part as well as the abutting heads of the rails to such a high temperature that when contraction sets in, thus drawing the rails together with tremendous force, the effect is to butt weld the insert permanently into place between the rail heads. The advantage claimed for this method is to be found in the fact that the metal in the head of the rail coming into contact with the car wheels is not melted and its physical properties not changed. As a result, it is said, welds made in this way have shown no tendency to cup after six years of hard service under heavy and frequent traffic, and the breakage of welds made in this way has been very small. Where breaks have occurred they have usually taken place either shortly

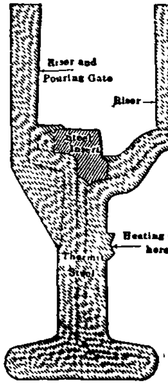


Fig. 28.—Thermit full section weld.

after welding, indicating something wrong with the procedure, or else they have occurred during the first winter.

Clark Joint. Thermit welds have also been used in connection with mechanical joints. A shoe of thermit steel is poured around the base of a bolted or riveted rail joint, thereby welding the base of the rail, but in no wise changing the properties of the head or running part. The joint needs no other bond, and is mechanically good. It has met with marked success in Baltimore and Cleveland, where it is known as the "Clark joint."

Electric Weld. The electrically welded joint was introduced over 20 years ago and has found wider application than any of the other modern types. It consists of heavy bars or plates from two to three feet in length spot welded to the web of the rail by the use of electric current. The process requires a heavy and expensive plant and is usually carried out by contract on a comparatively large scale. For this reason it is not well suited to installations on small systems. It is well adapted to the reclaiming of old track as well as for new work, and has been applied on open T-rail construction where expansion joints are installed at intervals to provide for expansion and contraction. The electrically welded joint with spelter head support was superseded by a chock head support bar joint which was introduced in 1916. A drop-forged steel chock is welded over the bars at the center to provide a head support for the abutting rails. A weld is made between the back of the rail head and the chock as well as above the bars. Electrically welded chock joints in which the use of the bars was dispensed with were first used in Chicago in 1918. The newer butt-weld process consists essentially in heating the ends, or abutting surfaces, of the rails to be joined, by having them in contact with a liquid flux which is heated above the welding point of steel by the passage of a comparatively small electric current. When the rail ends have reached a welding heat, they are forced together by means of a powerful clamp, while a hammering of the heated portion of the rail across the head gives a forging finish to the operation. Failures of the electric weld have in general been confined to fracturing of the rail at the end of the welded bar, and are no doubt the result of strains introduced into the web of the rail from the localized heating. Such breaks have been more prevalent in old than in new rails. On new work with rails having no bolt holes the failures have been reduced to a minimum which is not serious.

Arc Weld Joints, in which the plates are welded to the rail by means of an electric arc, combine features of ease and simplicity of application, low cost, and high conductance. This method of welding has also been applied to the welding of rails to steel ties and also to the welding of bolted and riveted joint plates to the base of the rail. There are three principal forms of arc welded joints, two of which use the metallic electrode process, and the third, the carbon electrode process. One of the two using metallic electrode process welds the joint plate to the web of the rail. The other metallic electrode method and the carbon electrode process weld the joint plate top and bottom to the head and base of the rail respectively.

Nichols Composite Joint consists of two plates which fit the web of the rail and are riveted snugly to it, but which do not come in contact with the fishing surfaces. The spaces thus left around the head and foot of the rail are filled with molten zinc, which expands upon solidifying and enters into all of the irregularities of the rail surface. If the rail and plates are properly cleaned before pouring the zinc, a good electrical contact is obtained, and a joint of high conductance is the result. The process is rather expensive, and is warranted, therefore, only on lines bearing heavy traffic. The fact that it has been a standard of construction in the city of Philadelphia for a number of years is an indication of its practicability.

Compromise Joints, or the joint between ends of two dissimilar rail sections are often points of trouble, due to the fact that it is difficult to make such joint plates with good contact for both rails, except with special facilities. It is too often the case that the compromise joint is overlooked until actually needed and a rough

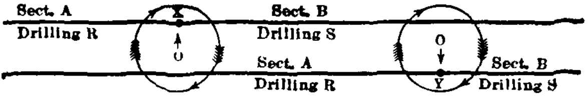


FIG. 29.—Am. El. Ry. Eng. Assn. uniform method for designating compromise joints.

blacksmith job is allowed to suffice only to cause trouble later. Some form of welded joint between the two rail ends is most satisfactory, and where joint welding apparatus is not available in the field, some companies weld together short pieces of the two rail sections, and use standard joint plates to connect these to the similar adjoining rails. A uniform method for designating compromise joints was approved by the Am. El. Ry. Eng. Assn. in 1912, and is shown by Fig. 29. The observer stands between rails at O, facing joints. All readings are made from left to right, as in reading a book, and as indicated by arrows on the figure. Joint *X* would read "one combination joint connecting Sec. A worn 1/4 in., drilled *R*., with Sec. B worn 1/8 in., drilled *S*."

Allowance for Expansion in Laying Rail. The Am. Ry. Eng. Assn. (1912) recommends that in laying new rail, standard expansion shims shall be used, that the temperature of the rail shall be taken by placing a thermometer on the rail, and that the openings between 33-ft. rails shall be as follows:

Temperature (degrees Fahr.)	Amount of opening
0– 25	1/4 in.
25– 50	3/16 in.
50– 75	1/8 in.
75–100	1/16 in.
Over 100	close

The above applies, of course, to open track. The engineers on street railway lines, through experience, have concluded that the spacing of rails with open joints is unnecessary in paved streets and ascertained by practice that if the rails were butted tightly together the damage to the rail end, due to the impact of the wheels, could be minimized. This damage is in direct ratio with the width of the opening between rail ends. Therefore, in order to obtain a perfect fit of the rail ends, most rail specifications now require that, in addition to the sawing, the rails be milled and finished, and, if necessary, filed so that a true and accurate rail end may be obtained. Further, the Am. El. Ry. Eng. Assn. has, in view of the desirability of obtaining a joint with no intervening space between the abutting rail heads, recommended, in its specifications for the manufacture of rails, in addition to the usual finishing of the rails at the ends, that the rails may be undercut $1/32$ to $3/32$ of an inch. There is now in use in one of the western cities, on T-rail construction, a special form of rail end, the rail being cut to a vertical bevel in the ratio of 1 to 6. This type of cutting enables the engineers to obtain a perfect fit of the abutting rail heads and with the ordinary bolted joint has been found satisfactory. It is questionable, however, if joints of this type would be of any particular benefit if laid on heavy traffic lines.

Grinding and Compromising Joints. Many engineers believe that all joints should be ground when first installed. There is no question that a joint should not be left if there is any variation whatsoever between the running surfaces of abutting rail ends. Once the wheels have an opportunity of pounding the rail the receiving side will rapidly cup out and the track will fail. In a double track, and where the traffic is in one direction, the receiving rail, after a period of years, becomes badly cupped. If the joints are ordinary plates they may be offset or replaced with step joints so that the bottom of the cup is on a level with the top of the delivery rail. The plates are then driven tight and bolted up in place and the adjoining rails ground level. It is important that the joints be ground back in proportion to the depth of the cup, this grinding extending sometimes for a distance of 6 ft. on each side. Rail treated in this manner and having proper foundation and ties will ride like new track. Rail could have its life considerably extended if, as soon as the joints begin to pound, the plates are pulled up and the rail so ground that the butting rail ends are a true, level surface. Such practice should be resorted to instead of neglecting the joints to such an extent that nothing is to be done except to lose the rail or apply some expensive method in reclaiming it, such as inserting short pieces of new rail.

Rail Corrugation. With the increased use of solid forms of track construction (concrete base, etc.), the advent of large cars, and the necessity of rapid rates of acceleration and braking, this form of maintenance trouble increased greatly. While the rail manufacturer is ready to lay the cause to modern traffic conditions, equipment and air brakes, many engineers place it with the rolling of the rail, and a satisfactory reason has not yet been offered. The curved head rail section and tilted rail both tend to retard the appearance of corrugation in new track. The corrugations must be removed by some form of grinding, either by an ordinary file or emery block set in a frame and operated by hand, or by one of the many forms of rail grinding machines. The grinding should be done as promptly as possible after the corrugations manifest themselves, as once started the trouble rapidly grows worse

Miscellaneous Track Fastening Material. The Am. El Ry. Eng Assn. (1922) adopted as recommended specifications the following, which are identical with Am. Soc. for Testing Mat. specifications as noted:

Low carbon steel track bolts	A.S.T.M. A76–20
Quenched carbon steel track bolts	A.S.T M. A50–21
Quenched alloy steel track bolts	A S.T.M. A51–21
Steel track spikes	A S.T M. A65–18
Steel screw spikes	A.S.T.M. A60–21
Steel tie plates	A.S.T.M. A67–20

Perhaps the most important improvement in the bolted rail joint in recent years and the one which will do more than any one other thing to lengthen the life of the joint is the substitution of improved bolts for the more inferior grades. The testimony in favor of bolts having a high elastic limit, and a great ultimate strength as against iron or even carbon-steel bolts is apparently conclusive, and the slight additional expense is completely justified by the decreased maintenance and additional life of the bonds and joints, to say nothing of the improved riding properties of the roadbed.

Track bolts	Square nuts Bolts per 200 lb. keg							Hexagon nuts Bolts per 200 lb. keg						
Diam., length, in.	½ in	9/16 in	⅝ in	¾ in.	⅞ in	1 in	1⅛ in	½ in	9/16 in	⅝ in	¾ in	⅞ in.	1 in.	1⅛ in.
2	770	568	441					790	611	458				
2¼	730	540	420					750	580	436				
2½	694	515	402	290				712	551	417	302			
2¾	662	493	384	277				678	525	398	289			
3	633	472	369	265	194			647	502	382	276	201		
3¼	606	453	355	254	186			619	480	366	264	193		
3½	581	435	341	244	179			593	460	352	254	185		
3¾	558	418	329	235	173	130	95	570	442	339	243	178	136	99
4				227	166	125	92				234	171	132	96
4¼				219	160	121	90				226	165	127	93
4½				211	155	117	87				218	160	123	90
4¾						114	84						119	87
5						110	82						115	84
Weight of 1000 nuts, pounds														
	112	146	218	245	374	525	747	93	122	182	216	316	462	685

Tie rods are usually of (about 2″ × ⅜″) flat steel or iron, with (about ⅞″) round threaded ends, and nuts on the inside and outside of the web of the rail. Some companies have used rods of a circular section throughout, the advantage being that they can be readily made in any machine shop. However, while the latter may answer admirably in open track or where the paving is macadam, the principal objection is that their diameter, which cannot well be made less than ⅞ in., is such that entirely too wide a joint is produced in block paving. The round rod also offers practically no resistance to turning as does the flat rod, and this element might tend to cause the nuts to loosen. Another advantage of the flat rod is that it lends itself more easily to installation on account of its greater flexibility. The function of the tie rod is to prevent spreading of the rails and, since this action is more severe on curves, it is very desirable to install both tie rods and either rail braces or brace tie plates on them. It has been found that a spacing of tie rods on 6-ft. centers has given very satisfactory results in most cases, although some engineers recommend a spacing as low as 5 ft. Tie rods should be located as near the head of the rail as

Track spikes, size, in.	No. in keg of 200 lb.	No. of kegs per mile, ties 2 ft. centers
5½ × 9/16	375	29⅓
5 × 9/16	400	26
5 × ½	450	23⅓
4½ × ½	530	20
4 × ½	600	17⅔
4½ × 7/16	680	15½
4 × 7/16	720	14¾
3½ × 7/16	900	11
4 × ⅜	1000	10½
3½ × ⅜	1190	9
3 × ⅜	1240	8½
2½ × ⅜	1342	7⅞

possible in order to be of the greatest value. The vertical play of the rails develops bending stresses in the tie rod in addition to the direct tensile stresses caused by the spreading action. The greatest bending occurs just inside the inside nut, and most of the failures of tie rods occur at this point or somewhere between it and the point where the section changes from flat to round, although some breaks have been known to occur between the two nuts. It is believed, however, that in the latter instances failure has been caused by loosening of the nuts sufficiently to permit play between them. It is because of these bending stresses that it is essential that a certain ductility in the material be provided for in the specifications.

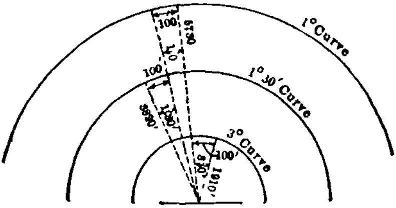


Fig. 30.—Designation of track curves.

Notes on Track Curves. In American practice a curve is designated by the number of degrees of angular measure subtended at the center of the circle by a chord of 100 ft. (Fig. 30). A 1-

deg. curve is a curve of such a radius that a chord of 100 ft. subtends a central angle of 1 deg., while an n-deg. curve has a radius such that a chord of 100 ft. subtends a central angle of n deg. The radius of a 1-deg. curve is 5730 ft. and the radius of an n-deg. curve is $\frac{5730}{n}$ ft. The maximum allowable curvature on new construction of trunk line steam railroads now seldom exceeds 3 deg. In street railway work, where curves of small radius are necessary, the curves are often designated by the radius instead of the degree of curve.

Super-elevation of Outer Rail on Curves. On curves the outer rail is elevated sufficiently to neutralize the centrifugal force of the train, which varies with the degree of curvature and the speed.

Table of Super-elevation for Curves

Degree of curve	Speed in miles per hour						
	10	20	30	40	50	60	80
	Super-elevation of outer rail in inches						
1	......	1/4	3/4	1 1/4	1 3/4	2 1/2	4 1/2
2	1/4	1/2	1 1/4	2 1/4	3 1/2	5	......
3	1/4	3/4	2	3 1/2	5 1/4	7 1/2	......
4	1/2	1	2 1/2	4 1/2	7	......	......
5	1/2	1 1/2	3	5 3/4	8 3/4	......	......
6	1/2	1 3/4	3 3/4	6 3/4	......	......	......
7	1/2	2	4 1/2	7 3/4	......	......	......
8	1/2	2 1/4	5	9	......	......	......
9	1/2	2 1/2	5 3/4	......	......	......	......
10	3/4	2 3/4	6 1/4	......	......	......	......
11	3/4	3	7	......	......	......	......
12	1	3 1/4	7 1/2	......	......	......	......
13	1	3 1/2	8	......	......	......	......
14	1	4	9	......	......	......	......
15	1 1/4	4 3/4	......	......	......	......	......

The above table (*Elec. Jour.*, 1908) gives the super-elevation for different speeds and curvature. It is not considered good practice to elevate any curve more than 7 1/2 in.

Fig. 31 shows a chart of the super-elevation of outside rail on curves as used by the Ohio Electric Railway Company. The chart bears the notation that the speed of cars and elevation of outside rails should be limited to figures of the chart shown inside the heavy lines. Thus on a 4-deg. curve the track elevation should not exceed 4 1/2 in., nor the speed of the car more than 40 miles per hour; on a 30-deg. curve the elevation should not exceed 5 in. nor the speed of the car more than 16 miles per hour.

Formula for Super-elevation. The amount of super-elevation of outside rail on curves is given by the following formula. This formula is an approximate one, but the error amounts to less than one per cent in the case of a super-elevation of 9 in. on standard gage and is proportionately less for other super-elevations.

$$e = 0.00001165\, GS^2n$$

where e = super-elevation of outer rail, inches
G = gage of track, inches
S = speed of train, miles per hour
n = degree of curve.

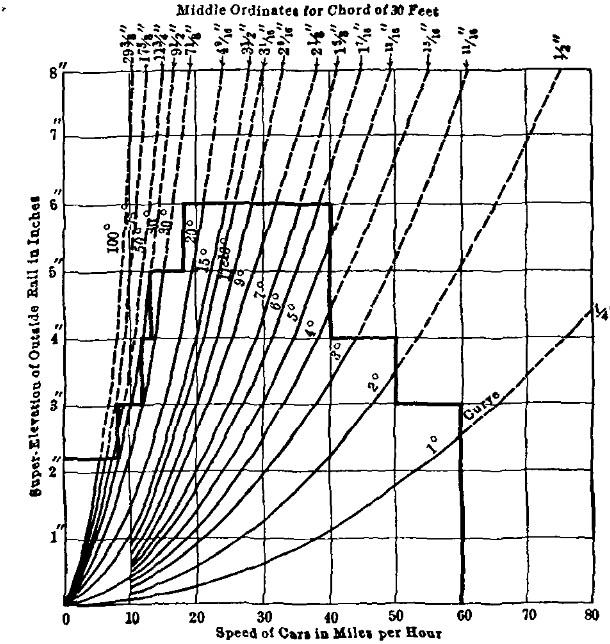


Fig. 31.—Ohio Electric Ry. chart for super-elevation of outside rail on curves.

Rail Bending. In shaping a rail for use in a given curve, it is convenient to stretch a cord from end to end of the concave side of the head and bend the rail until the length of the middle ordinate *(distance from center of cord to side of head) is equal to the value* given in the following table for the curvature and rail length in hand. The middle ordinate for any other length of rail or degree of curvature within the limits of the table may be found by proportion.

Middle Ordinates for Curving Rails

Degree of curve	Length of rail						
	33 ft.	30 ft.	28 ft.	26 ft.	24 ft.	22 ft.	20 ft.
	in.	in.	in.	in.	in.	in.	in.
1	5/16	1/4	3/16	3/16	3/16	1/8	1/8
2	1/2	1/2	7/16	3/8	5/16	1/4	3/16
3	13/16	11/16	5/8	9/16	7/16	3/8	5/16
4	1 1/8	15/16	13/16	11/16	5/8	1/2	7/16
5	1 7/16	1 3/16	1 1/16	7/8	3/4	5/8	9/16
6	1 11/16	1 7/16	1 1/4	1 1/16	7/8	3/4	5/8
7	1 15/16	1 5/8	1 7/16	1 1/4	1 1/16	7/8	3/4
8	2 1/4	1 7/8	1 5/8	1 7/16	1 3/16	1	7/8
9	2 1/2	2 1/8	1 7/8	1 5/8	1 3/8	1 1/8	15/16
10	2 13/16	2 3/8	2 1/16	1 3/4	1 1/2	1 1/4	1 1/16
11	3 1/8	2 5/8	2 1/4	1 15/16	1 11/16	1 3/8	1 1/8
12	3 3/8	2 13/16	2 1/2	2 1/8	1 13/16	1 9/16	1 1/4
13	3 5/8	3 1/16	2 11/16	2 5/16	1 15/16	1 5/8	1 3/8
14	3 15/16	3 5/16	2 7/8	2 1/2	2 1/8	1 3/4	1 1/2
15	4 1/4	3 9/16	3 1/16	2 11/16	2 1/4	1 15/16	1 9/16
16	4 1/2	3 3/4	3 1/4	2 13/16	2 3/8	2 1/16	1 11/16
17	4 13/16	4	3 1/2	3	2 9/16	2 3/16	1 3/4
18	5 1/8	4 3/16	3 11/16	3 3/16	2 11/16	2 5/16	1 7/8
19	5 3/8	4 7/16	3 7/8	3 3/8	2 7/8	2 7/16	2
20	5 5/8	4 11/16	4 1/8	3 9/16	3	2 9/16	2 1/8

Finding Degree of Curve. It is sometimes necessary to find the degree of a curve on a completed track; this can be done by the use of a tape, by the following method: Some convenient point on the outer rail as *C* (Fig. 32) is selected. From the gage side of the outer rail at *C*, sight along the gage side of the inner rail on the line

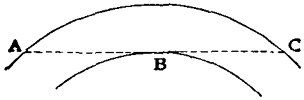


Fig. 32.—Finding degree of curve.

CB. The point *A*, where the line *CB* produced cuts the gage side of the outer rail again, is thus determined, and the distance *AC* is measured. The length *AC* is now compared with the values given in the table and the corresponding degrees of curve in the table will be the value sought. The values given in the table for *AC* are the long chords corresponding to a constant middle ordinate (equal to the gage of the track which, in this case, is 4 ft. 8.5 in.). The distance measured may vary several feet from that found in the table due to the inaccuracy of the alinement of the track, but since curves are nearly always made in even degrees, except in special cases where local conditions make it necessary, the degree of curve can almost always be determined by the above method in a very short time.

Table of Chord Lengths for Outer Rail Corresponding to Middle Ordinate of 4 ft. 8½ in.

Deg. of curve	1°	2°	3°	4°	5°	6°	7°	8°	9°	10°	11°	12°	13°	14°	15°
Length of AC in ft.	463	328	268	232	208	190	176	164	155	147	140	134	129	124	120

Degree of Curve to Connect Two Tangents. (See Fig. 33.) The radius of a circular curve required to connect two tangents may be found by dividing the *apex distance* ($P.C.t$ in Fig. 33) by the tangent of one-half the total deflection angle. Dividing 5730 by this radius gives the degree of the curve. These rules may be expressed as follows:

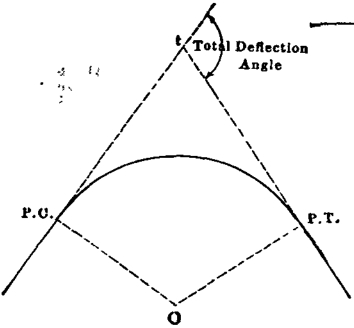


Fig. 33.—Curve to connect tangents.

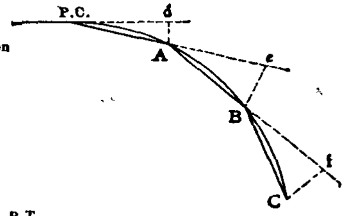


Fig. 34.—Laying out curve.

$$\text{radius of curve} = \frac{\text{apex distance}}{\text{tan. } \tfrac{1}{2} \text{ total deflection angle}}$$

$$\text{degree of curve} = \frac{5730 \times \text{tan. } \tfrac{1}{2} \text{ total deflection angle}}{\text{apex distance}}$$

Laying Out a Circular Curve by Chord Deflection Distance. (See Fig. 34, also Chord Deflection Distance, below.) Assuming a chord length (100 ft. is generally used), lay off $P.C.d$ on the tangent produced through $P.C.$, such that

$$P.C.d = \sqrt{(\text{chord length})^2 - (\tfrac{1}{2} \text{ chord deflection distance})^2}.$$

Locate A at the extremity of the first chord so that dA perpendicular to $P.C.d$ is equal to one-half the chord deflection. Produce $P.C.A$ to e so that Ae is equal to the chord length. Locate B so that AB is equal to the chord length and eB is equal to the chord deflection. Produce AB to f so that Bf is equal to the chord length. Locate C so that BC is equal to the chord length and fC is equal to the chord deflection.

Note: Results sufficiently accurate for many purposes may be obtained by taking the distance $P.C.d$ equal to the chord length instead of taking it equal to

$$\sqrt{(\text{chord length})^2 - (\tfrac{1}{2} \text{ chord deflection distance})^2}$$

Chord Deflection Distance.

$$\text{(chord deflection distance)} = \frac{(\text{length of chord})^2}{(\text{radius of curve})}$$

$$= \frac{(\text{degree of curve}) \times (\text{length of chord})^2}{5730}$$

TABLE OF CHORD DEFLECTION DISTANCES FOR 100-FT. CHORD

Degree of curve	Radius, feet	Chord deflection, feet	Degree of curve	Radius, feet	Chord deflection, feet
1	5730.0	1.745	21	274.4	36.44
2	2864.9	3 490	22	262.0	38.17
3	1910.1	5.235	23	250.8	39.87
4	1432.7	6.980	24	240.5	41.58
5	1146.3	8.724	25	231.0	43.28
6	955.4	10.47	26	222.3	44.98
7	819.0	12.21	27	214.2	46.68
8	716.8	13.95	28	206.7	48.38
9	637.3	15.68	29	199.7	50.07
10	573.7	17.43	30	193.2	51.76
11	521.7	19.17	31	187.1	53 45
12	478.3	20.91	32	181.4	55.13
13	441.7	22.64	33	176.0	56.82
14	410.3	24.37	34	171.0	58.47
15	383.1	26.11	35	166.3	60.14
16	359.3	27.83	36	161.8	61.80
17	338.3	29.56	37	157.6	63.46
18	319.6	31.29	38	153.6	65.11
19	302.9	33.01	39	149.8	66.76
20	287.9	34.73	40	146.2	68.40

The above table gives chord deflection distances for 100-ft. chord and even degrees of curve. Values of chord deflection for other curvatures within the limits of the table may be obtained from the table by proportion.

Track Spirals. Where the track passes from tangent to curve, the direction and super-elevation change suddenly. To avoid the *shock and lurch of the train, due to an instant change in the re-*lative position of cars, trucks, etc., an easement is made from tangent to curve. This easement allows the train to take the path of a spiral, *i.e.*, the degree of curve increases at regular intervals from tangent to curve until the maximum degree of curvature, or of the simple curve, is reached. The Am. El. Ry. Eng. Assn. uniform *system of track spirals, adopted in 1923,* is shown in Manual Section W13-23. The spirals of this series are of the Searles type, all having the same angular functions and differing only in linear dimensions. The switch easements are derived from the plain spirals in such a manner as to permit the conversion of a plain curve into a branch-off with the minimum disturbance to existing work. At *least one switch easement is provided for each spiral which cuts in* without changing the alinement. A.E.R.E.A. recommended dimensions for switches and mates are used throughout. The number which designates each spiral indicates the length of each arc of that spiral on the center line of standard gage track. As special trackwork is usually dimensioned on the inner rail of curves, and staked out on the ground either on this line or on an offset line therefrom, the functions are tabulated for the inner rail of standard gage track. For other gages, the same dimensions are to be used for the inner rail, thus agreeing with the A.E.R.E.A. definition of switch radii. It is believed that a sufficient number of spirals and switch easements are included to cover all possible requirements without further interpolation. If spirals are desired for curves of still longer radii, those given in "Field Engineering" by Searles & Ives, are recommended. The Searles type of spiral is a *multi-compound curve in which the radius of curve progresses in* inverse arithmetical ratio from arc to arc. It is constructed upon a series of equal arcs and the angle subtended by the first arc will be the common difference for the angles subtended by the succeeding arcs. This form of curve has been proved by many years of experience to be the most suitable for the purpose of eliminating the shock of transition from tangential to circular motion of a railway car. In the A.E.R.E.A. series the base spiral is one having successive angles of 1 deg., 2 deg., 3 deg., 4 deg., 5 deg., 6 deg., 7 deg., 8 deg. and equal arcs of 10 ft. From this base a series is calculated by proportion for the lengths of arcs given in the designations of the individual spirals. The dimensions of these spirals are calculated and tabulated for the inner rail of standard (4 ft. 8½ in.) gage track shown. This alinement of the inner rail is to be followed for all gages to simplify special trackwork manufacture and to agree with the A.E.R.E.A. definition of switch radii.

L. R. Brown, Engineer N. Y. State Rys., shows simple and quite usable solutions of some compound curve problems in the *Elec. Ry. Jour.*, Vol. 55, p. 160, and Vol. 60, p. 704.

Laying Out Easement for Moderate Speeds. The following method of laying out an easement curve is from Trautwine's "Field Practice of Laying Out Circular Curves for Railroads."

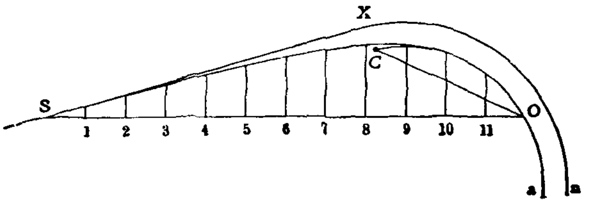


FIG. 35.—Laying out easement.

The process of laying out the curve is simple and the easement is sufficiently long for moderate speeds and curvatures. Let *xn*, Fig. 35, be part of the curve; and *xs* its tangent. Divide the chord deflection distance (see Chord Deflection Distance) by 10. The quotient will be *cx*. Set every stake of the entire curve inward this distance, *cx*, so that the curve shall be removed to *ca*. The radius of the curve is thus shortened by the length *cx*; but this is negligible.

From x measure on the tangent 100 ft. to s; and from c measure 50 ft. to the curve at o. After stretching a cord from s to o, lay off the eleven equidistant ordinates, if for rail laying; or only the middle one (6), or it and the two quarter-way ones (3 and 9), if for grading. (Since xs is always 100 ft., and co 50 ft., the distance apart of these eleven ordinates will always be nearly 12.5 ft.) The ordinates themselves in feet are found for any curve by multiplying cx by the following multipliers:

Ord.	Mult.	Ord.	Mult.	Ord.	Mult.	Ord.	Mult.
1	0.180	4	0.645	7	0.975	10	0.715
2	0.355	5	0.775	8	0.990	11	0.430
3	0.505	6	0 890	9	0 905		

Gage and Flangeway on Curves. The following method of determining the minimum flangeway required is from a report of the Committee on Way Matters of the Am. El. Ry. Eng. Assn., 1909.

Having a fixed rail groove, it is desired to know if that groove will properly serve under given conditions, and if not, how to alter it to meet them. Without discussing the question of wear on wheels or rails, let us find the minimum groove in which a given wheel flange will run, on the assumption that the equipment is geometrically perfect. In a car traversing a curve, the longitudinal axis of the car truck is normal to the radius of curvature of the track drawn to the mid-point of the truck, the wheels being held rigidly parallel to the axis. The portions of the wheel flanges which determine the size and shape of the minimum groove are the portions below the shaded sections shown in Fig. 36. The shaded areas are sections of the wheels on a level with the wheel-tread. If we project these portions of the flanges upon a radial plane, we obtain the contours of the minimum grooves, as shown at AB. To obtain these projections (see ABC, Fig. 37), pass a series of planes perpendicular to the axis of the truck. They will cut lines from the surface of the flange. Project these lines by means of track arcs upon the radial plane AB (shown rotated about line AB, as an axis, into the plane of the paper). The outline curve FCH, tangent to all of these projections, is the contour of the required minimum groove. The gage-line of a grooved rail or of a groove is customarily taken as the intersection of a horizontal plane through the tread-line with the head side of the groove produced, the rounded corner being ignored. On this basis, the position of such a gage-line for the minimum groove as compared with a gage-line as determined by the standard gage is determined as follows: If in Fig. 36, with the mid-point O of the axle as center, we describe

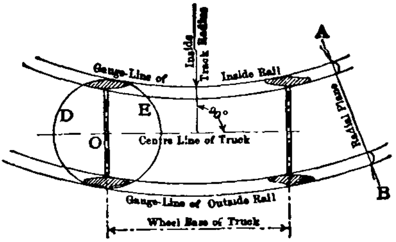


FIG. 36.—Flangeway on curves.

the circle D–E, of diameter equal to standard gage and draw track arcs tangent to D–E, one on each side and produce them to A–B (Fig. 37), we determine the position of the standard gage-line. Only one such determination—at m–n—is shown in Fig. 37. For wheels set to a gage ¼ in. less than standard gage and with the wheel gage-line taken at a point on the fillet of the flange ¼

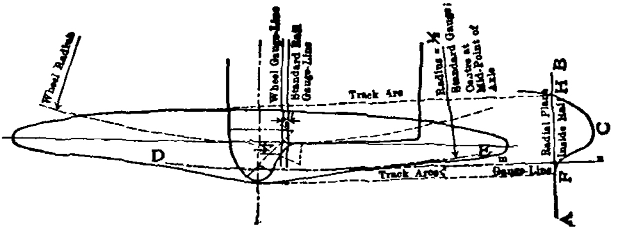


FIG. 37.—Flangeway on curves.

in. below the level of the tread, it is found that the standard gage-line and the minimum groove gage-line coincide for all flanges used at the present time.

The Am. El. Ry. Eng. Assn. recommends the following rules for determining gage and flangeway on curves: Condition: Wheel gage is assumed to be A.E.R.E.A. standard, namely, 4 ft 8¼ in., taken

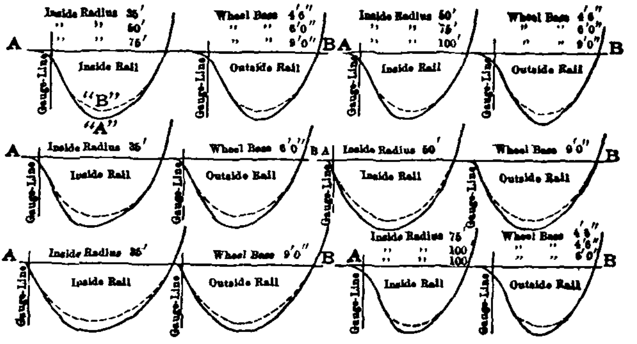


FIG. 38.—Miminum grooves for Am. El. Ry. Eng. Assn. flanges A and B

between fillets of flanges, ¼ in. below treads; track gage, 4 ft. 8½ in.

Rule 1. For A.E.R.E.A. standard wheel flanges and one wheel base use standard track gage and determine the minimum flangeways required by graphical plotting of groove, cross-section of wheel flanges, wheel base and radius of curve, as described above, allowing ⅛ in. extra width in outer flangeway for irregularities in wheel setting and special work.

Rule 2. If various types of wheel flanges or various lengths of wheel bases are operated, determine, by graphical plotting, the inner flangeways required for both the maximum and minimum conditions. Use inner flangeway to suit maximum conditions and widen gage by the difference between the maximum and minimum width of flangeways as above determined. Make outer flangeway $\frac{1}{8}$ in. wider than minimum outer flangeway for maximum conditions plus the amount of the widening of the gage.

Rule 3. Rolled guard rails having too narrow flangeways should be planed on gage side of inner rail and on guard side of outer rail to obtain required width.

Rule 4. If inside flangeway be too wide, increase the gage the difference between the width of the flangeway and the narrowest groove required. Make the width of outside flangeway equal to that required by the maximum conditions of outside radius, wheel flanges and wheel bases, increased by the widening of gage and also by $\frac{1}{8}$ in. to be allowed for irregularities in wheel settings and special work.

Rule 5. In expressing specification for gages and flangeways measure gage on a plane $\frac{1}{4}$ in. below the head of rail at fillet, except that when steam railroad wheels are to be used give distance on plane $\frac{5}{8}$ in. below the head of rail, making special notation thereof.

Rule 6. Measure flangeways horizontally from gage-line opposite head of rail. Give vertical slope of guard if rails are to be modified.

Victor Angerer in a paper before the Keystone Railway Club (1913) said that theoretically for track laid to true gage every combination of radius of curve and wheel base of truck, with a given wheel flange, calls for a specific width of groove to make the inside of the flange of the inside wheel bear against the guard and keep the flange of the outside wheel from grinding against the gage-line and possibly mounting it. It is manifestly impracticable to provide guard rails with such a variety of grooves or to change the grooves of the rolled rail. The usual minimum of $1\frac{9}{16}$ in. is wide enough to pass the A.E.R.E.A. standard flanges on a 6-ft. wheel base down to about 45-ft. radius, and the maximum width of $1\frac{11}{16}$ in. down to about 35-ft. radius. On curves of larger radius the excess width should be compensated for by a corresponding widening of the gage. If the groove in the rolled rail is too narrow for given conditions, it must be widened by planing on the head side of the inside rail, to preserve the full thickness of the guard, and on the guard side of the outside rail to preserve the full head. Unusual wheel bases such as 8 ft. or 9 ft. may require widening of the gage on some curves. This widening of gage is necessary only to bring the guard into play when the groove is too wide for some one combination of wheel and flange. In T-rail curves the guard is formed of a rolled shaped guard, or a flat steel bar, bolted to the rail. In special work and curves in high T-rail track a girder guard rail is often used. This is desirable, as it gives the solid guard in one piece with the running rail. The idea that a separate guard can be renewed when it is worn out does not work out in practice, as it is usually the case that when the guard is worn the running rail is also worn to such an extent that it will soon have to come out also. The only drawback to the girder guard rail in high T-rail track is the compromise joint between the two. This, however, may be properly cared for, as described elsewhere.

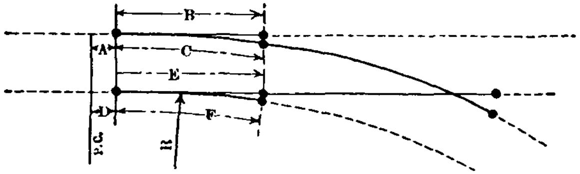
Hard Center and Solid Manganese Special Work. In such special work as frogs and crossings at the present prevailing prices, few lines can afford anything less durable for tracks in paved streets than "hard center" work. In this special work the pieces are formed of rails held together by a casting into which they are partly fused or welded, or they are made of an entire steel casting. In either case a recess is left in the central part into which a hard metal plate is set and fastened in various ways. The best metal for the centers is manganese steel, although some work is still made with chrome or tool steel centers. On account of the difficulty of machining manganese steel and because heating it to some marked degree destroys its toughness and wearing quality, certain limits are placed on the methods of applying and fastening manganese steel castings as centers in the main body of the special work. One method is to use a pendent lug and key going through the lug and body, pulling the center down into the recess. Another is to use a combination of wedges applied to the edge of the center and between it and the wall of the body portion. Another method is by use of a nut and brass set screw between horizontally extending lugs on the center and corresponding recesses in the body portion; and still another is the method of simply bolting down the center by vertical bolts running through both the body and center plate. In all except the last-mentioned method, zinc or spelter is poured in between the lower side of the center and the body. In the bolted-down center zinc is also used in the same way in some cases, while in others the under surface of the center and the top surface of the recess are finished by grinding and planing, and the two pieces are bolted together without the zinc centers. Much stress has been laid on the so-called renewability feature of special work, but any mechanically jointed center that is easily detachable would, when put under repeated strains, be likely to detach itself. On the other hand, this renewability feature has been applied very little in practice, as where centers are renewed on account of wear, it is very difficult to make a good job out of the surfaces, even by a very large amount of grinding of the adjoining parts on account of the necessarily unequal wear on the two sides of the rail. A good manganese steel center should outlast the adjoining portion of the body or rails, particularly the curved portions which exist in nearly every piece of special work. The wearing out of those portions of special pieces which had been made of ordinary rail outside of the manganese steel center led to the introduction of solid manganese steel special work in which the entire frog, crossing or other piece was made in one manganese steel casting with the arms cast and ground to correspond to the shape of the adjoining rail and joined to it with the regular joint plate. The main drawback is the greater cost, but in many cases the severity of traffic will warrant the greater initial outlay.

Special Work Specifications. The Am. El. Ry. Eng. Assn. has adopted recommended specifications for special track work, the various sections of the Manual being as follows:

W104-16 Material for use in manufacture of special work
W105-23 Solid manganese steel special work for girder rail
W106-23 Cast steel hard center special work
W107 23 Iron bound hard center special work
W108-23 Plain bolted special work
W109-21 Rail bound insert type special work

Flange-bearing Special Work. In either the hard center or solid type, the bottom of the grooves should be raised so that the wheel when crossing the intersecting groove will have a bearing on the flange on this raised floor, this flange bearing being a necessity with narrow tread wheels. If the wheel treads are wide enough to span the intersecting groove and still leave a good amount of bearing of the wheel tread on the running surface and wide enough so that when crossing a groove at right angles the blow may be distributed on a considerable area, this flange bearing is not necessary. Chicago specifications for hard center work provide that the total length of the level (flange-bearing) portion shall be twice the distance between the points formed by the intersecting guard and gage lines, and that inclines shall begin at the ends of this level portion and slope down uniformly to a depth of one inch at the ends of the manganese plate, the length of the slope to be not less than six inches. Other companies have adopted an approach slope as long as 18 inches to avoid the shock incident to abrupt rises in the flangeway. It is also common and good practice to raise the flange-way at the approach to the first intersecting flangeway and continue the shallow way clear through the crossing or series of crossings.

Switch Tongues and Mates. The important factor in the switch is the tongue bearing or pivot which should be constructed to hold the



Standard radius	Radius for all gages *R*	Dimensions			
		A	*B* and *C* mate	*D*	*E* and *F* tong. sw.
100 ft. lateral	97'7½"	0"	13'6"	0"	13'6"
200 ft. lateral	197'7½"	18"	16'6"	18"	16'6"
200 ft. equilateral	197'7½"	0"	13'6"	0"	13'6"
350 ft. equilateral	347'7½"	12"	16'6"	12"	16'6"
50 ft. lateral*	47'7½"	−6"	10'0"	−6"	10'0"
75 ft. lateral*	72'7½"	0"	12'0"	0"	12'0"
100 ft. equilateral*	97'7½"	−6"	10'0"	−6"	10'0"

* Should be used only where special conditions will not permit the use of standard dimensions shown in first four lines.

FIG. 39.—A.E.R.E.A. standard switches and mates.

heel of the tongue steady and to give the greatest possible resistance against the knocking down of the tongue at the heel end. It must also have means of taking up the wear so as not to allow the tongue to get too loose, and still it must work loosely enough to permit throwing devices, like electric track switches, to move the tongue under all conditions. Special attention must be given to the tongue switch to take up wear as it occurs. The tongue is now usually reinforced by lateral flanges against the thrust of the weight of cars. *Tongue locks, which are devices in which a spring holds the tongue* in one or the other position, should not be necessary in a properly constructed tongue switch if the bearings are kept as snug and tight as they should be, but are safeguards against a tongue which is too loose or which has tendencies of accidental throwing between trucks.

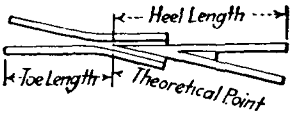
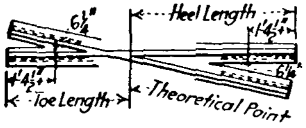


FIG. 40.—A.E.R.E.A. standard frogs for standard rail and M.C.B. wheels.

Frog No.	Angle	Toe length	Heel length	Total length
4	14°15′ 0″	3′2″	4′10″	8′0″
5	11°25′16″	3′4″	5′8″	9′0″
6	9°31′38″	3′6″	6′6″	10′0″
7	8°10′16″	4′5″	7′7″	12′0″
8	7° 9′10″	4′9″	8′3″	13′0″
10	5°43′29″	6′0″	10′6″	16′6″

Dimensions of Special Work. The Am. El. Ry. Eng. Assn. standard dimensions for switches and mates are shown in Fig. 39; for frogs for standard section rails and M.C.B. wheels, in Fig. 40; for turnout and crossover frogs for girder grooved, girder guard, plain girder and standard section rails, in Fig. 41. The Association has also adopted, in its Manual W10-21, standard designs and engineering data for tongue switch construction in connection with lateral or side turnouts, equilateral or diamond turnouts, and for crossovers. The Association has also recommended distances



Frog angle	Total length	Grooved rail		Plain rail	
		Toe length	Heel length	Toe length	Heel length
7°30′	12′0″	5′0″	7′0″	4′6″	7′6″
9°30′	10′0″	4′3″	5′9″	3′9″	6′3″
13° 0′	8′0″	3′6″	4′6″	3′0″	5′0″

FIG. 41.—A.E.R.E.A. standard frogs for turnouts and crossovers.

between tangent track centers through special work layouts which are believed to be sufficient in range to permit their use throughout the country without requiring radical changes in prevailing center distances. They have been prepared from the records of the special trackwork manufacturers and are intended prmarily for use in the tangent track portions of special trackwork. It is desirable, however, to adopt some one of these distances for tangent tracks generally. The distances have also been used in connection with the study of the matter of standardization of branchoff frogs, as these cannot be standardized unless uniform track centers are used. The recommended track-center distances are as follows:

Distance between track centers	Dummy gage or devil strip
9′ 2½″	4′ 6″
9′ 5½″	4′ 9″
9′ 8½″	5′ 0″
10′ 0″	5′ 3½″
10′ 6″	5′ 9½″
11′ 0″	6′ 3½″
12′ 0″	7′ 3½″
13′ 0″	8′ 3½″

Note that the above table indicates devil strip for standard gage; it is obvious that this will vary with other track gages.

Open Track Point Switch. As shown in Fig 42, the outer *rails of the track as seen in facing the turnout are continuous*, one along the main track and the other as the inner lead rail of the turnout; each is kinked slightly to protect the points of the switch rails which are placed between the continuous rails. The switch

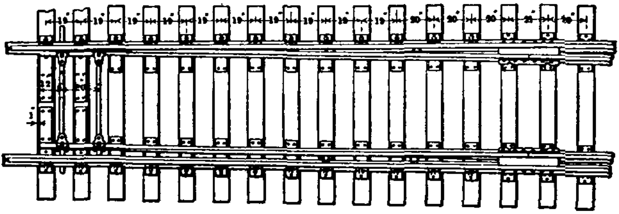


FIG 42 —Open track point switch.

rails are bent and planed so that the gage side of the head will be straight and the other side conform to a given spread at the heel and thickness at the point without cutting away the web. The Committee on Track of the Railway Engineering Association, 1912, recommends a spread of 6¼ in. between gage-lines, a throw of 5 in. at the first rod and a thickness of ¼ in. at the point which is afterward ground to ⅛ in. and the top corner rounded. A ⅜-in reinforcing bar is riveted to the web on each side and carried back as far as the heel connections will permit. The bottom of the switch rail is planed to fit on base of stock rail where bases overlap, Fig. 43*a*. Stop blocks are used as shown, Fig. 42, to support the free portion of the switch rail from the stock rail; which has a brace at each tie. The supporting plates for the two rai's are planed with a step to raise the top of the switch rail ¼ in. to provide for hollow tires. The top of the switch rail is planed down to be ⅛ in. lower than the stock rail and this planing runs out or rises ¾ in. in the following distances:

Length of switch rail	Length of planing
33 ft.	12 ft.
22	9
16½	7
11	5

The above lengths of switch rail are arranged for cutting from 33-ft. rails.

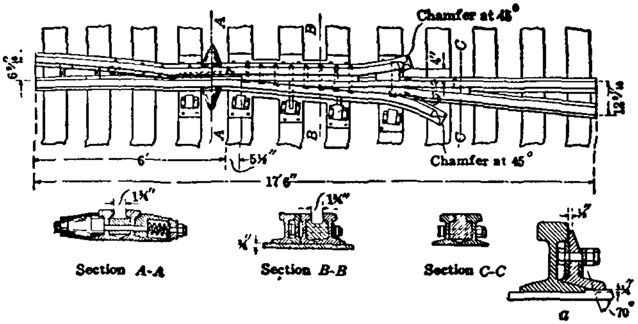


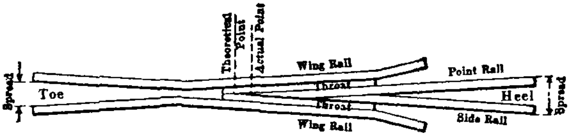
FIG. 43.—Open track spring frog (No. 11).

The committee claims that when the corresponding switch angle exceeds one-fourth the frog angle, the switch point presents the worst feature in the alinement and there is an economic loss both in space occupied and in cost of turnout. On this basis the committee recommends the fol'owing lengths of switch points:

16½ ft. for frogs over No. 6, and including No. 10
22 ft. for frogs over No. 10 and including No. 14
33 ft. for frogs over No. 14
11 ft. for frogs No. 6 and under when required.

Frogs Nos. 8, 11 and 16 are recommended as meeting all general requirements for yards, main track switches and junctions, with the object of eliminating other numbers and reducing the stock pile. The lengths shown in the drawings submitted are 13½, 17½ and 24 ft., respectively, for the three numbers. The rails are all bolted through the webs, using fillers or washers for spacers, while for the rigid frogs the rail bases are riveted to a plate.

The standard **rigid frog**, with the various dimensions, is shown by Fig 44.



No. of frog	Angle	Length	Actual Point to heel	Actual Point to toe	Theoretical Point to heel	Theoretical Point to toe	Spread Heel	Spread Toe
	deg. min.	ft. in	ft. in	ft. in.	ft. in.	ft. in	in.	in.
4	14 15	6 0	3 7	2 5	3 9	2 3	11¼	6¾
5	11 25	7 0	4 3½	2 8½	4 6	2 6	10 13/16	6
6	9 32	8 0	5 0	3 0	5 3	2 9	10½	5½
7	8 10	9 0	5 8½	3 3½	6 0	3 0	10 5/16	5⅛
8	7 10	10 0	6 2	3 10	6 6	3 6	9¾	5¼
9	6 21	11 0	6 10½	4 1½	7 3	3 9	9⅝	5
10	5 44	12 0	7 7	4 5	8 0	4 0	9⅝	4 13/16

FIG 44 —Dimensions of standard rigid frog for open track.

The **spring frog** is used for main line work where there is but little traffic on the side track on account of the better support of the wheel treads at the frog point. For the side track, the spring yields to the pressure of the wheel flanges and the frog acts otherwise like a rigid frog. With hollow treads the wings receive heavy blows from wheels coming in the direction to reach the heel or wings first unless the latter are chamfered to inclined planes as shown in Fig. 43 To protect the frog point and prevent derailments, a guard rail is necessary on each track with a flangeway of about 1¾ in to guide the wheels past the frog. Where the turnout is on the outside of a curve, a guard rail is often placed in advance of the switch point to prevent the flanges from crowding and possibly getting behind the point when set for main track. Cutting the side track point about 2 ft short allows the guard rail to reach more nearly opposite the main track point.

Derailing Switches. Derailing switches may be used as additional safety devices at points where it is imperative that cars or trains do not pass except under certain exact conditions. Under favorable conditions cars may be expected to start unaided on a grade of about 5/10 of 1 per cent, but wind will start cars down an easier grade, and heavy wind may start them on level track The setting of brakes should not be depended upon to hold cars that are left alone; consequently, wherever there is likelihood that cars on side track may start from gravity or be blown or easily pushed, the only safe policy is to provide for derailing them before they can get far enough to obstruct main track A derailing switch may consist of a single moving rail connected with a switch stand, although sometimes two moving rails will be used, as in the stub switch. The rails should be set to guide the wheels away from the main track. Switch points for derails may be made more blunt than those commonly used in turnouts, but for derails in side tracks old point rails too badly worn for main track service may be used. The throw of the switchpoint should be such that a derailed wheel may pass between the point and stock rails without spreading them apart, and at the heel of the point rail the nuts of the splice bolts should come on the gage side. In side tracks much used the derails should be connected with the main line switch or switch stand, so that the operation of closing the switch for main track opens the derail, and *vice versa*. This connection is usually made by means of throw rods and bell crank. There are also various special forms of derailing devices other than the common ones mentioned above. In order to hold the wheels to the ties a long guard rail should be laid on the track about 8 in. from the opposite rail, extending from a point in advance of that where the wheels are derailed.

Catch Sidings. For stopping runaway cars or trains on heavy grades without derailing, resort is sometimes had to catch sidings. Camp gives an example of such provision as found on the Canadian Pacific road between Hector and Field, B. C., near the summit in the Rocky Mountains, where there is a 9-mile grade of 4 4 per cent. Along this grade there are spur tracks or "blind sidings" 1 mile apart, each tended by a switchman. Each spur track runs up into

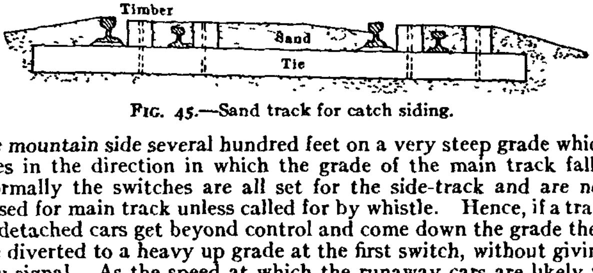


FIG. 45.—Sand track for catch siding.

the mountain side several hundred feet on a very steep grade which rises in the direction in which the grade of the main track falls. Normally the switches are all set for the side-track and are not closed for main track unless called for by whistle. Hence, if a train or detached cars get beyond control and come down the grade they are diverted to a heavy up grade at the first switch, without giving any signal. As the speed at which the runaway cars are likely to enter such a siding is high the curvature of the turnout should be easy and the angle of the switch points small. Wherever it is feasible to do so, it would be well to have the switches for such sidings turn from the outside of a curve in main track This arrangement would permit of easy curvature in the turnout, or perhaps enable the turnout to branch off at a tangent. In lieu of the up-grade arrangement the catch siding is sometimes buried in sand to the depth of a few inches over the rails. Sand tracks are more common in Europe than in this country. A cross-sectional view of a sanded catch siding in use at Dresden, Saxony, is shown in Fig 45. The rails of the diverting track are laid gauntlet fashion, on the same ties with the main rails, and the stretch of diverting track is provided at both ends with a switch for connecting with the main line. Guard timbers or angle irons for retaining the sand are placed at both sides of each rail of the siding, which gradually dips deeper until it is covered by 2 or 3 in. of sand. The arrangement is considered very efficient for the purpose. In this particular instance the catch siding is 1640 ft. long and 1148 ft. of it is covered with sand. In very dry weather the sand is kept damp. The braking effect of sand sidings is discussed in *Engineering* (London, England, 1897), and in the Bulletin of the International Railway Congress (1899). The *Street Railway Journal* (1906) gives the results of a series of tests on a catch siding as installed on the New Jersey & Hudson River Ry. & Ferry Company's line, where the catch siding is located on a 7 per cent grade and the approach of the sand track for a distance of 1500 ft. is also a 7 per cent grade. The catch siding is 180 ft. long. The car used in making the test was one of the company's standard closed cars, weighing 22 tons and equipped with M.C.B trucks and 33-in. cast wheels. The results of the tests were as follows:

No. of test	Speed M.P.H. entering sand track	Depth of sand over head of rail	Distance car ran	Remarks
No. 1	14.5	2½ in.	80 ft.	Free rolling.
No. 2	19.5	2½ in.	180 ft.	Front truck left track when leaving sanded section, after slight application of air brakes.
No. 3	15.0	80 ft. of ½ in. 40 ft. of 2½ in.	120 ft.	Free rolling.
No. 4	23.0	80 ft. of 1 in. 100 ft. of 2½ in.	180 ft.	Brakes set and wheels slid on last 40 ft. of test, leaving a coating of sand ¼ in. on rail.

On this road in climbing the sides of the Palisades, the tracks take a zigzag course up the face of the cliff and ascend at an average grade of about 7 per cent. The catch sidings are installed on all the steepest grades and the switches are always left open so that it is necessary for a car to come to a dead stop at this point, and for the conductor to get off to throw the switch for the main line. An automatic tripping mechanism is used whereby the car, after opening the switch, automatically resets the switch point for the catch siding.

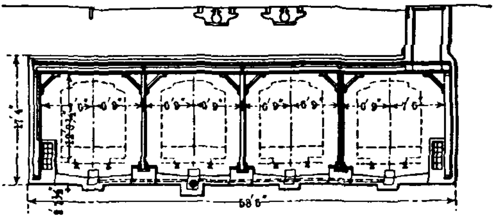


FIG. 46.—New York Rapid Transit Subway. Typical 4-track section above water.

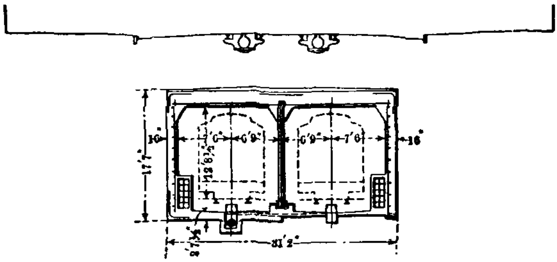


FIG. 47.—New York Rapid Transit Subway. Typical 2-track section, Lexington Ave.

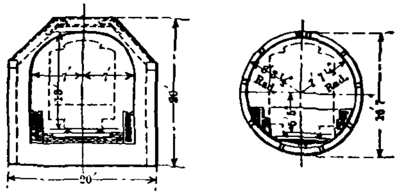


FIG. 48.—Hudson & Manhattan concrete and iron tunnels.

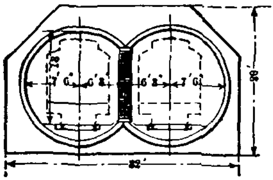


FIG. 49.—New York Rapid Transit Harlem River double tunnel.

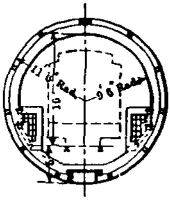


FIG. 50.—Pennsylvania R. R. North River tunnel.

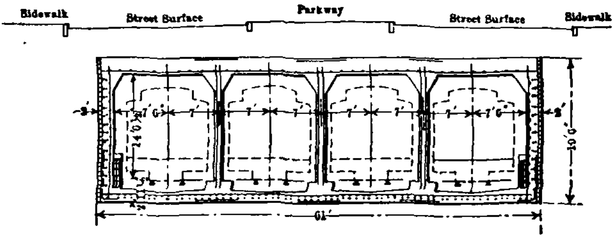


FIG. 51.—Fourth Ave. subway, Brooklyn.

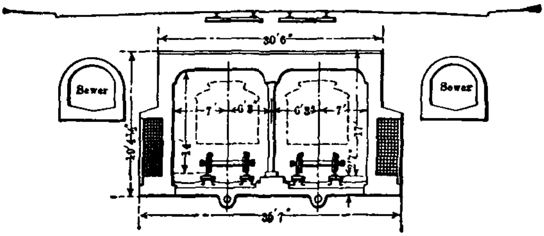


FIG. 52.—Market St. subway, Philadelphia.

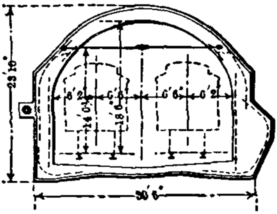


FIG. 53.—Boston subway, Washington St. tunnel.

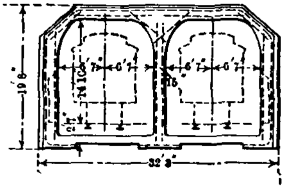


FIG. 54.—Cambridge subway Boston.

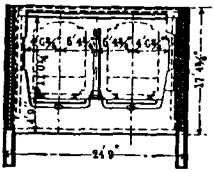


FIG. 55.—Berlin subway, Schoneberg extension.

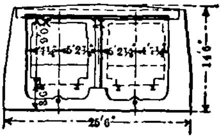


FIG. 56.—Buda-Pest subway.

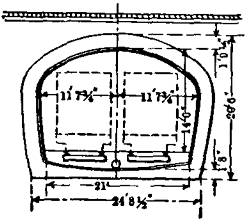


FIG. 57.—Metropolitan railway of Paris.

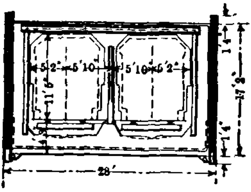
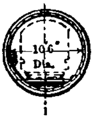
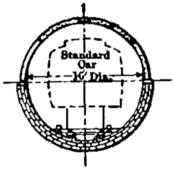


FIG. 58.—Hamburg subway in districts with ground water.





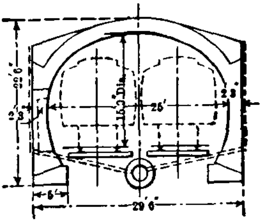


FIG. 59. FIG. 60. FIG. 61.
FIG. 59.—City & South London Ry.
FIG. 60.—Great North & City Ry., London.
FIG. 61.—Metropolitan & District Ry., of London.

Electric Track Switches. In the operation of the Cheatham switch, the car is run under a trolley pan (placed a little more than a car length in front of the switch point) with current on if it is desired that the switch be set in one direction, or with current off

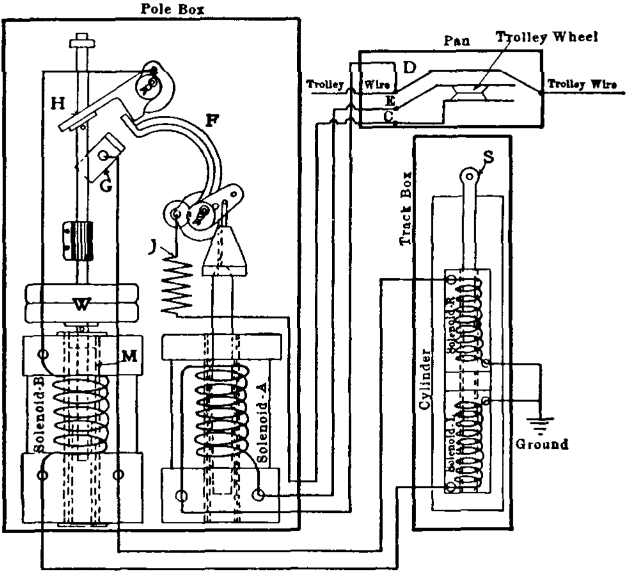


FIG. 62.—Cheatham electric track switch.

if it is to be set in the other direction. Referring to Fig. 62, assume that the car is using current as it passes under the pan. As the trolley wheel strikes contact E of the pan, the current used passes through solenoid A, the plunger of which throws the circuit changer F into contact with plate G; as the trolley wheel proceeds, it makes contact with *C*, and current for operating the switch point passes from the trolley wire through *D*, solenoid *A*, *E*, trolley wheel *C*, resistance *J*, *F*, *G*, and solenoid *R*, to ground, solenoid *R* moving the switch point to the right (in this case) through switch rod *S*. To set the switch point for the other direction, the car takes no current (except lights and heater circuits) as it passes under the pan; these auxiliary circuits require insufficient current to operate solenoid *A*, the circuit changer *F* remains in contact with *H*, and current flows from trolley through *D*, *A*, *E*, *C*, *J*, *F*, *H*, solenoid *B* and solenoid *L* to ground, solenoid *L* moving the switch point to the left (in this case). As current flows through solenoid *B*, its plunger raises the weight *W* slowly against the damping effect of mercury in the cup *M*, and if current flows for an abnormal length of time, as in case of a car at a standstill with trolley wheel in pan, the tripper arm *H* is raised, allowing the circuit changer *F* to fly back and open the circuit between *F* and *H*.

The operation of other electric track switches is similar to that described above, the design of trolley pan, pole boxes and track boxes varying in the various types. A small motor is sometimes substituted for the solenoids in the track box. It is most important that the track box be water tight, and in some types it is filled with oil for better insulation of the contained electric parts. An interlocking device may be added to prevent a following car from operating the switch until the first car has passed over it.

Machinery of the Track Maintenance Department. On account of the increasing necessity for economy in the use of manual labor in recent years, there has been a general trend toward the adoption of special tools, such as improved types of work cars, crane cars and automatic dump cars; an increased use of power shovels both for grading and for loading cars; a rapid adoption of pneumatic or electric tie tamping tools, electric arc welders and power drills in regular maintenance, as well as construction work; a general rearrangement of storage yards, including installation of various forms of labor saving devices therein.

The various forms of arc welders had their origin in the use of the acetylene flame. With the arc welder, defective rail joints can be repaired more quickly, with little or no disturbance to traffic, and for about 10 per cent of the cost, as compared with the method of cutting out the defective joint, installing an insert with two joints, and the incidental restoration of pavement and bonding.

Grinding equipment is a complement to the welding equipment and should be proportionate to it. Grinding apparatus may be classified into four general type groups: (1) push files, hand-operated, reciprocating; (2) rotary grinder, flexible-shaft type; (3) rotary grinder, machine type, mounted on small trucks; (4) reciprocating grinder, machine type, mounted on small trucks or special carriage. The larger machine types and reciprocating grinders are so effective and efficient that at least one machine of each type may be used advantageously on almost any property.

Pneumatic and electric tie tampers have shown remarkable savings. The pneumatic type is in more general use, but certain features commend the electric type, principally that of availability of power. Mechanical tie tamping can be done for half the cost of manual tamping, the men used being of a higher grade but only about 30 per cent in number. The tamping itself is much superior to that obtained with the best hand labor.

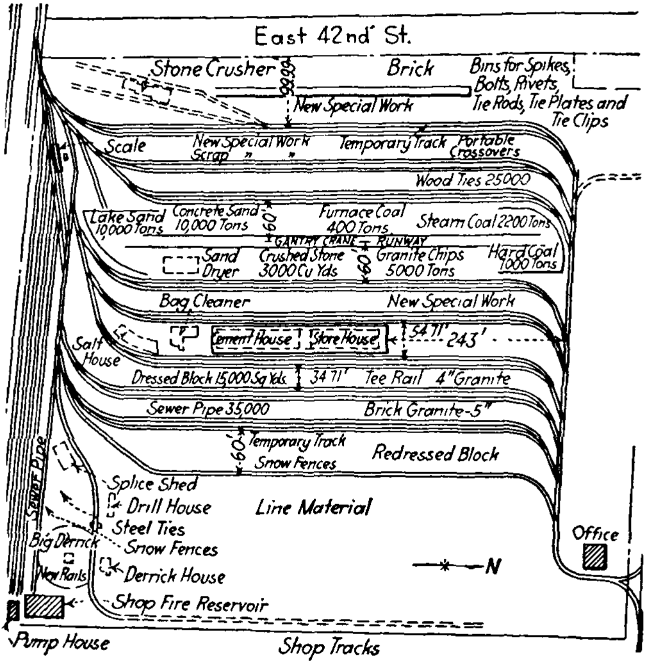


FIG. 63.—Cleveland storage yard.

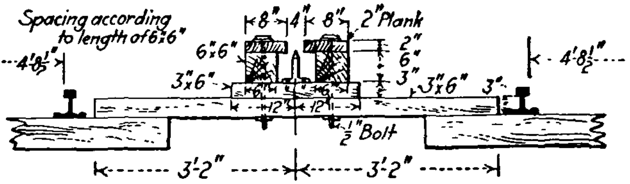


FIG. 64.—Ground trolley contact, between tracks in Cleveland storage yard.

Power drills will cut the cost of drilling operations by 75 per cent as compared with hand drills, the saving being in costs of labor, drill bits and sharpening. Drill grinding machinery of the portable type with special tool holders enable any type of drill bit to be resharpened by the average laborer on the job much more quickly and better than if sent to the shop.

The electric shovel has come into quite general use not only for excavation and grading on extension work but for breaking up concrete and many other forms of reconstruction work Under favorable circumstances it will reduce the cost of excavation by two-thirds.

The pavement plow is an efficient aid in the removal of block pavements of all types It has been stated that it will do as much work of the most difficult kind in an hour or less as five men can do in three nine-hour days, and for less than 25 per cent of the man labor cost.

Both air drills and the so-called "skull cracker" have been effective in removing concrete paving base in reconstruction work. Air drills are used in connection with the compressor apparatus for the operation of tie tampers. The "skull cracker" is a casting of about one ton in weight attached to a derrick or power shovel arm and rigged with a release trigger so as to drop from a height of about 12 feet onto the concrete.

In concrete mixing apparatus three kinds have been developed: (1) moderate sized batch mixers; (2) very large batch mixers with self-loading hoppers mounted on trucks and self-propelling, generally on the track, but sometimes on the roadway surface alongside; (3) the assembly of complete mixing plants on a train of two or more cars. Because of its mobility, the small batch mixer operated by a gasoline engine may be considered the most serviceable for all-round use. The larger self-propelled mixers are particularly adaptable on jobs where a large volume of concrete must be produced rapidly. The special concrete mixing plant mounted on trains of cars has been used in a few instances, its particular advantage being in the elimination of storage of concreting materials on the street.

Work cars have been most radically improved in design for various special operations, it having been found that the use of haphazard old equipment and ordinary flat cars as work cars was most uneconomical. The automatic side dumping car has very greatly saved labor in unloading, as well as saving time of equipment on the road and in lessening delay to passenger car traffic while unloading between cars under regular service conditions. It has been authoritatively stated that automatic dump car equipment has saved as much as 30 cents per foot in the cost of track work. The locomotive crane and the pillar crane mounted upon cars are of particular value in handling special work installations, in loading and unloading rails and in the general handling of materials.

The order of merit of various labor saving appliances, as indicated in a summary of replies to a questionnaire by the *Elec. Ry. Jour* is as follows: (1) crane car; (2) air tamper; (3) arc welder; (4) automatic dump car; (5) concrete mixer; (6) pavement plow; (7) electric shovel; (8) power drill; (9) acetylene torch; (10) rail grinder; (11) stone crusher; (12) ballast spreader.

Hand Tools for Track Work. The following table prepared by R. C Cram, Engineer Surface Roadway, Brooklyn Rapid Transit, shows suggested lists of tools for track construction and maintenance gangs of several sizes.

Tools	Construction, 60 men	Repair, 10 men	Spec. wk. 16–18 men	Section, 6 men
Adzes	2	1	2	2
Axes	1	1	1	1
Bars—claw	5	2	6	3
Bars—lining	20	8	12	4
Bars—tamping	20	4	12	4
Benders—rail (furnish as needed)				
Boots—(furnish as needed)				
Brooms—switch	4	1	4	3
Brooms—push	3	2	2	2
Brushes—wire	2	1	1	
Cars—hand				1
Cars—push	1			
Chains	2		2	
Chisels—cold	2			
Chisels—track	20	8	20	2
Chisels—asphalt	6	2	2	
Compressors—bond	1	1	1	1
Dippers—water	2	1	1	1
Drills—⅞ in (min number)	4	4	5	4
Drills—15/16 in (min. number)	4	4	5	2
Drills—1 in. (min number)	4	4	5	2
Drills—1⅛ in. (min. number)	4	4	5	2
Drift pins	4	4	3	
Files—hand—round	1	1		
Files—hand—flat	1			3
Files—hand—rail	1	1	1	
File holder—rail	1	1	1	
Flags (furnish as needed)				
Frames—Hacksaw (at least 1 doz blades to each frame)	4	4	6	2
Gages—track	2	1	3	3
Gages—wood (insulated)				1
Grinder—keystone tool	1	1	1	1
Hoes	4	1	2	
Hooks—grass				2
Hose—rubber	100 ft	15 ft	30 ft	
Hydrant connection	1			
Jack—track	6	2	5	3
Jack—wrecking	2			
Jim crows			1	1
Lanterns	50	12	35	15
Lanterns—signal				3
Level—boards	2	1	2	1
Level—pocket	2	1	2	
Light clusters				
Mattocks	1		2	1
Mauls—spike	8	4	8	4
Oilers—steel	1		1	
Old men	2	3	2	[illegible]
Pails	6	2	4	3
Picks	20	8	16	10
Picks—tamp.				4
Pipe—water	250 ft			
Plows				
Punches—center	2	1	1	2
Punches—track			5	
Punches—thread	2	2		
Rammers—sand	6	2	4	
Rammers—paving	1	1		
Rasps			3	

Tools	Construction, 60 men	Repair 10 men	Spec wk 16–18 men	Section, 6 men
Ratchets	4	3	6	2
Reamers	2	2	8	
Rules—6 ft folding	1	1	1	1
Saws—crosscut	1		2	1
Saws—hand	1	1	1	1
Scythes				2
Shovels—track sq pt	20	6	10	12
Shovels—L H				4
Shovels—scoop	8			4
Shovels—wood				2
Sledges	10	3	10	1
Spike pullers			2	1
Tongs—rail	12	4	10	8
Tongs—tie				4
Tapes—linen			1	1
Tapes—steel	1	1	1	1
Tarpaulins	1		1	
Tie rod—square	1	1		1
Wrenches—hydrant	1	1	1	
Wrenches—monkey	1		1	1
Wrenches—stillson	1			
Wrenches—compressor	1	1	1	1
Wrenches—track	10	4	12	6
Wrenches—tie rod	4	1	4	1
Wheelbarrows	6	3	6	2
Water barrels		1	2	

SECTION II

CAR HOUSES AND SHOPS

While it is not possible within the limits of this volume to go into any great detail on the subject of buildings, it has been thought desirable to include some notes on the design of car houses and shops from typical structures and from the reports of the Am El Ry Eng Assn committees on Car House Design (1908) and on Buildings and Structures (1922), as well as some notes on the layout of the repair shop which appeared originally in the *Electric Railway Journal*, 1912

Layout of Tracks for Car House

Grade of Tracks. The grade of tracks in the car house should be somewhat above that of the tracks in the street to prevent the entrance of water into the house

Convenience of Operating. In an operating car house the saving of time is of paramount importance, and the special work should be so planned that the incoming and outgoing cars will not interfere with each other, and the tracks should be constructed in the best manner In a storage house inconvenience and loss of time may be tolerated, and here, for the sake of safety on the main line, trailing switches are preferred by many roads

Car-house Track Layouts. Figs 1 to 12, inclusive, illustrate special work designed to meet certain conditions Where convenient to have it, a Y incorporated in the special work will permit the operation of cars from one end, as is necessary with cars of some designs and is convenient when operating service and other cars in tandem This same object can be attained by a loop, which is more convenient but not always feasible, owing to a lack of room Fig 1 shows a track layout where the car house is built on the street line and it is desirable to introduce as few frogs and switches in the main line as possible The three track bay arrangement is objectionable on account of the roof span, which would require trussing Fig 2 shows a track layout which will permit frequent operating on one of a pair of tracks Operating on the other track requires the use of a cross over in the street Fig 3 shows a layout for an operating house which introduces a minimum amount of special work in the street Fig 4 shows a gauntlet track layout beside the main track to minimize frogs and switches in the main track Fig 5 shows a similar layout to Fig 2 except that the operating tracks are on the right hand side of the house Fig 6 shows a layout for a house on a through line operating in either direction The Y is formed by a cross over on the two center tracks Fig 7 shows a large operating house which calls for a long, narrow track approach. The Y is formed at the entrance to the street. Fig. 8 shows a track layout for a storage house which is built well back from the street. Fig. 9 shows a third track approach to a car house, a layout which requires a wide street. Fig. 10 shows a track layout where the approach to the lot is restricted at the street. Fig. 11

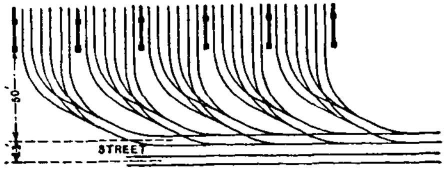


FIG. 1.

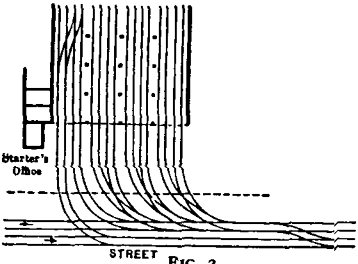


FIG. 2.

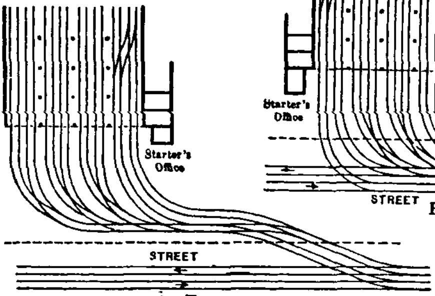


FIG. 3.

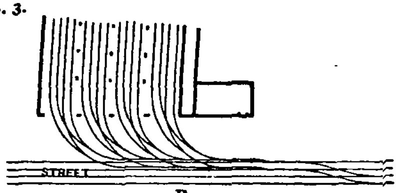


FIG. 4.

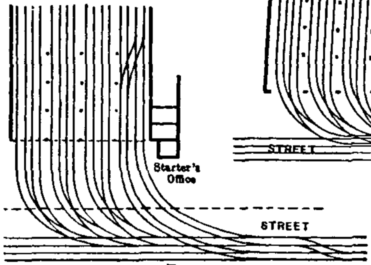


FIG. 5.

FIGS. 1–5.—Car-house track layouts.

shows a common type of terminal layout; special work built with 4-in., 48-lb. T-rail. Inside of 3 years the mates were all renewed and within 6 years the special work was entirely worn out and was replaced with 4¼-in. rails with hardened centers according to the layout shown in Fig. 12. The car house has a capacity of 127 cars, and there are operated from it 336 cars daily on five different lines, or one car every 90 seconds during rush hours.

The track layout of a double-end car house for single-end cars is shown in Fig. 13, which is the Luzerne car house of the Philadelphia Rapid Transit Co.

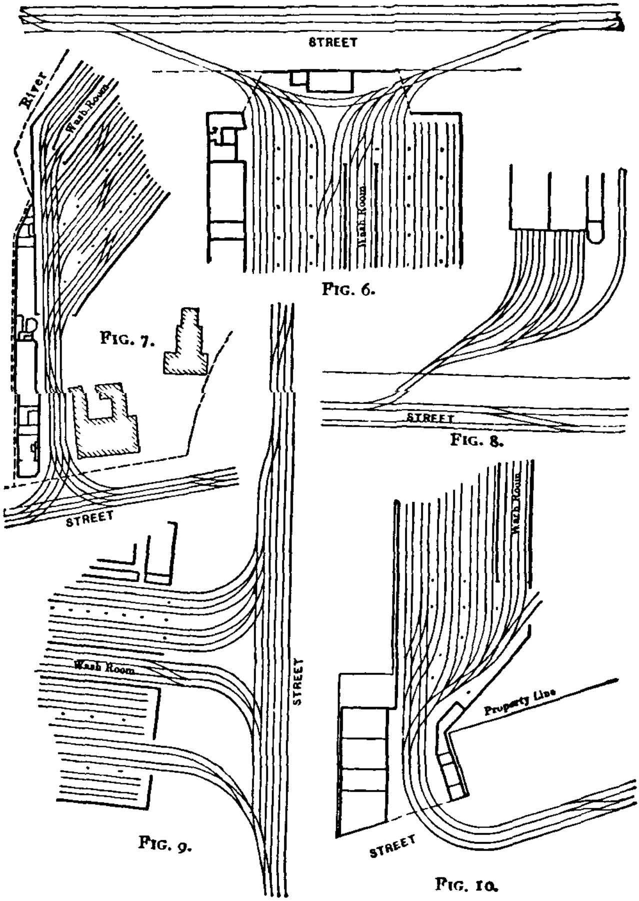


FIG. 6. FIG. 7. FIG. 8. FIG. 9. FIG. 10.

FIGS. 6–10.—Car-house track layouts.

The main feature of the track plan at the Forbes St. terminal of the Pittsburgh Railways (Fig. 14) is a loop around the substation and transportation buildings. One side of this loop is a ladder track having all switch points facing the direction of car movement. Cars, therefore, which approach the terminal from either direction may easily be turned by passing around the loop, and at no time will

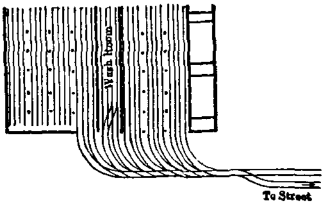


FIG. 11.

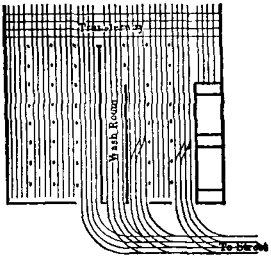


FIG. 12.

FIGS. 11–12.—Car-house track layouts.

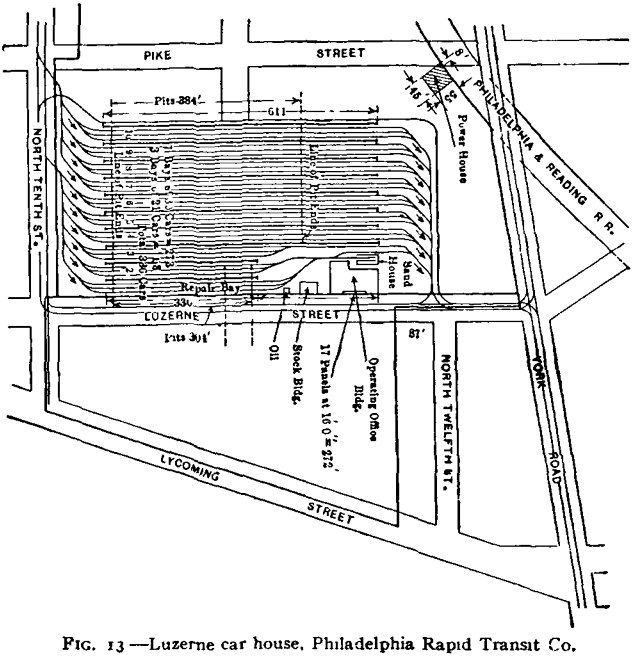


FIG. 13.—Luzerne car house, Philadelphia Rapid Transit Co.

they encounter a facing switch point except when backing into the car house for a trailer or for inspection at the end of the day. This arrangement also makes it possible simultaneously to couple a half dozen trailers to as many motor cars on as many tracks without having one crew get in the way of another. It is the intention to store many trailers here between the morning and evening rush hours, thus saving the dead mileage to their regular car houses at the ends of the several lines. Fig. 17 shows the layout of the Hooker St. car house and yard of the Springfield (Mass) Street Railway, typical of the outdoor storage plan which is gaining so much favor, and an excellent example of combined operating and light maintenance in a single plant

Housing Car-house Special Work. In general, car-house special work requires no more protection from the weather than does special

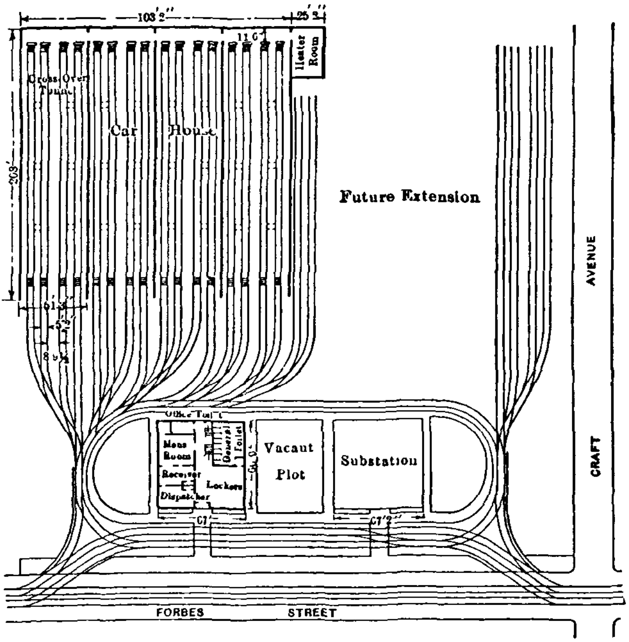


FIG. 14.—Forbes St. Terminal, Pittsburgh Rys.

work on any other part of the track. Whether or not car-house special work, ladder track, for instance, should be covered will depend upon the possibilities of keeping the car house sufficiently warm during freezing weather. In this connection, it should be noted that ice should be melted from incoming cars in time for overhauling. An exposed ladder track will require a door for each track or group of tracks entering the car house from the ladder track, while a covered ladder track will require only one door at the entrance. For this reason, the necessary temperature will be more easily maintained with a covered ladder track. A covered ladder track requires the expense of extra wall and roof, but its trolley wire construction may be simpler and more durable than that for exposed special work.

Clearances. Frequent accidents have shown the necessity of establishing proper clearances between cars and between cars and posts, walls and other fixed portions of the building. The minimum clearance should be 2 ft.; on curves it is well to increase this on account of the constantly increasing size of cars.

Spacing of Tracks. Clearances also determine the minimum spacing of tracks. Eleven feet center to center is satisfactory for a storage house, with 2 ft. additional if posts occur between tracks. This spacing will also do for inspection section of pit room in an operating house, but where a great amount of work is to be done it is desirable to increase this distance.

Transfer-ways. In closely populated districts where it is not feasible to have all tracks entering ones, it is necessary to install a transfer-table. This is equally necessary in a house from which a large number of cars are operated, and where there is constant shifting from closed to open cars and *vice versa*. For convenience, it should be located centrally. Its position inside the building, however, if it passes through party walls (even though fire doors be used in the openings) adds to the fire risk and will affect the insurance rates. For this reason it is often advocated that the table be placed outside the building and at the rear. When so placed, however, it requires the passing of cars through the pit room when being shifted and oftentimes requires the moving of a large number of cars—more than if placed near the center of the building. The objection that a transfer-table within the house causes a loss of storage space can be answered by the recommendation of a flush transfer-table.

Cross-overs. While generally desirable to omit all special work within the house, on certain types of layout cross-overs are necessary within the building for the convenience of operation. A right-hand cross-over is preferred to a left-hand one as being the more convenient.

Design of Car-house Building

Convenience of Operation and Working. For the convenience of operation and working, the operating force should be placed at the front of the building. The starter, or whoever has charge, should be placed where he can see all incoming and outgoing cars; he should be in close touch with the lobby, in order to call the men assigned to duty, and to preserve discipline among the men. The superintendent, or any other official interested in the operation of the cars, having quarters at the car house, should be placed in an equally advantageous position. That portion to be set apart for working and repairs should be placed far enough back in the house to prevent interference from shifting cars.

Quarters Other than Operating. Besides the quarters for operating and repair forces, which are found connected with almost all car houses, some roads desire to provide quarters for such departments as machine shop, blacksmith shop, paint shop and road and line departments. If a power station or substation is to be located on the premises it is better to have it in a separate building from the car house. Some substations, however, are built as part of the car house, but they are of fireproof material. It is sometimes found necessary to provide a garage or stable under the same roof as the car house, but in such a case it should be shut off from the car house by fire walls, and the penetration of gas or ammonia to the storage portion should be prevented. Where operating houses are located away from populous centers it is often desirable for the road to provide some place (a separate building on account of fire risk) for employes to obtain meals, or interest other parties in providing accommodations of this kind. In similar locations quarters for sleeping are also desirable for men who have to be held for work on snow or night duty. These quarters may be located in the car house. Fig. 15 shows the layout of the Middletown car house and shop of the

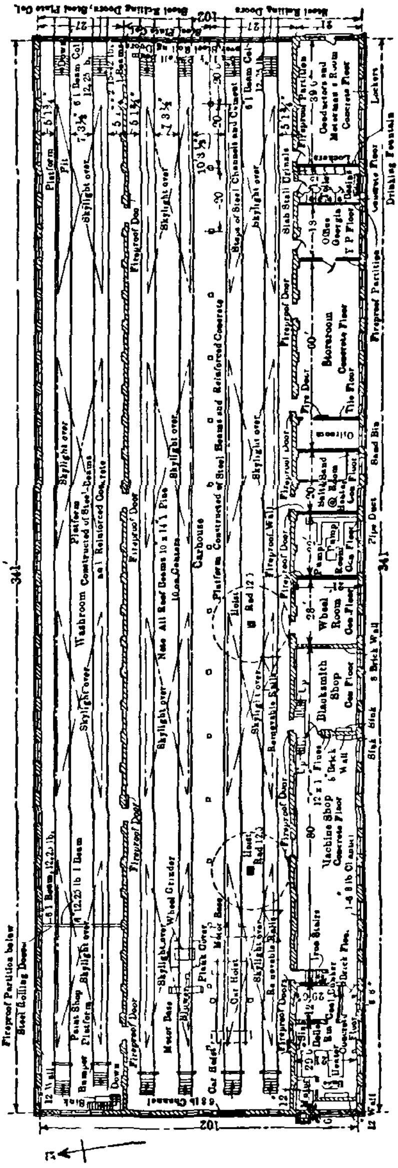


FIG. 15.—Middletown car house and shop, Connecticut Co.

Connecticut Co. The offices and shops are arranged along one side of the building. The first room to which entrance is obtained is that for the conductors and motormen, where lockers, a drinking fountain and other conveniences are installed. Following this are the toilets, the master mechanic's office, storeroom, oil room, salt and sand room, pump room, wheel room, blacksmith shop, machine shop, coal storage and boiler room.

Fire Protection. The importance of providing fire protection should not be underrated, and at no time should it be forgotten. In selecting the location for the house, proximity to buildings of inflammable material or having contents that burn easily should be avoided. Car houses should also be located where a good hydrant service is obtainable, and as near to a fire station as possible. If this is not done a private service with ample supply of water must be installed. Provision must also be made for a secondary supply of water, either in reservoirs under floor, or in elevated tanks.

Fire Prevention. No road should expose more than a certain per cent of its rolling stock to the risk of destruction by any one fire, as the loss of cars means the loss of revenue. Divide the house by party and curtain walls where it is practicable to do so without interfering with the operation of cars or increasing the cost of the building to a prohibitive extent. The omission where possible of combustible material is recommended, especially below the grade of top of rails. Its use should be avoided on outside walls and cornice work. Sheathing partitions inside are undesirable, and floors and roof, if they are to be of wood, should be mill construction, with heavy timbers and planking.

In the construction of a car house, corners and recesses where rubbish would tend to accumulate should be avoided. Special care should be taken to avoid the accumulation of oily waste and rags. Fireproof receptacles should be provided for such material and they should be regularly and frequently emptied. Ample light and cleanliness will aid greatly in reducing the possibility of fire to a minimum.

Insurance Regulations. It is recommended that the designer familiarize himself with the regulation of the fire underwriters and adopt their suggestions as far as practicable. Insurance requirements are important factors in determining many details of construction, and a consultation with the underwriters when planning may save many expensive changes. The underwriters stipulate that no section of the house shall contain more cars than amount to the value of $200,000.

Automatic Sprinklers are to be preferred as the best possible fire protection in enclosed places where the sprinklers will be opened by the heat of the fire. The very fire itself which is sought to be extinguished sets in operation the influence which extinguishes it, at the particular spot where the fire is and at no other, and entirely independent of human action. This applies to all buildings whatsoever. There should always be two sources of water supply, and any two of the following methods can be adopted: city water with adequate pressure; elevated tank; pressure tank; underwriter fire pumps.

Auxiliary Protection. Sand pails, chemical extinguishers and water pails should also be provided. Small hose lines are advisable for reaching sparks and flames in places not reached by water from the sprinklers.

Outside Protection. For open spaces, including yards, automatic sprinklers are not usable, since the heat will not operate to open the sprinklers. For such places the available methods of protection are: (*a*) Universal nozzles on standpipes; (*b*) standard fire hose and nozzle; (*c*) open sprinklers set in operation by human agency.

Universal Nozzles on Standpipes are the best protection. There is no possibility of delay during a serious fire through the bursting

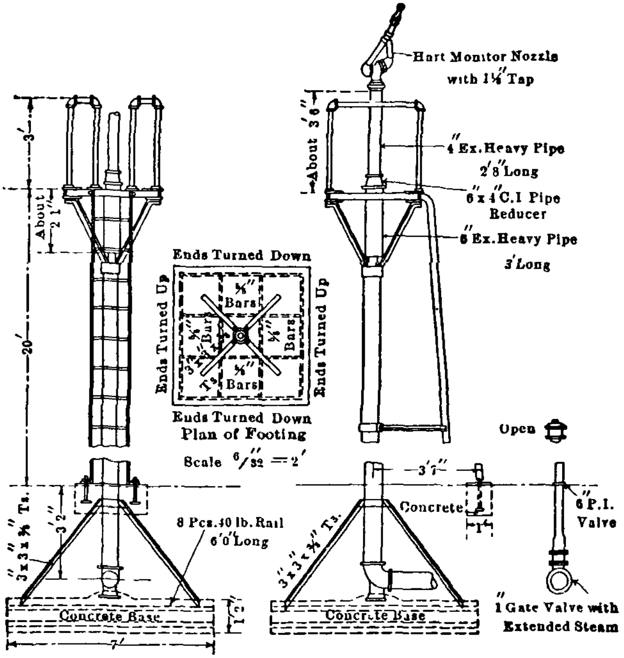


FIG. 16.—Water tower, Puget Sound Traction, Light & Power Co.

of hose occasioned by kinks, or momentary excessive water pressure, or by the accidental cutting of the hose by the running of the cars over it in trying to remove them from car yards. The nozzle should not be less than 1⅛ in. or more than 1¼ in., if the water supply and pressure can be had. Since the operator cannot move such nozzles bodily to the place of the fire, the range of the nozzle should not be more than 100 ft. in order to secure the greatest degree

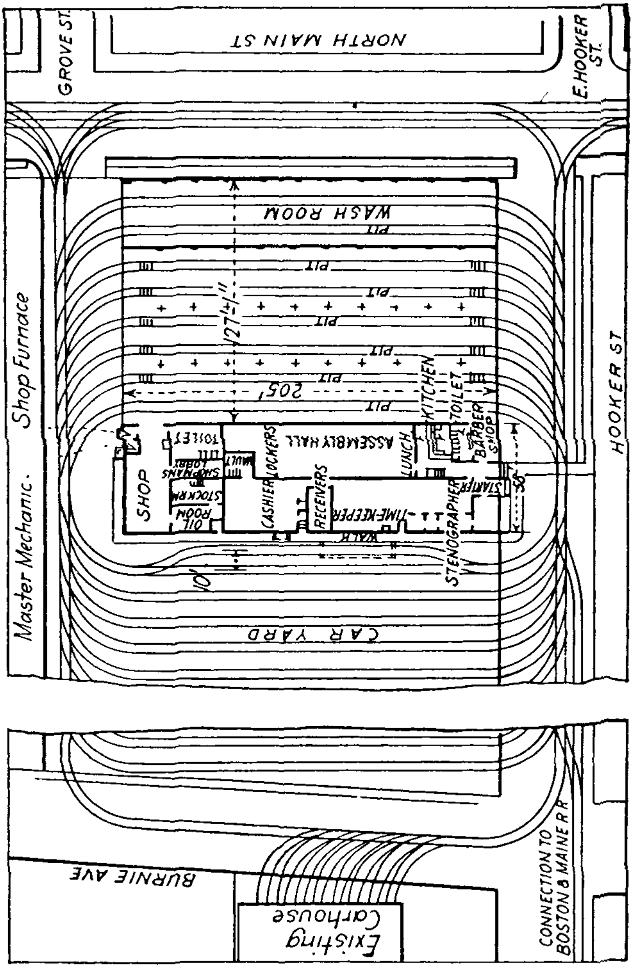


FIG. 17.—Springfield (Mass.) car house and operating yard.

of efficiency, which distance will necessitate as high a pressure as can be safely maintained. High water pressure is advisable for the further reason that the operator may be at a safe distance and still place water upon the fire. Intense heat may prevent the operator from placing water upon the spot desired if the water pressure is too low. This condition applies to nozzles on hose as well. The universal nozzle should be located at a height of from 10 to 12 ft. above the tops of cars. The pressure at the nozzle should be 100 lb. if possible. For this pressure the discharge for 1¼-in. and 1½-in., 1¾-in. and 2-in. nozzles will be, respectively, 466, 671, 904 and 1194 gal. per min. It is believed that the effective way to extinguish a fire, especially one which has gained much headway, is to concentrate one or more heavy streams of water upon one particular car in the yard and maintain it there until its effect is shown. This can be readily accomplished with a standpipe and universal nozzle. Pumps should be provided to supply at least two nozzles at one time. Nozzles should be located so that their range circles will overlap safely. Universal nozzles are the best protection for practical installations where cost is considered; they are certainly to be preferred over standard hose and nozzle in yards where cars are stored. Universal nozzles should be supplemented by small hose and nozzle for getting at sparks and flames not accessible to the stream of the universal nozzle, also by water pails and chemical extinguishers.

Fig. 16 gives the details of such a water tower as installed in the yards of the Puget Sound Traction Light & Power Co. The towers are connected to the city water main by a 6-in. supply line, and each system is equipped with a steamer connection for the use of the city fire department. In order to cover the yards thoroughly, the towers were spaced so that any car could be reached by at least two streams. Tests show that where the static pressure is only 65 lb., the nozzle pressure is 47 lb. and the discharge 255 gal. per minute. With a static pressure of 125 lb., the nozzle pressure is 88 lb. and the discharge 350 gal. per minute. Thus a car can be flooded with water at the rate of 400 gal. or 500 gal. per minute, making it almost impossible for fire to spread to neighboring cars.

In a proposed layout for the Cleveland Railway the yard is 750 ft. × 454 ft., including car-inspection structure, offices and utilities buildings. The 18 standpipes are located around the yard at intervals of 90 ft. The standpipe nozzle tops are 1⅛ in. in diameter and the supply mains are of 8-in. and 6-in. diameter. Two hydrants are also installed to reinforce the interior sprinkler protection of buildings. The water supply is forced through the mains by a 750-gal. electrically driven underwriters' pump of centrifugal type, taking suction from a 75,000 gal reservoir maintained on the premises. Under maximum pressure the fire pump will deliver 1000 gal. of water per minute, which can be concentrated on any one car in the yard.

Standard Fire Hose and nozzles are the next best available protection. Hose should be 2½ in. for ordinary cases. Nozzles should be not less than 1⅛ in. Pressure should not be too great, because men who are not professional firemen cannot handle the nozzle under great pressure. Pressure at nozzle should probably be 40 lb. to 60 lb.

Open Sprinklers operated by human agency are probably a very effective means for checking an early fire, and perhaps for extinguishing a fire which has gained more or less headway, but the cost appears to be too great for general use, although for small installations, where the sections are not large and where the controlling valves can be operated by hand the cost may warrant the use.

Details of Car-house Design

To design the most economical car-house building for a particular service in a particular location the designer must be able to design the several types of construction in detail and he must be able to estimate the costs of the materials and labor necessary for the finished structure. The general determination of the most econom-

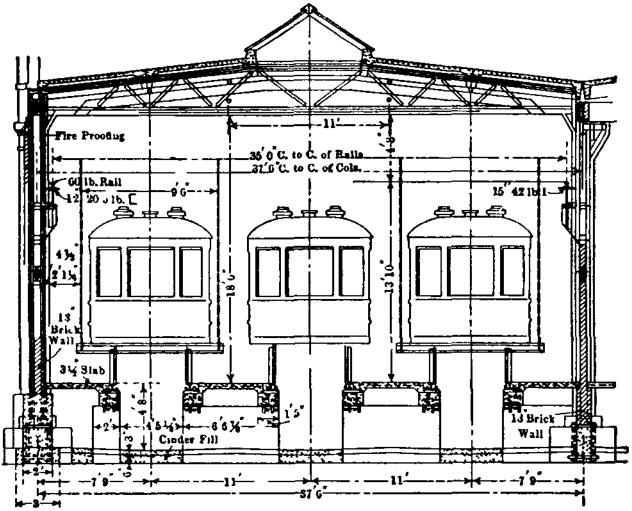


FIG. 18.—Luzerne car house, Philadelphia. Cross-section of bay for light repairs.

ical design of car house consists in a determination of the most economical foundation, wall, form of roof and form of skylight. These individual parts must finally be considered with regard to their suitable relationship to each other. Among the general decisions which must be made at the outset by the designer are: Shall the building be short and wide or long and narrow? Shall two-, three- or four-track units be used? Shall the foundation be of stone, brick or concrete? Shall the walls be of brick, terra-cotta, concrete blocks, iron or concrete? Shall the roof framing be of wood, unprotected steel, protected steel, reinforced concrete or a combination of these? These must finally be decided with regard to their

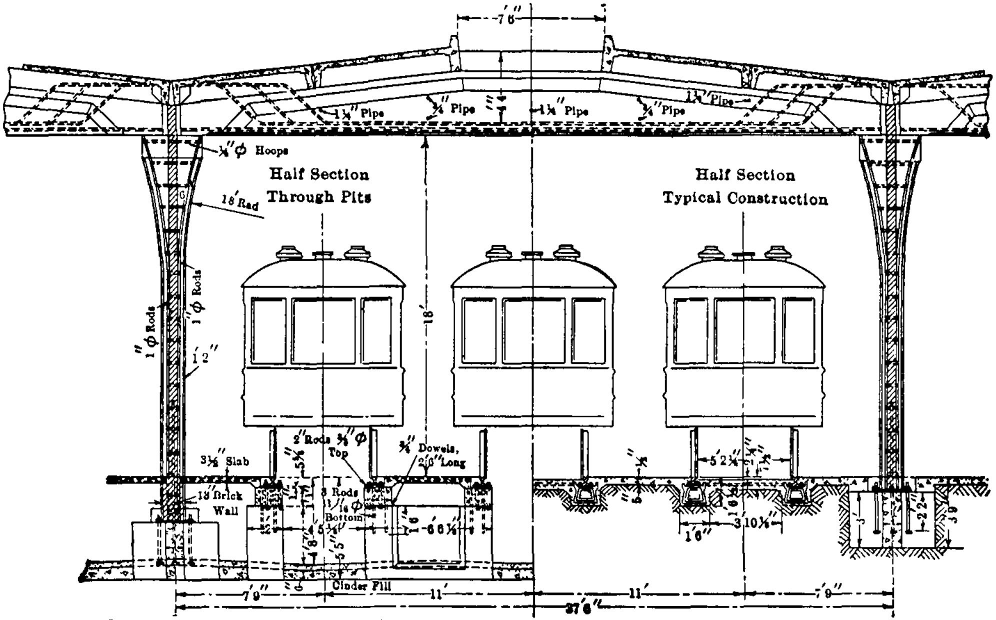


FIG. 19.—Luzerne car house, Philadelphia. Pit section and flush section of inspection bays.

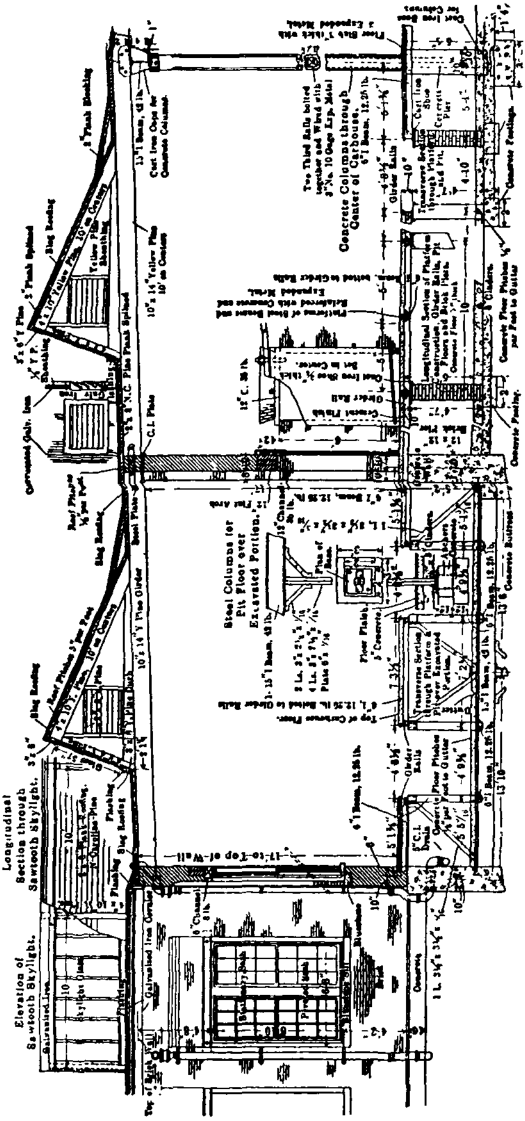
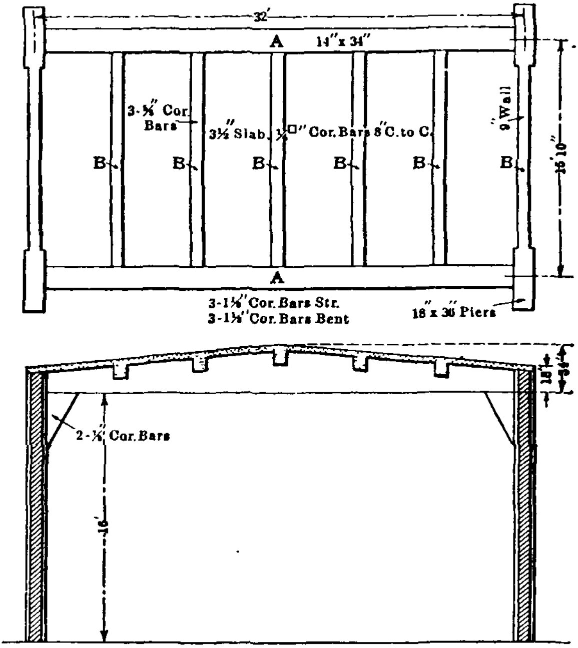


FIG. 20.—Middletown car house, Connecticut Co.

suitable relationship to each other. At the outset the designer should acquaint himself with the following: municipal regulations and building ordinances, rules and requirements of fire underwriters, character of soil upon which the foundation is to rest and the costs of materials and labor.

Figs. 18 and 19 show the construction of the Luzerne car house of



	Quantity	Cost
Concrete:		
Girder A, 26×14×36.15÷144	82	
Girder B, 5×8×14×14.66÷144	57	
Knees, 3×4×14÷12	14	
Slabs, 14.66×28.9×3.5÷12−4.75×4.66×3.5÷12	104	
Total concrete at 20 cents	257	$51.40
Steel:		
Girder A, 6−1⅛c. r. ×34	895.	
Girder B, 5×3−⅝c. r.×17	344.	
Knees, 2×2−⅞c. r.×9	95.5	
Slabs, 1×16×34× area ¼ sq. c. b.	190.	
Total steel	1524.5	45.74
Forming:		
2560 b. m. at 25 cents per 1000=65+5 for knees		70.00
Total cost		$167.00
Total cost per sq. ft.		0.33

FIG. 21.—Typical study sheet for car-house construction material.

the Philadelphia Rapid Transit Co. The girders, posts, roof slabs and floors are of reinforced concrete and the walls are of brick. The concrete portion was built by the unit method, whereby the concrete members are cast separately and lifted into place.

Fig. 20 shows the construction of the Middletown car house of the Connecticut Co. The floors and footings are of concrete, the walls are of brick and the roof is of mill type construction framed with yellow pine girders.

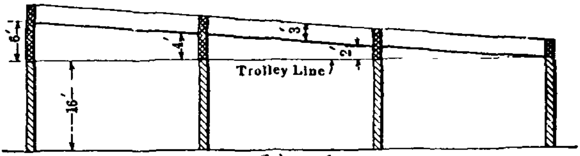
Foundations. If the soil is of good bearing capacity little study need be given to the matter of foundations. A car house is not a heavy structure, the walls being light, and the load of cars being well distributed by track work. The materials to be selected for foundations are largely determined by the supply near at hand. Concrete is probably as common and as satisfactory as any. Where soft yielding soil is encountered and piling found necessary, concrete piles have been used. In such cases and under certain conditions they undoubtedly prove more economical than wooden ones.

Walls. Brick masonry or concrete monolith seems to be the most suitable construction. If a cheaper form of wall is wanted, a heavy mill frame with a 2-in. cement plaster curtain makes a good fire-resisting scheme and one which is inexpensive to maintain. Corrugated iron is sometimes used, but is not recommended. There are in the market several forms of asbestos boards, but their cost does not warrant their selection in preference to masonry.

Roofs. A flat type of roof is the only suitable one for a car house. It should be as close to the trolley wire as possible. Light can be brought through the roof by means of skylights or monitors. The Am. El. Ry. Eng. Assn. committee strongly favored a mill type of heavy construction for the roof and advised against the use of steel. The use of concrete for roof construction may be advantageous because of its fire-resisting quality. Its cost, however, may be prohibitive especially where long spans are necessary. Water should be taken off by valleys rather than by gutters and taken down within the building. For the covering of the roof there are a number of materials, all of which have some merit, but none of them are any more satisfactory than a tar and gravel roofing well laid with first-class materials.

Fig. 21 by C. A. Neff and T. P. Thompson, *Electric Railway Journal*, 1913, is a typical one of forty odd study sheets used by them in determining the proper roofing design for the car house of the Virginia Railway & Power Co. These sheets covered the various systems of steel supports with slabs, tile, plates as well as "monolithic" and "unit built" structures.

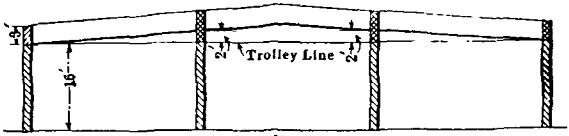
Fig. 22 is a study sheet prepared by these engineers to show the results of an investigation of three different plans of roofing three units in one building. Scheme No. 1 shows the roof high on one side and sloping all one way. Scheme No. 2 shows the roof high in the center of the center unit and sloping two ways, scheme No. 3 shows the roof high over the center of each unit and sloping two ways over each unit. In each scheme the slope per foot is the same. As the necessary distance between track and trolley was fixed at 16 ft., it was considered that all wall used above the horizontal line 16 ft. above the tracks should be charged against each scheme in considering the relative cost. In scheme No. 3 there are only 2 ft. of brickwork above the trolley line at the outside wall, whereas in schemes Nos. 1 and 2 there are 3 ft. of brickwork. This results from the fact that in scheme No. 3 a girder 3 ft. deep in the center and only 2 ft. deep at the wall line was possible, whereas girders 3 ft. deep throughout were required in the other two schemes for the same strength. For purposes of comparison, the walls were considered as 13-in. brick walls, the figures being based on brick laid in the walls at $15 per 1000 and each wall being 400 ft. long.



Scheme 1

Brick Work above Trolley Line Indicated thus

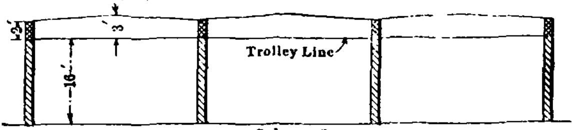
(9+7+5+3) x 800 x 18 x .015 cts.=$5184.00



Scheme 2

Brick Work above Trolley Line Indicated thus

(3+5+5+3) x 800 x 18 x .015 cts.=$3456.00



Scheme 3

Brick Work above Trolley Line Indicated thus

(2+2+2+2) x 800 x 18 x .015 cts.=$1728.00

SUMMARY

Scheme No. 3 saves over scheme No. 1	$3,456.00
Scheme No. 3 saves over scheme No. 2.	1,728.00
Scheme No. 3 saves over form of roof on certain car houses built by others	5,580 00

FIG. 22.—Typical study sheet for car-house roofing schemes.

Posts. The 1908 Am. El. Ry. Eng. Assn. committee favored the adoption of posts for the roof supports in preference to trusses. In the pit room they provide a means for supporting the car handling apparatus; they are convenient for holding the aisle sprinkler pipes, and also the standpipes and hose. They can also be used to support brackets for the fire pails. Where there are posts it is easy to introduce curtain walls, either the entire height of building or dropped down 6 ft. or more from the ceiling. Plaster concrete partitions along lines of roof posts make excellent fire curtains.

Trolley Troughs. One of the most satisfactory ways of holding up the trolley wire is by a suspended plank. The plank serves not only this purpose, but also when the trolley pole leaves the wire, prevents the grounding of current through contact of pole with sprinkler pipes, etc., in the building.

Bumpers. At the dead end of every track, and 3 ft. from the wall, there should be a stop to prevent the cars doing damage. There are several kinds of stops, such as the common cast-iron shoe bolted to the rail or fastened with clipping blocks, and bumping posts, consisting of hard pine timber buried in the ground, or concrete. There is also the post secured with bolt and straps to the rail.

Doors. Doors are necessary for those portions of the house which have to be heated and those storage portions that are seldom used. They are less necessary on other parts of the house and may be omitted if one is satisfied that there is no fear of intrusion by those bent on thieving, setting fire or other malicious mischief. It is desirable to have the doors swing out. Both swinging and roller types of doors are being used. The former has many points in its favor, but the latter is a great convenience where room will not permit the use of swinging doors. Where swing doors are used they may be partly glazed. Roller type doors are sometimes motor driven for greater convenience and more rapid operation.

Floor Construction (Recommended by Am. El. Ry. Eng. Assn., 1911, Committee on Buildings and Structures).

(*a*) *Car Houses.* Concrete floor with cement finish is best for permanent construction. Otherwise crushed stone and screenings, or ashes is advised.

(*b*) *Car Shops.* Concrete floor with cement finish is recommended, except in machine shop, if subjected to heavy service, where creosoted wood block on concrete foundation should be installed.

(*c*) *Power Houses.* For engine room, concrete with cement or tile finish is advised. For boiler room, concrete and cement finish or a floor of brick laid on edge in cement mortar is recommended.

(*d*) *Offices, Employes' Room, etc.* Maple or combed-grain yellow pine wearing floor should be used. If wearing floor is supported by sleepers embedded in concrete, it is always best to install a false floor underneath the wearing floor.

(*e*) *Toilet and locker-room floors* should be of concrete with cement finish, and connected with sink and trap to drain, so that the floor may be conveniently washed and scrubbed down.

(*f*) *For waiting rooms or shelters and platforms,* floors of concrete and cement finish are recommended for permanent structures. Rolled broken stone and screenings or ashes may be satisfactorily installed for cheap open shelters for temporary construction.

Lobbies. As already stated, the lobby should be under the eyes of a supervising official; and any plan by which the men are left to themselves is undesirable. The lobby should be provided with seats, tables, bulletin boards, and the like; counters should be arranged for the filing of time sheets and other papers requiring the signatures of conductors. All seats are better attached to the wall for the sake of cleanliness. Metal lockers should be provided directly in the lobby or in an adjoining room. For those men who are of a quiet disposition a separate room should be provided, where reading and the playing of quiet games can be indulged in. This room should have a table for letter writing. Separate rooms are also advisable for making out time sheets, reports and accident blanks. All parts of the lobbies and toilets should be so designed as to suffer as little as possible from the destructive work of idle hands. Adjoining the lobby should be ample toilet facilities, with provisions to keep them in sanitary condition. The floor should be such as to admit flushing; partitions should not run to the floor; as little wood as possible should be used; sharp corners should be avoided; and there should be good provisions for light and air. In the larger houses there should be equally well-planned toilets adjoining the pit room, and possibly a private toilet for the officials. If the house is small a central toilet with one or two water closets under lock and key may answer for all employes.

Pits. The pit room, one of the most important parts of the house, should be placed well back, and the area should represent

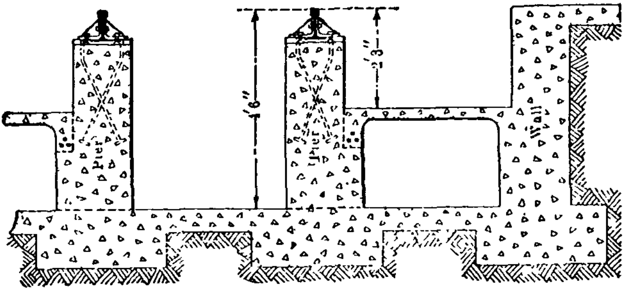


FIG. 23.—Section showing drop floor beside pit.

a capacity of at least one-third—or, better, one-half—of the car storage. It should be well lighted, to save artificial light, and amply heated for the sake of the comfort of the men and for the melting of the snow around the trucks. Pits should have an average depth of 4 ft. 6 in. from the top of the rail. Preferably, the floors of the pits should be of concrete and slightly crowned to shed water. The main floor on one or more tracks should be dropped next to the rail on each side of the pit to facilitate working about the truck and the running boards. Fig. 23 shows a drop floor at the side of the pit. This floor is 27 in. below the top of the rail. It can be less if preferred, and with some equipments 12 in. proves satisfactory. The natural lighting for the pit room is through the monitor of saw-tooth type with vertical glass. A cross, or transverse pit, with the floor slightly below the floor of pit and terminating in a room at the side, is a convenience for moving material from pit to pit. The room can be used for storage or, if well lighted, for a work room. An opening can be provided so that material can be lifted to the floor above. By referring to Figs. 23

to 29, several schemes of pit construction will be found illustrated. (See also Figs. 18, 19 and 20.) A form of construction that utilizes the rail for the support of the floor, as well as to carry the cars, will be found simple and economical. It also does not pocket the heat under the floor and gives greater clearance for working and for passing from pit to pit. In the form of pit construction shown by Fig. 29, a single pit sufficient in size to cover all tracks is constructed in the car-house floor. Tracks over this pit are supported on columns made of a single 3 × 3 × 5/16 in. angle with a gusset plate at the top to serve as a point for attaching a strut, two diagonal tie rods and the rail clips. Two angle lugs at the bottom serve to spread the load on the concrete pedestals and as a means of anchoring the supporting structure to the pedestal foundations. These track columns are spaced 9 ft. 6 in. apart and are built in bents similar to bridge buck braces. They are paired to support the rails of parallel tracks, and are stiffened diagonally by round tie rods with turnbuckles for adjustment. A 4 × 3 × 5/16 in. angle between columns serves as a strut to prevent overturning and as a support for the walkway between the tracks. In order to provide sufficient girder strength between the pit columns 100 lb. A.S.C E. rail has been used to span the 9-ft. 6-in. interval between them. The method of fastening the rail at the columns is by means of clips bolted to the horizontal leg of the structural angle strut.

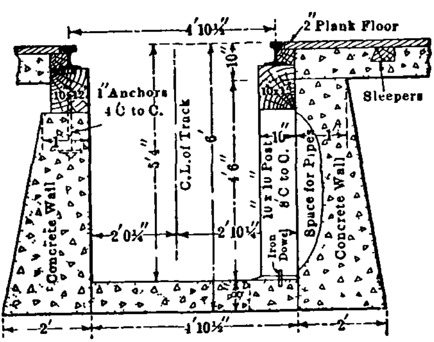


FIG. 24.—Section of pit construction deeper and more expensive than usual supports.

An installation of push buttons by means of which errand boys may be called to pits will save much time.

Apparatus. While the apparatus for the handling of the cars is not a matter of car-house design, still there must be provision made for its installation. Supports sufficient to take cranes and body hoists must be provided, as well as provisions made for motor and wheel hoists. An example of wheel drop pit and wheel hoist is shown in Figs. 25 and 26 Motor hoists may be constructed similar in design to Fig. 26, but care should be taken in location that the wheel hoist does not tie up the motor hoist, as the operation of changing motors or changing armatures in split-frame motors usually requires much less time than that of changing wheels.

Stock Room. The stock room should be placed convenient to the pit room and should be of sufficient size to contain all the stock. It should not be so large as to make the care of stock difficult, but

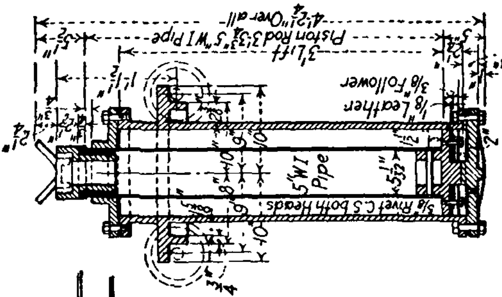


FIG. 26.—Pneumatic pit jack.

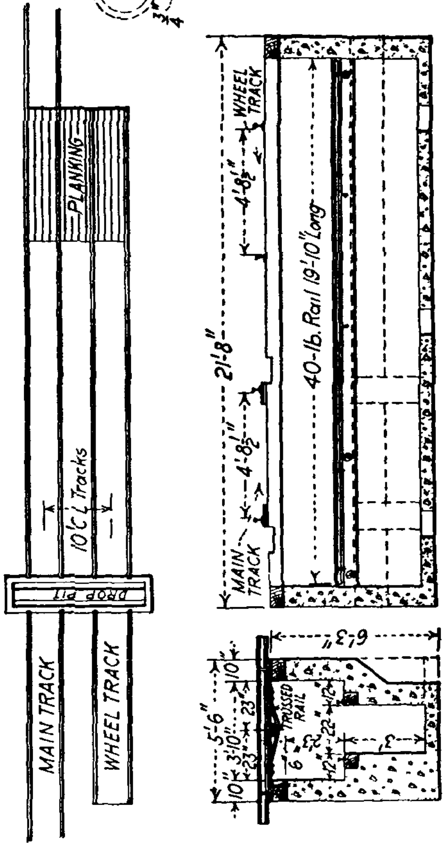


FIG. 25.—Wheel drop pit, El Paso & Southern.

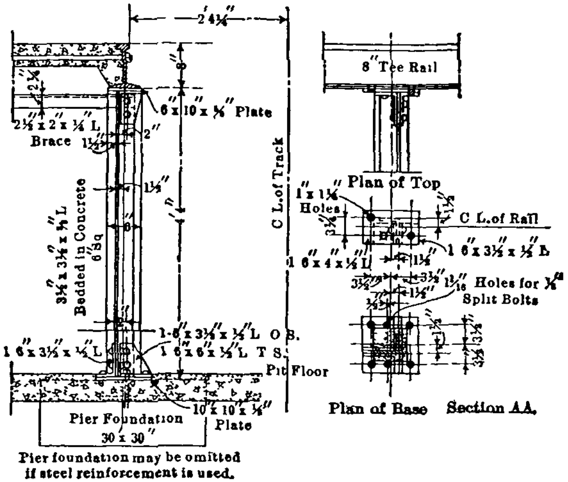


FIG. 27.

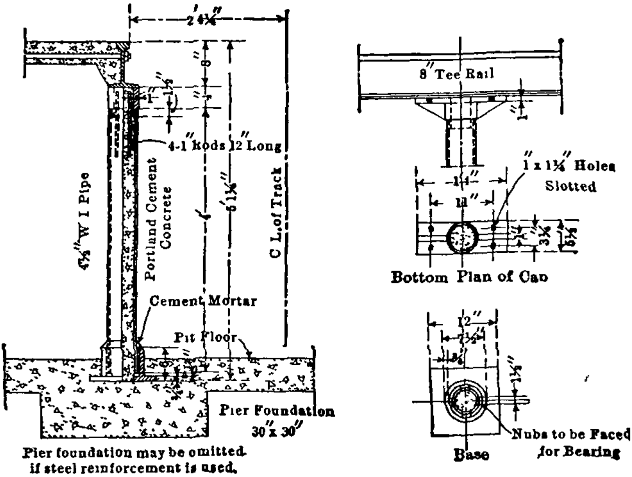


FIG 28

FIGS. 27, 28 —Types of pit construction with 8-in T rail on pier or column supports.

should be a well-lighted, neatly designed room which will inspire the keeper with neatness (and neatness means economy where there are a large number of small parts to be kept in stock). Broad benches with lockers beneath them, drawers for screws and small parts, and bins neatly lettered and painted will be an object of pride as well as an incentive to saving

In the movable bin construction used in the stock room of the Syracuse Rapid Transit Co the bins are of galvanized iron and are built up as follows: Carrying channels are bolted vertically to the wall at 26-in centers and are perforated for the insertion of the shelf brackets at any desirable intervals The shelving consists of horizontal sections 12 ft. long which have a series of holes 4 in apart so that the vertical partitions can be riveted to the shelf at intervals of 4 in or multiples thereof.

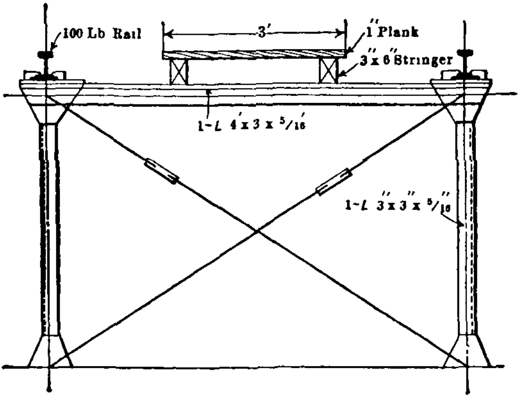


FIG 29—Pit construction on structural steel.

Blacksmith Shop. If any repairs are contemplated, it will generally be found necessary to provide a place for blacksmith work This place should be at the end of a track so that the trucks can be run in The small type of blacksmith shop will require a work bench and an opportunity to install a few hand-drilling machines, etc As the amount of work and the size of the house increases, more machinery is required With small roads a fully equipped machine shop is maintained for doing all the work of the road, including a small but well equipped foundry for brass work.

Paint Shop. A small paint shop for the renovation of cars is provided as a part of the car house on some roads The paint shop must be so located that it will be absolutely dustless If the building in which it is located is heated indirectly, the air should be blown into the paint shop first Preferably, however, the paint shop should be heated by steam, even though other parts of the house use hot air.

Wash Room. The use of the wash room and the washing of cars with the hose is being abandoned by many roads; instead, the entire floor is drained, and with hose bibbs at frequent intervals the cars are washed at any point by the use of pail and brush.

Oil Room. While some prefer a separate building for the oil storage it can be made a part of the car house, if properly designed. The floor should be dropped below the outside floors, dished to the center and properly drained. There should be barrel racks, shelves for small vessels and waste receptacles, all of incombustible materials. Proper provisions should be made for heating the oil so that it may be fit for use in the winter. Tank systems, having the tanks buried outside in the ground and the oil piped into the house, afford a good method of caring for oil and one not liable to much waste. Am. El. Ry. Eng. Assn. Manual, Section B206-21, describes a design of oil house which, it is stated, is economical and convenient from an operating standpoint, and at the same time meets all municipal and insurance requirements.

Sign Room. It is convenient to have racks for the storage of signs used on cars and they should be convenient to the operating section, so that the shifting of signs can readily be made. There should be facilities for the pasting of signs for the dashers. This work should be done in a room set apart for it.

Sand. It is important to provide for the storage and drying of sand. The apparatus for the drying should be placed close to the operating end of the house and should allow circulation of air as well as of heat. The process of drying is slow at the best, so that it is well to have a large storage space adjacent to the heater. This space should be filled with air-dried sand in the summer, and, with the assistance of the heater, an ample supply can be kept on hand during the winter. In the Luzerne car house of the Philadelphia Rapid Transit Co., an elaborate sand-storage and distributing equipment has been installed to make it easy for the crews to fill their boxes with clean sand in the shortest possible time and without placing bins throughout the building.

For this purpose the company built a sand house where the sand brought from the drying plant in hopper-bottomed cars is dumped and then raised by a motor-driven bucket elevator to a belt conveyor which distributes its load throughout the sand-handling panel extending across the rear of the car house. This panel is 16 ft. wide and is lighted by means of steel-sash windows in the rear wall. Outlets from the bottom of the sand storage are provided at the partition walls with a supply pipe for each side of the partition. To obtain sand, it is necessary only to raise a weighted valve and the sand receptacle will be filled.

Salt. Less space is required for the storage of salt, but full provision for its handling is just as necessary.

Heating Systems. The following systems are recommended by the Am. El. Ry. Eng. Assn. (Manual Section B203-11):

(*a*) For car houses, a blower system, where the air is blown over steam coils and through the building.

(*b*) Large car shops, the blower system, except in the paint shop, where direct steam radiation is advised. For the small shop, either direct steam or a hot water heating system.

(*c*) Isolated waiting rooms, only practicable to heat if attended—generally, standard coal stoves are advisable. If waiting room is large and pretentious, steam or hot water heating system may be used.

(*d*) Unisolated waiting rooms. Direct steam or hot water system is generally advisable.

(*e*) Small isolated ticket booths and the like may be satisfactorily heated with the ordinary electric car heater.

Lighting. The pit room should be well lighted, but a proper disposition of lights will make their number comparatively small. One satisfactory and economical arrangement is to place lamps on one side of pit about 18 ft. on centers. Each lamp is portable and enables the inspector to place it close to any desired point, and a hook attached to the lamp allows it to be hung up. Sufficient light should be provided in those places where trolleys are usually shifted. For the sake of the eyes, light should not be stinted in the reading, report and other rooms attached to the lobbies. All interior wiring should be installed in conduits.

Wiring. Rule 41 of the National Electric Code, reproduced in Section B200-23 of the Manual of the Am. El. Ry. Eng. Assn., should be consulted in connection with the installation of electric wiring in car houses and repair shops.

Road and Line Departments. When quarters for the road and line departments are to be furnished, they should be placed so as to have yard room adjoining. The headquarters themselves should be of ample size to take the more or less bulky materials that are used. Proper provisions should also be made for the locking up of copper and other valuable portions of the stock.

Express Accommodations. Companies doing or contemplating doing an express business may wish to establish a terminal at the car house. In such cases the room should be so situated that the express cars and teams receiving and delivering shall not interfere with the operation of the regular cars, and will not be in the way of future improvements of the house. It should also be placed so that the clerk in charge could, if he had spare time, do other duties connected with the general routine.

REPAIR SHOP DESIGN

The *Electric Railway Journal*, in 1912, made a compilation of figures from ten electric railway repair shops, showing ground and building areas in relation to number of cars owned, and the relative area of the various shop departments. Those figures and the accompanying discussion were reproduced in the first edition (1915) of the Electric Railway Handbook. The data was later taken up by the Buildings and Structures Committee of the Am. El. Ry. Eng. Assn , revised and enlarged, and, based upon it, the committee proposed, in 1922, two typical shop layouts, one for 150 and the other for 250 cars, which, the committee stated, represent the best modern practice. Much of the following data are from the report of that committee.

The following table shows the relation of total shop areas to the number of cars for which each of 15 shops were designed, as well as the two typical shop layouts proposed by the 1922 Committee.

RELATION OF TOTAL SHOP AREAS TO NUMBER OF CARS MAINTAINED

Railway	Shop area sq. ft.	Number cars	Sq. ft. per car
Montreal Tramway—Youville	112,052	770	145
Milwaukee Electric Co.—Milwaukee.	284,600	1150	247
New York State Rys.—Syracuse	65,999	214	308
Pacific Electric Co.—Torrance	296,880	1937*	153
Union Traction Co.—Anderson	79,710	294	271
United Rys. & E. Co.—Baltimore	243,400	1453	167
Birmingham Ry., L & P. Co.—Birmingham	48,508	307	158
Georgia Ry. & Pr Co—Atlanta	64,204	321	200
York Railways Co—York	20,324	79	257
Knoxville Ry & Lt. Co—Knoxville.	16,066	76	211
Louisville Railway Co.—Louisville	128,210	616	208
Virginia Ry & Pr. Co.—Norfolk	33,570	165	204
Dallas Ry. Co—Dallas	34,509	181	191
Newport News & Hampton—Hampton	20,125	103	195
Portland Ry. & Lt Co.—Portland	142,950	1032	139
Average, 15 shops			203 6
A.E.R.E A. typical 150 car shop	30,532	150	204
A.E.R.E A. typical 250 car shop	50,073	250	200

* This includes a large number of freight cars.

Size of Buildings. The actual areas of the building or buildings required for housing the various departments of the repair shop of any electric railway are, of course, subject to a number of obscure factors. Probably the easiest way to arrive at a basis for comparison is to consider the areas in square feet per car served by the shop.

It is granted that this is an exceedingly rough, approximate method, but, in view of the fact that there is necessarily a certain relationship between size of shop and number of cars, the above table was prepared to show the range through which this unit varies. In determining the areas given, only buildings have been included. No yard space of any kind is considered, and covered transfer tables where they exist have been omitted.

The preceding table indicates that 200 square feet per car is a fair average floor area, with 45-ft. cars. The same result may be arrived at in a somewhat different manner by assuming that each car in the shop occupies a space 60 ft. long and 16 ft. wide. The length of 60 ft. is based on a 45-ft. car body with a 7-ft. 6-in space at each end for passageway, trucks and waste room. The width is assumed arbitrarily from the track spacing existing in many shops. This gives a total area of 960 sq ft required for each car which stands in the shop. However, the whole of the shop is not devoted to housing car bodies; the table on page 109 shows that only approximately half of the shop is used for housing cars and the other half for machinery and floor work. If each car requires 960 sq. ft. of space in which to stand, the total area of shop for each car contained in it will be twice that, or approximately 1900 sq. ft. Then, if 10 per cent of the equipment is held in the shop at any one time for repairs, accidents or rebuilding, 10 per cent of 1900 sq ft. of shop area will have to be provided for each car owned. This amounts to 190 sq. ft. of shop per car, which checks fairly well with the figures indicated in the preceding table. This is undoubtedly the most accurate method for arriving at the proper size of shop for any electric railway, as it permits allowance for the requirements of variations in length of car body as well as for unusual conditions which might necessitate frequent shoppings due to numerous accidents or excessive wear. The latter allowance may be made by using judgment in selecting the expected percentage of cars held in the shop. In crowded city streets, for instance, where accidents are common, it would not be unreasonable to assume that 12 per cent of the cars owned might be held in the shop. On the other hand, where equipment is first class and conditions good, it is safe to reduce the figure to 8 per cent. This method can be expressed by the formula:

Shop area in square feet $= 2 \times L \times S \times C \times P$

where L = length of average car plus 15 ft.

S = track spacing (generally 16 ft)

C = number of cars owned by the railway

P = maximum percentage of cars held out of service in the shop (generally 10 per cent).

Ground Areas. In the foregoing only the areas of buildings or floor areas of the various departments have been considered. It is, of course, undesirable to build a shop upon a plot of ground of the exact size of the building. There are many classes of material, such as wheels, iron castings, lumber, scrap and the like, which can be stored conveniently outside of the shop building, and this is undoubtedly better than to have them occupying valuable space under a roof, or to have them kept in a storeyard at some distance from the shop. In addition to this it is not unusual to make use of the shops for storing and repairing track material so that a need exists for a plot of ground considerably larger than the shop buildings. Unfortunately no set relation appears to exist between building and ground areas. However, the following table has been prepared to give a conception of present practice in regard to the total area of ground required. The figures have been worked out in three ways, each column showing the same area expressed in different form

AREAS OF SHOP GROUNDS

	Area of ground and buildings, acres	Per cent of building area	Area per 100 cars owned, acres	Area per car owned, sq. ft.	Remarks
Milwaukee..	9 5	144	0 63	273	
Baltimore	11 9	214	0 67	292	
Seattle	9 0	470	2 00	870	Separate buildings
Rochester	0 9	115	0 31	133	Basement area not included. No transfer table.
Anderson ..	5 5	342	1 83	800	
Minneapolis. .	19 7	510	2 50	1090	Separate buildings. Includes track material.
Syracuse	3 6	236	1 44	620	
Chicago	11 0	180	0 49	214	Basement area not included.

It will be noticed from this table that a marked difference in area of ground required exists between those shops which are composed of separate buildings and those in which all departments are grouped under one roof or are located in two or three large buildings separated only by narrow transfer tables. The former arrangement offers a much greater protection against disastrous fires, permits better natural lighting facilities and, especially for large shops, gives an opportunity for greater flexibility in design. However, it appears that the spaces between the different buildings are not needed aside from their value for fire protection, and that separating departments in different buildings necessitates a considerably greater ground area for the shops. The low figure of 0 31 acre per 100 cars owned which is shown for Rochester is due to the fact that this shop has a basement extending under the entire shop building. This in effect makes a two-story shop and, as has been mentioned before, gives the shop an unusually large storeroom. The area outside of the shop usually allowed for heavy storage has in this case been put into a basement. The same arrangement obtains in a limited degree at Syracuse, where part of the shop building is provided with a storage basement. Chicago has a layout characterized by decidedly restricted building area in consideration of the total number of cars owned, so that it is evident that the minimum ground area required for a complete set of shops is somewhere between 0 49 acre per 100 cars owned, as at Chicago, and 1 83 acres, as at Anderson. The latter shop, as shown in Fig. 39, is

RELATIVE AREA OF DEPARTMENTS EXPRESSED AS PER CENT OF WHOLE SHOP AREA

Railway	Over-haul and trucks	Machine wheels and axles	Smithy and welding	Air brake	Office	Arma-ture room	Carpt. and mill.	Paint shop	Paint mix-ture	Sand blast	Store room	Oil house	Wash room
Montreal Tramways	17.9	11.5	4.2	0.2	1.7	5.2	18.2	26.0	0.6	.	10.2	0.6	3.7
Milwaukee Electric	19.0	7.8	5.7	1.5	2.8	7.4	22.5	17.4	1.0		13.9		1.6
New York State Railways, Syracuse	30.2	6.0	1.8		2.6	2.3	21.9	19.4	0.3	0.7	11.9	1.1	1.8
Pacific Electric, Torrance	14.7	8.6	4.4	0.3	1.0	5.4	24.1	24.8	0.8	1.8	12.1	0.8	1.2
Union Traction, Anderson, Ind	21.6	4.2	4.0		1.5	4.0	25.9	15.4	3.0	. . .	15.3	3.0	2.1
United Railways and Electric, Balto	16.2	7.4	3.8		4.1	6.1	26.2	16.7	2.5	.	12.5	2.5	2.0
Birmingham Railway and Light Co	18.9	15.5	3.1		3.3	3.6	22.0	19.1	0.7	. .	8.7	2.0	3.1
Georgia Railway and Power Co	31.0	7.1	2.3	.	1.8	1.5	25.5	19.2	0.4	.	6.9	0.9	3.4
York Railways Co., Penna	34.2	7.1	2.8		2.8	2.8	19.9	14.2	2.8		9.5	1.1	2.8
Knoxville Railway and Light Co	33.2	8.2	5.2		1.6	2.5	19.6	18.1			8.1	1.0	2.5
Louisville Railway Co	20.2	7.7	2.3		0.7	2.9	29.1	22.4	0.6		12.7	0.3	1.1
Virginia Railway and Power Co	30.0	7.0	2.3		1.2	4.5	14.6	16.1			*23.8	0.5	. . . .
Dallas Railway Co	24.6	8.5	3.7		1.1	1.8	18.7	22.3			10.3	1.3	7.7
Newport News and Hampton	27.4	12.4	5.2		0.4	6.0	19.9	11.9			14.1	1.7	1.0
Portland Railway and Light Co	21.8	7.8	5.2	0.3	0.3	3.9	24.6	19.6	0.7	2.6	12.3		0.9
Averages	24.0	8.5	3.7	0.6	1.8	4.0	22.2	18.8	1.2	1.7	12.2	1.3	2.5
A.E.R.E.A. typical layouts.													
Shop for 150 cars	21.7	11.7	4.2	1.0	1.1	4.6	20.3	18.5	0.9	2.6	10.0	0.9	2.5
Shop for 250 cars	19.2	12.1	4.0	1.4	1.0	5.0	22.6	18.6	0.9	1.7	9.4	1.0	3.1

*Store room at Norfolk includes space for supplies for Lighting Department.

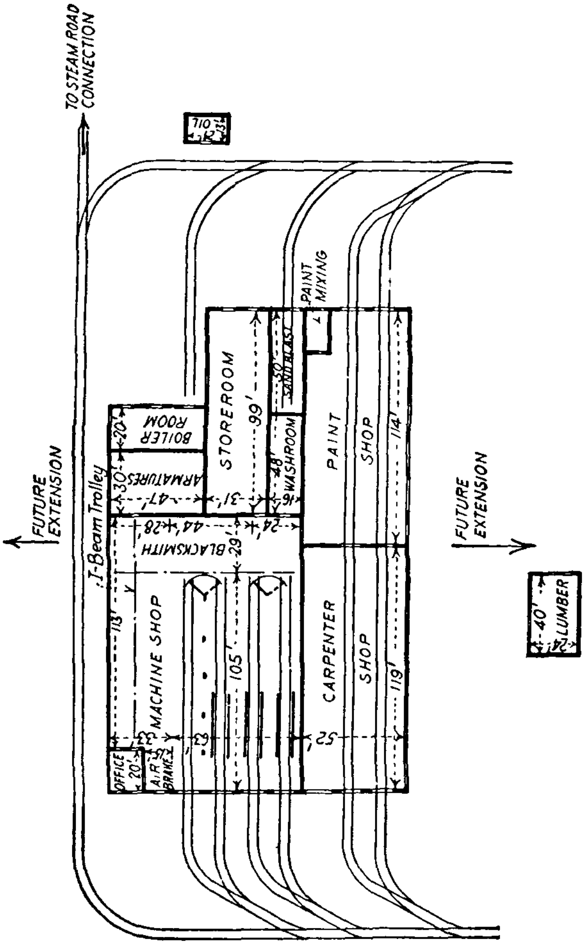


FIG. 30.—Am. El. Ry. Assn. typical shop for 150 cars.

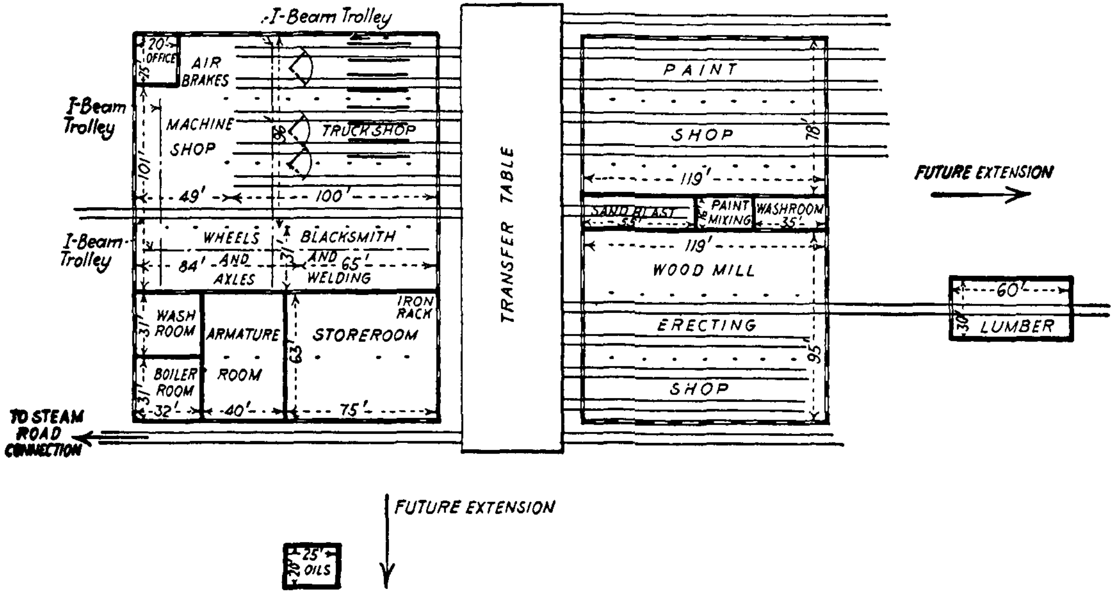


FIG. 31.—Am. El. Ry. Assn. typical shop for 250 cars.

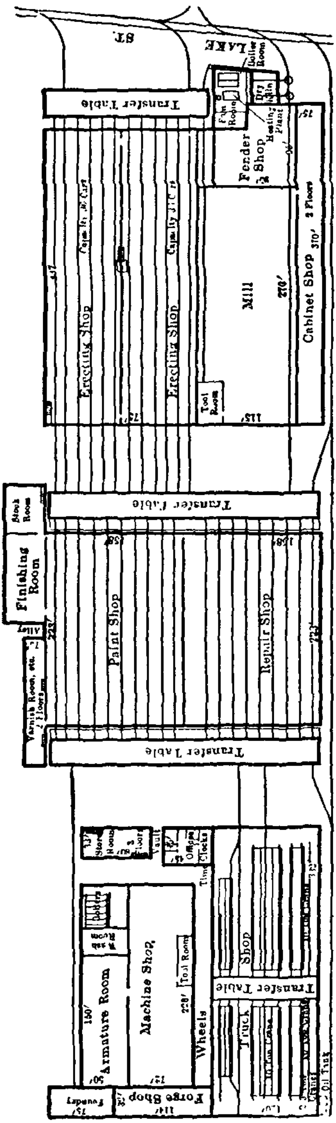


FIG. 32.—Chicago shop arrangement.

located on a triangular plot of ground which is at best difficult to utilize. In round numbers, it is probable that 1 acre per 100 cars owned is sufficient, or at least desirable, for a compactly designed shop. When separate buildings are used for the different departments, and especially where considerable quantities of track material are stored, 2 acres per 100 cars owned appears to be the least ground area which should be provided.

Examinations of the shop arrangements shown in Figs. 32 to 41 disclose the somewhat unexpected condition that only a minority of the ten contain distinct provision for future extension. It is, of course, possible to extend any shop provided that sufficient ground area is provided around it, but in most of the layouts shown an extension could only be made by shifting departments or adding to the sides instead of the length of the buildings.

Relative Area of Shop Departments. The 15 shops studied by the Am. El. Ry. Eng. Assn. 1922 Committee on Buildings and Structures were used in the compilation of the table shown on page 109, where the relative floor area occupied by the various departments is shown, expressed as percentages of the whole shop area. It will be noted that with few exceptions the relative areas of the various shops agree reasonably well with the average, and with those of the typical shops proposed by the Committee, the plans of which are shown in Figs. 30 and 31.

Miscellaneous Considerations in Shop Design. Experience has shown the necessity for proper clearance between cars and posts, walls and other fixed parts of buildings. The minimum clearance should be 2 feet; on curves it is well to increase this to provide for increase in the size of cars.

The distance between track centers has generally been inadequate and the present tendency is strongly in favor of wider spacing. Sixteen feet is a desirable spacing, and it should not be less than 15 feet in any event. This, of course, does not apply to inspection pits in car houses which matter is covered on page 100.

For smaller shops undoubtedly ladder tracks are to be preferred to transfer tables. For a moderate sized shop, local conditions often dictate which shall be used. For a large shop, a combination of the two is often desirable, as it gives additional flexibility and speed in hand-

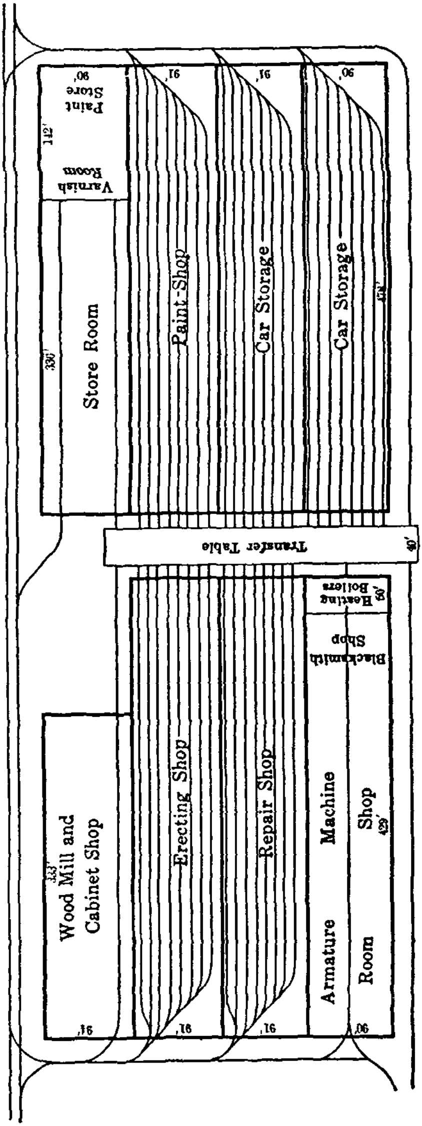


FIG. 33.—Baltimore shop arrangement.

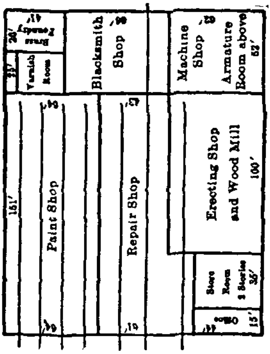


FIG. 35.—Portland shop arrangement.

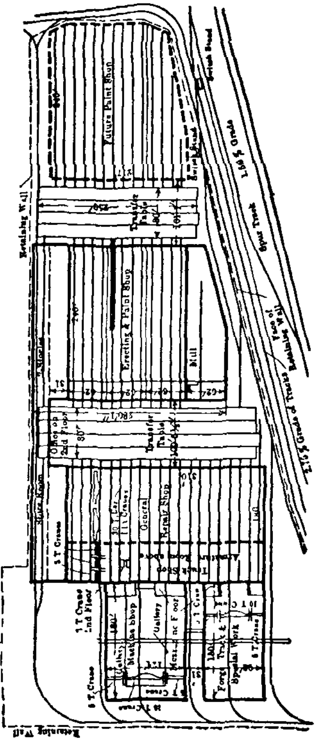


FIG. 34.—Milwaukee shop arrangement.

ling cars through the shops. For a small number of tracks, the first cost of a ladder track will be lower, while for a large number of tracks the transfer table will cost less. In designing a new shop layout, it will be well for the engineer to make up detailed estimates covering both and then make a study of the advantages of each to ascertain which will be best adapted to his particular case. The advantages of ladder tracks are: (*a*) Cars can be more readily removed from shop in case of fire; (*b*) do not have to depend on one piece of apparatus to handle cars; (*c*) cost of operation is less; (*d*) permits trucking over it; (*e*) lower maintenance for small number of tracks. The advantages of transfer tables are: (*a*) Flexibility; (*b*) less space required; (*c*) lower maintenance for large number of tracks; (*d*) less first cost for large number of tracks.

Two types of transfer tables are now in service, namely, pit type and flush type. A questionnaire sent out to representative companies indicates that the pit type is less expensive and more desirable.

Some companies now using transfer tables have had trouble with them due to using old discarded motors to drive them and on account of improper design or construction. If the transfer table is to be used, it should be carefully designed and built rugged enough to stand hard service. Whether the transfer tables should be in the open or under cover depends on climatic conditions. In cold climates where there is much snow and ice it is well to have it under cover. Where pit type tables are used, suitable drainage should be *provided.*

It is important to make use of a number of labor saving devices, and in the design of the buildings this should be taken into consideration. Prominent among them are wheel drop pit and pit jacks, jib crane with electric hoist for truck repairs, overhead trolley and travelers, hood for blowing out motors, pit wheel grinder, truck transfer tables, transveyor or portable crane, oxy-acetylene cutting and welding outfit, arc welder and pneumatic tools.

Individual drive of shop machinery has more advantages over belt drive, as far as building construction and natural lighting is concerned. Especially is building construction affected when the machine tools are of heavy type which means a rigid support for lineshafting.

It is desirable to locate the shops so as to reduce non-revenue mileage and to obtain a steam road connection. While the latter is not absolutely essential, it is a great convenience in unloading car load lot materials.

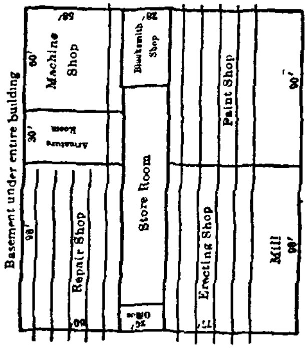


Fig. 37.—Rochester shop arrangement.

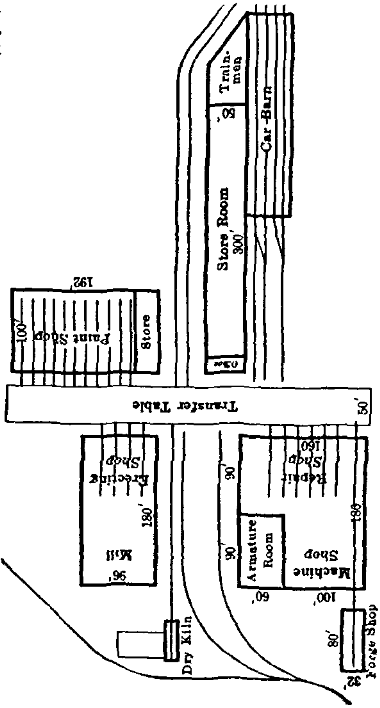


Fig. 36.—Seattle shop arrangement.

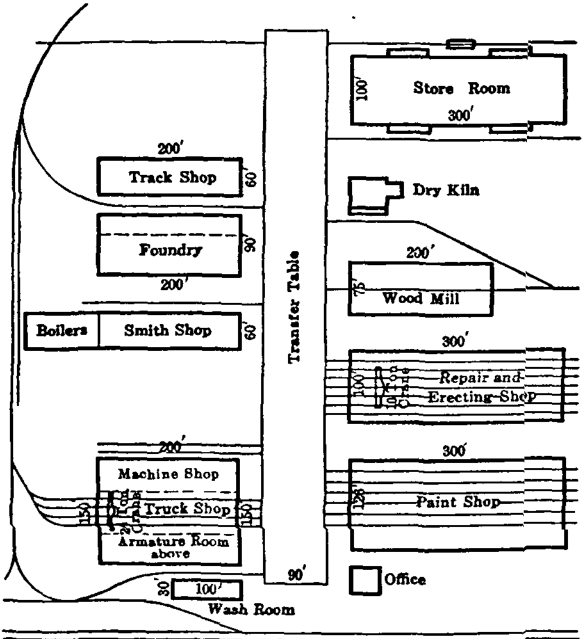


FIG. 38.—Minneapolis shop arrangement.

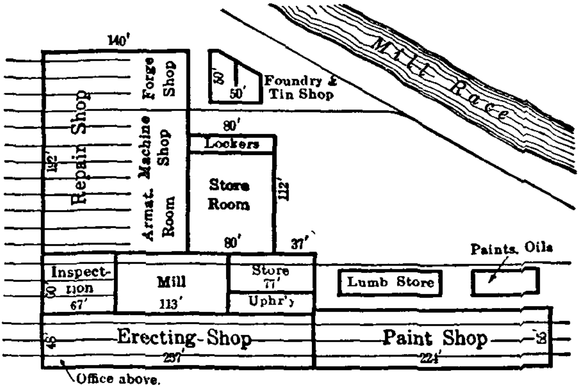


FIG. 39.—Anderson shop arrangement.

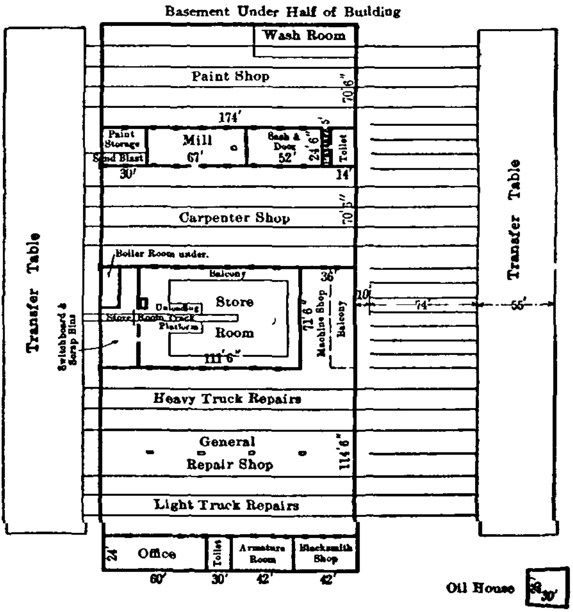


FIG. 40.—Syracuse shop arrangement.

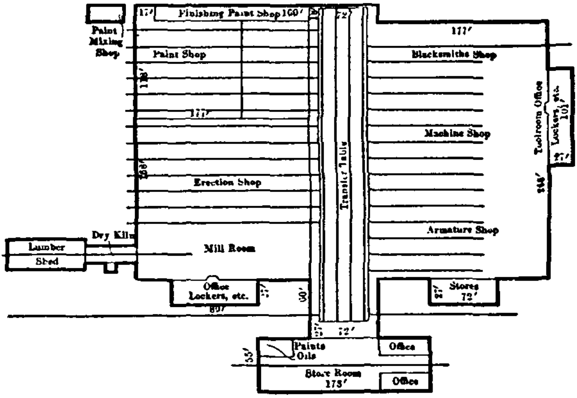


FIG. 41.—Montreal shop arrangement.

Of the methods of raising car bodies off trucks, electric hoists of the screw type appear to be preferred as being the best adapted for the purpose. The truck overhaul shop of the Am. El. Ry. Eng. Assn. typical layout is accordingly so equipped. For large shops electric traveling cranes can be used to advantage but for a moderate size shop, the advantages gained are more than offset by the additional cost of the apparatus and building details.

In laying out the yard track arrangement, some provision should be made so that cars may be readily turned.

SECTION III

TRAIN MOVEMENT

Schedules. In the preliminary determination of schedules for an electric railway the factors concerned bear directly on the amount of travel between various points and the time intervals during which such movements exist.

Where S = schedule speed in miles per hour (including time for stops)
H = headway in minutes
D = length of line in miles
N = number of cars if single cars be used, or train units if more than one car be used
T = time in minutes occupied in running between terminals during one single trip

then $S = \frac{60\,D}{T}$ $\quad N = \frac{120\,D}{SH}$

$H = \frac{120\,D}{SN}$ $\quad S = \frac{120\,D}{HN}$

Chart for Headway Calculations. If the schedule speed and any one of the following factors—headway in seconds, headway in feet, cars per mile—are given, the other factors may be obtained quickly by reference to the chart developed by H. M. Wheeler

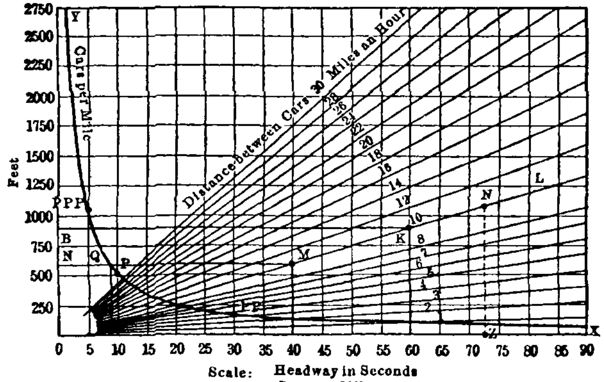


FIG. 1.—Headway chart.

(Fig. 1). The curved line marked "cars per mile" is a hyperbola and shows the relation between cars per mile, read on the horizontal scale, and feet as shown on the vertical scale. The diagonal lines converging at the left and marked "distance between cars" show for various headways in seconds (at the bottom of the chart), the spacing of cars in feet, by the use of the diagonal line corresponding to the proper schedule speed.

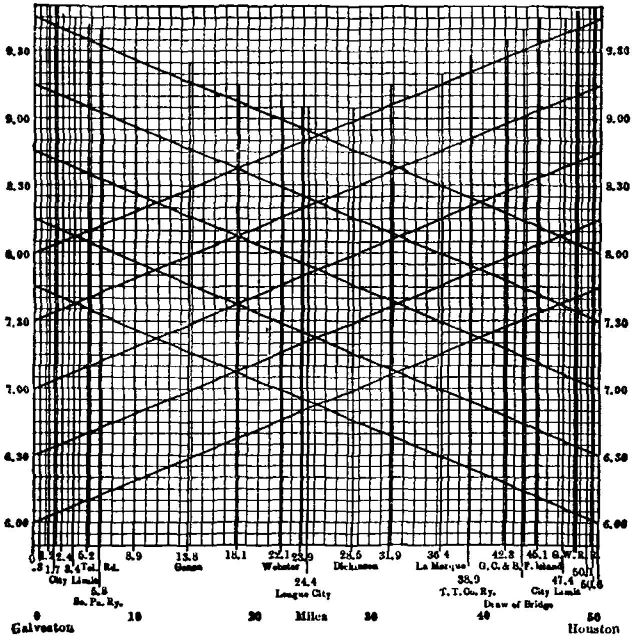


FIG. 2.—Preliminary interurban train schedule.

Graphical Train Schedules or Train Sheets. Many factors entering into the proper construction and successful operation of railways are at once apparent from the graphical train schedule which is plotted with time of day in hours and minutes against distances in feet or miles. It is convenient if the coordinate drawn through each hour division be made a heavy line on the coordinate paper and if these sections be subdivided into sixths or twelfths, representing 10- or 5-minute intervals, respectively. It is customary to designate the distance between stations in feet or miles, and the location of any points of special engineering interest, such as branch lines, railway crossings, city and township limits, etc. Sloping lines may then be drawn to represent the progress of a train from station to station, as shown in Fig. 2, reproduced from Harding's "Electric Railway Engineering." The inclination between these lines and the time axis depends upon and represents graphically the schedule speed of the train. A chart made up of such straight lines as shown in Fig. 2 is sufficiently accurate for a rough preliminary study of traffic possibilities, power requirements and substation locations, but before exact time tables can be adjusted and meeting points determined, a very much more accurate and detailed

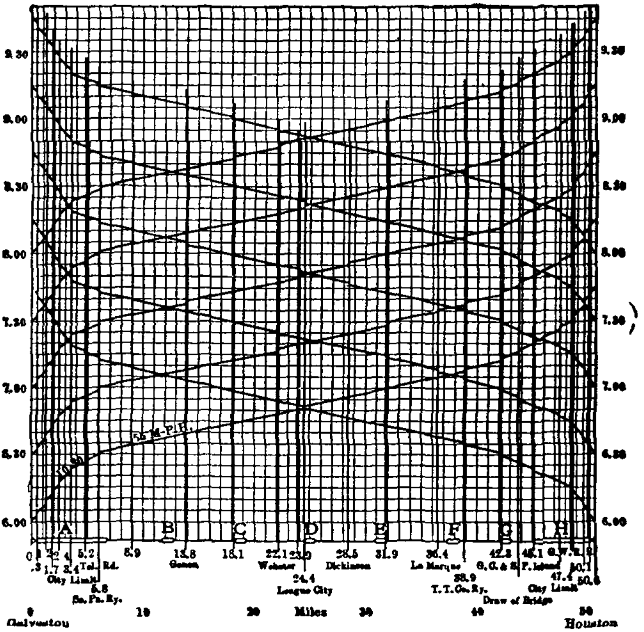


FIG. 3.—Final interurban train schedule.

graphical train schedule must be drawn. Such a schedule involving several different schedule speeds over various sections of road as well as a representation of the time elapsed in making station stops is shown in Fig. 3. Such a graphical schedule enables the engineer to predetermine not only the number of cars necessary to maintain a given schedule and the position of all such cars at any moment, but it also locates the meeting points at the crossing of the schedule lines, and when used in conjunction with the power curves of the various cars, as described later, it aids in locating substations and determining the average and maximum loads on substations and power stations. Another illustration of the use of train sheets is

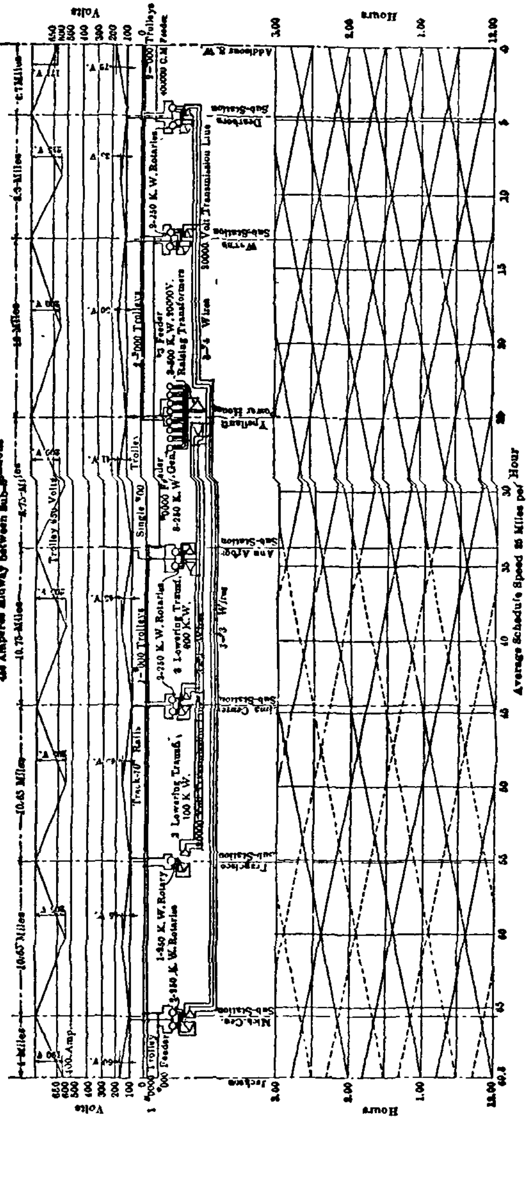


Fig. 4.—Train sheet and diagram of electrical distribution—Detroit, Ypsilanti, Ann Arbor & Jackson Railway.

shown in Fig. 4 drawn up by W. Nelson Smith. In this case some of the details of the electrical distribution system are also shown. The particular schedules which have been illustrated are relatively simple ones. With the addition of express and local service and in some cases freight and mail trains, with the necessity of meeting the schedules of trunk and branch lines, a graphical schedule may become rather complicated, but by the use of a large scale drawing such solutions are made with little difficulty.

Frequency of Stops. The following table shows the usual frequency of stops as encountered in various classes of railway service:

Steam locomotive through service	1 stop in 100 miles
Steam locomotive local service	1 stop in 20 miles
Steam locomotive suburban service	1 stop per mile
Electric interurban express	1 stop in 10 miles
Electric interurban local	1 stop in 2 miles
Electric suburban	1 to 2 stops per mile
City elevated or rapid transit	2 to 3 stops per mile
City surface lines	5 to 10 stops per mile.

The location of stopping places in city streets is often determined by the location of intersecting streets. Due to the important effect of frequency of stops, both on schedule speed and power costs, efforts should be made to spread stopping places as far as is consistent with good service. It is believed that this requirement will be met by a minimum spacing of 500 feet in congested business districts, 600 feet in closely built residence districts, and up to 1200 feet in suburban territory, depending upon the density of population. On a street railway serving a number of cities and thickly settled suburban communities in eastern Massachusetts, a 27 per cent reduction in stopping places resulted in a 12 per cent reduction in the actual number of passenger stops made by cars.

Duration of stops varies approximately as follows for different classes of railway service:

Through trains, steam	5 minutes
Local trains, steam	2 minutes
Interurban cars, electric	10 to 30 seconds
City rapid transit trains, electric	10 to 20 seconds
City surface cars, electric	5 to 12 seconds.

The rate at which passengers board and leave cars, together with other minor conditions, practically determines the length of service stops in street railway practice. The design of car and method of fare collection is largely responsible for the rate of passenger movement, as is illustrated by Fig. 5. These curves show average stop times as related to the number of passengers boarding or alighting (in whichever direction the maximum movement occurred) in different types of cars, and were made up by averaging a large number of observations made in a number of cities.

Effect of Stops. The effects of frequency and duration of stops are severe, both on schedule speed and energy consumption. The general relation between maximum and schedule speed, as effected by frequency of stops, is shown by Fig. 6 (Standard Handbook). Figs. 7 to 10, inclusive, show more detailed information as applicable to a definite equipment of a suburban type car with four GE-203 motors, 30-in. wheels, 67/17 gears, weight with passengers 47,000 lb. The free running speed is about 31 m.p.h., line voltage average 500, and 200 feet of coasting is allowed before application

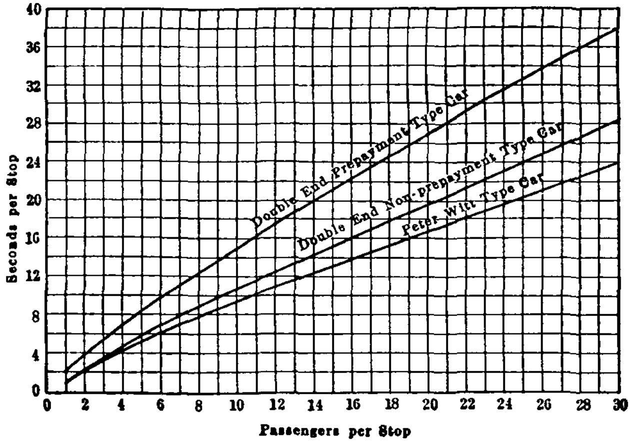


FIG. 5.—Comparative passenger stop times.

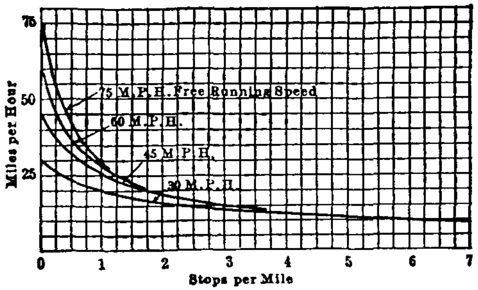


FIG. 6.—Relation between maximum and scheduled speed and stops per mile.

of brakes. Fig. 7 shows the effect of frequency and duration of stops on schedule speed. Fig. 8 shows the additional time required by one additional stop with various rates of acceleration and braking, or conversely, the time which might be saved by the elimination of one stop under such conditions. Fig. 9 shows the effect of frequency of stops on total energy requirements, while Fig. 10 shows the additional energy required by the addition of one stop, or conversely, the energy which might be saved by the elimination of one stop.

Speed Limitations. Maximum speed may be limited for a given equipment by the safe maximum armature speed (see tables, pp. 254 to 259). There also may be limitations applicable to specific locations, such as those due to grade crossings or other public high-

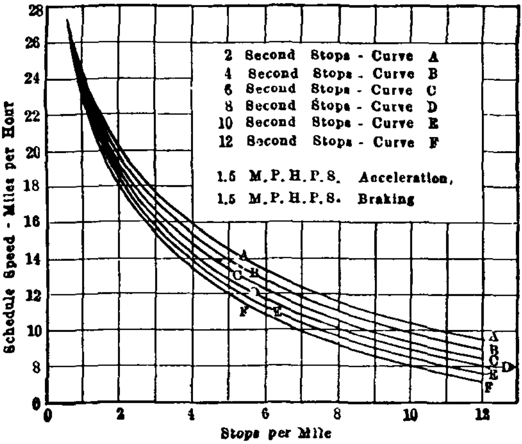


FIG. 7.—Effect of frequency and duration of stops on schedule speed.

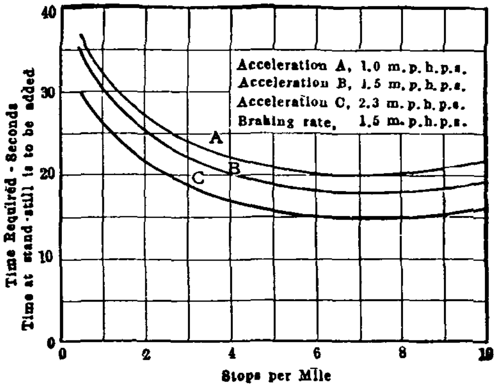


FIG. 8.—Additional time required for one additional stop, under various conditions.

way interference, those due to track special work, and those due to track curves. For track curves with proper superelevation of the outer rail, the safe maximum speed in miles per hour is usually considered to be equal to the square root of the radius of the curve in feet.

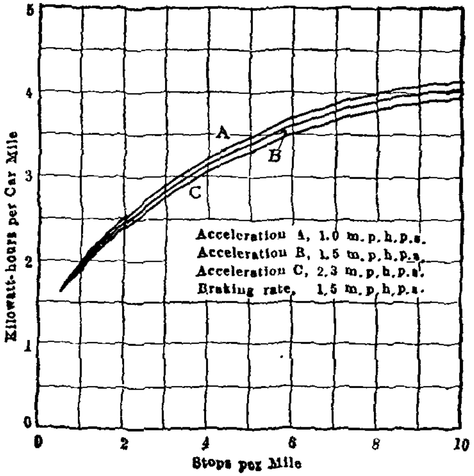


Fig. 9.—Effect of frequency of stops and rates of acceleration on energy per car mile.

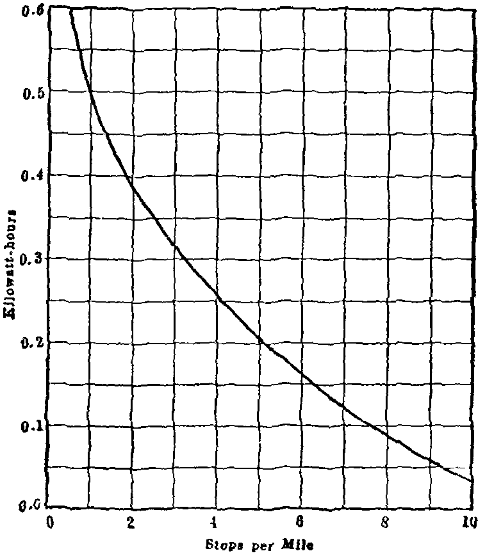


Fig. 10.—Additional energy required by one additional stop.

Train Resistance

Train resistance may be defined as the resultant of the forces, exclusive of those which are evidenced by internal losses in motor equipment, which oppose the motion of the train at a constant speed on a tangent level track in still air. For convenience train resistance at any speed is expressed as the number of pounds tractive effort at the driving wheel treads necessary to keep the train moving at that constant speed on tangent level track in still air. From the results of tests by many investigators most of the many train resistance formulas have been approximated, but while many of these formulas have been shown to give values nearly equal to those secured by test on certain equipment and track and under certain conditions, it is not safe to depend on any one formula to give very close approximations in universal application. This is because there are so many conditions affecting the result. The variation of some factors, such as temperature change in the bearings, may be comparatively slow, while that of others, such as of air resistance, may be great from moment to moment during the operation of the train. One of the points of considerable disagreement among the formulas for train resistance is the manner in which train resistance varies with speed. At low speeds this is of but little importance compared with other factors, but at high speeds it becomes one of the most important factors in determining the speed for economic operation.

The more recent train resistance formulas recognize three principal ordinary components which are conveniently referred to as (1) journal friction, (2) rolling friction, and (3) air resistance.

Journal Friction. Journal friction of the car wheel axles has been found to depend upon the weight and speed of the train, and as the weight of the train increases the journal friction becomes of less importance per unit weight of train until it reaches a lower limiting value of about 3.5 lb. per ton weight of train. Lower values of journal friction have been secured in tests. The 1904 Proceedings of the Am. Ry. Eng. and Maint. of Way Assn. states that a coal car with 40 to 50 tons of coal will not have a journal friction of more than 2.5 lb. per ton, while the same car empty will have a journal friction of about 5 lb. per ton. As noted on page 143, the condition of the lubricant has a considerable effect on journal friction.

Rolling Friction. Rolling friction, including rail friction, increases with the speed of the train and depends upon the diameter of the wheels, general design of the truck and the condition of wheels and track. It is greater as the wheel and rail surfaces are more imperfect and increases with track irregularity and flexibility. Track irregularity and flexibility are also important in setting up oscillations and concussions, the damping of which consumes energy. In this process flange friction is also increased. All of these effects are so closely related that for ordinary work it is impracticable to separate them.

Air Resistance. Tests to determine the value of air resistance have yielded the most widely differing results. Its value depends upon the speed of the train, the area of cross-section and lateral area of the locomotive and each car, the shapes of the front and rear ends of the train, and the number of cars composing the train.

Effect of Individual Car Weight on Train Resistance. Train resistance has been found to depend upon the weights of individual cars making up the train, that is to say, given two trains of the same total weight but *made up of a different number of cars,* the train resistance will be different for the two. Tests indicate that the train of heavier cars will have the lesser train resistance, and that this difference is not entirely due to different air resistance. (See Journal Friction, above, also Fig. 24 and Train Resistance of Freight Train, p. 136.)

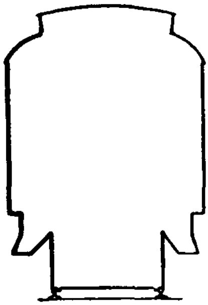


FIG. 11.—Outline of projected area considered in air resistance calculation.

Train Resistance Formulas. Two formulas for train resistance are in common use as applicable to the operation of single cars and to trailer and multiple unit trains, as well as to the usual locomotive and passenger car train, these being known as the Armstrong and the Blood Formulas. Another formula, as developed by the University of Illinois, is considered more nearly correct for application to freight trains. No such definite formula has been developed for train resistance in tunnels or subways, although several tests have been made, the results of which vary widely and are applicable only to conditions similar to those of the tests.

Armstrong Formula. This formula is based on one of similar form proposed by W. J. Davis after a series of tests on the Buffalo & Lockport Railway, and modified by A. H. Armstrong following a number of later tests, including the Zossen tests in 1902–3, Electric Railway Test Commission tests in 1904–5, tests at Schenectady in 1905–6, and by the University of Illinois in 1910, 1914 and 1917. It is suitable for application to single cars and to trailer, multiple-unit or locomotive trains up to the usual limits of passenger train operation. The formula for single cars is

$$f = b + cS + \frac{daS^2}{W}$$

where f = train resistance, pounds per ton weight of car

a = area of cross-section of car, square feet. This is the area included within the outline of the car body and a rectangle of a width equal to the average width of the trucks and with its bottom line at the heads of the rails. (See Fig. 11)

b = constant component of journal friction

$= \frac{50}{\sqrt{W}}$ but limited to a minimum value of 3.5

c = coefficient of variable component of journal friction and rolling friction
= 0.03 for ordinary conditions and car weight; higher values for cars weighing less than 30 tons, and for poor track conditions; maximum, 0.07
d = air resistance coefficient
= 0.0015 for cars with pointed ends of the extreme type
= 0.002 to 0.0025 for usual rounded end suburban car or electric locomotive with rounded ends or sloping front
= 0.004 for cars with perfectly flat ends
S = speed, miles per hour
W = weight of car, tons (of 2000 lb.)

The formula for single cars under usual conditions is therefore

$$f = \frac{50}{\sqrt{W}} + 0.03\,S + \frac{0.002\,aS^2}{W}$$

It is assumed that air resistance is increased one-tenth by the addition of each succeeding car in a train. **For train operation,** therefore, add one-tenth to the air resistance term of the formula $\left(\frac{daS^2}{W}\right)$ for each car after the first, provided all cars are of similar cross-section. If cars vary considerably in cross-section, the total air resistance term should be varied accordingly.

For application of the Armstrong formula to cars of common weight and trains of various numbers of cars, see Figs. 13, 15, 17, 19 and 21. In these charts the values assumed for the cross-sectional areas correspond closely to actual practice, and are as follows:

Weight of car, tons	Area, sq. ft.
10	75
15	83
20	88
25	93
30	100
40	113
50	120
60	120

Blood Formula. *The train resistance formula proposed by* John Balch Blood in his paper before the American Society of Mechanical Engineers, in 1903, is as follows:

$$f = E + BS + \left(C + \frac{D}{W}\right)S^{1.8}$$

where f = train resistance, pounds per ton
E = *coefficient of sliding friction*
= 7 for light electric cars
= 6 for medium weight electric cars
= 5 for heavy electric cars
= 4 for average passenger trains
= 3 for heavy freight trains

B = coefficient of rolling friction
= 0.15 for light track construction
= 0.12 for heavy track construction
C = coefficient of side resistance
= 0.0016 for ordinarily constructed cars
= 0.0014 for cars with vestibules
D = coefficient of head and stern resistances
= 0.25 for cars of small cross-section
= 0.30 for electric cars of medium cross-section
= 0.35 for large electric or suburban trains
= 0.40 for largest express trains
S = speed, miles per hour
W = weight of car, tons (of 2000 lb.)

In the use of the Blood formula, it is often assumed that the coefficients E, B, C and D vary directly with car weights; such coefficients as used by many engineers are shown by Fig. 12.

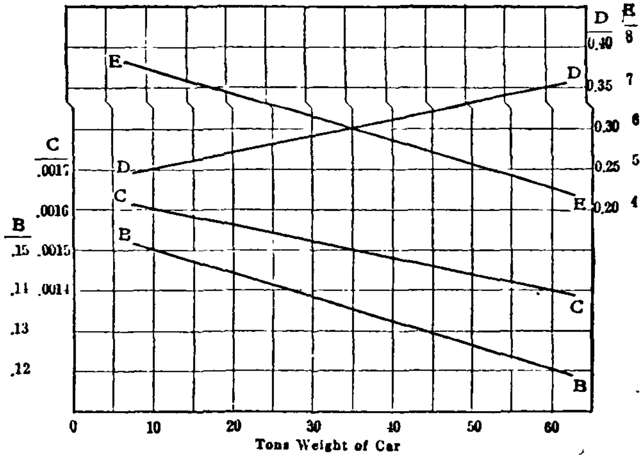


FIG. 12.—Coefficients for Blood train resistance formula.

For **train operation,** the formula as given above should be used for the leading car or locomotive, while for succeeding or trail cars the coefficient D, representing head and stern resistances, should equal zero. The total train resistance, in pounds per total weight of train, is determined by applying the formula to each class (or weight) of cars in turn, taking such proportion of each result as the weight of such car or cars bears to that of the entire train, and adding these quantities. For instance, to find the train resistance of a train made up of two 50-ton motor cars with one 30-ton trailer coupled between: total train weight = 130 tons; train resistance

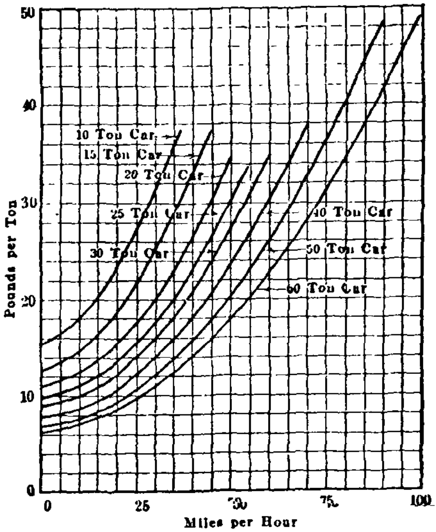


FIG. 13.—Train resistance, single car (Armstrong formula).

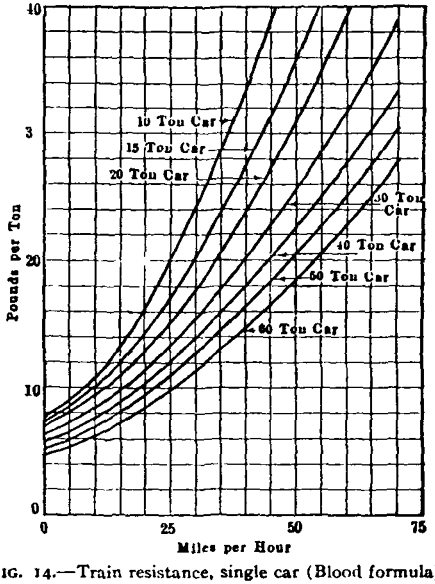


FIG. 14.—Train resistance, single car (Blood formula).

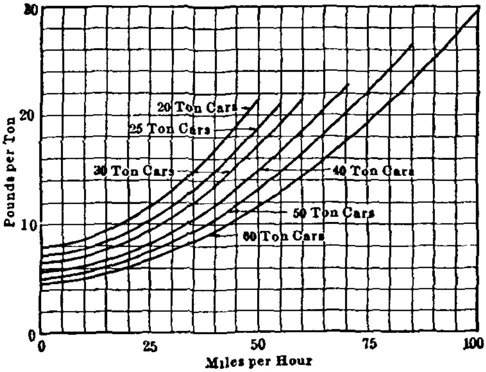


FIG. 15.—Train resistance, two-car train (Armstrong formula).

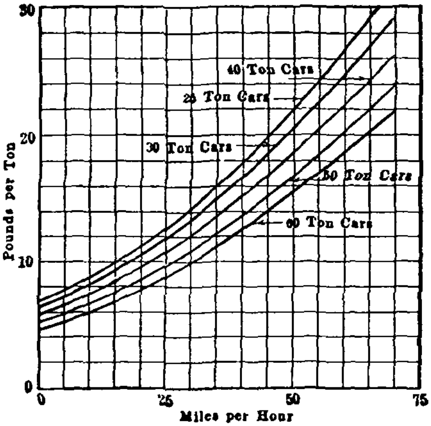


FIG. 16.—Train resistance, two-car train (Blood formula).

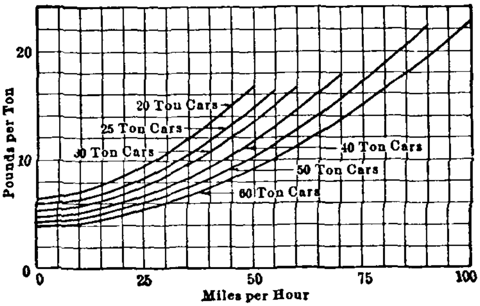


FIG. 17.—Train resistance, three-car train (Armstrong formula).

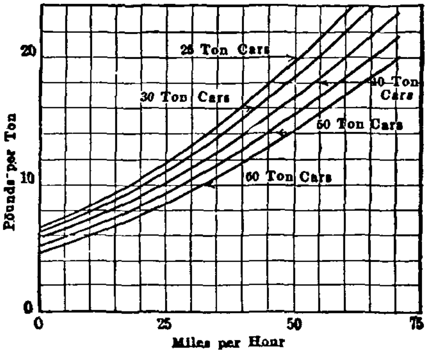


FIG. 18.—Train resistance, three-car train (Blood formula).

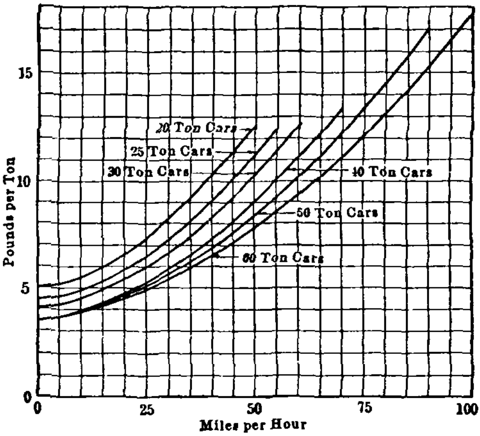


FIG. 19.—Train resistance, five-car train (Armstrong formula).

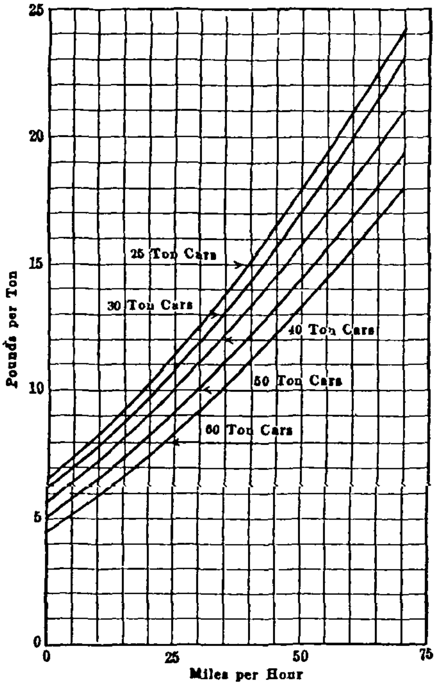


FIG. 20.—Train resistance, five-car train (Blood formula).

at any speed $= \frac{50}{130}$ of train resistance at that speed for leading car from 50-ton car (Fig. 14) $+ \frac{30}{130}$ of that for 30-ton trail car (Fig. 22) $+ \frac{50}{130}$ of that for 50-ton trail car (Fig. 22).

For application of the Blood formula to motor cars and trailers of various weights, and to trains of various numbers of cars, see Figs. 14, 16, 18, 20 and 22. The coefficients as shown by Fig. 12 have been used in the construction of these charts.

Train Resistance in Tunnels and Subways. The air friction element of train resistance may be very materially increased when the car or train is traveling in a tube, the extent of such increase probably depending upon the relative cross-sectional areas of car and tube, and the facilities for relief of differences in air pressure due to the motion of the train. One authority has suggested that

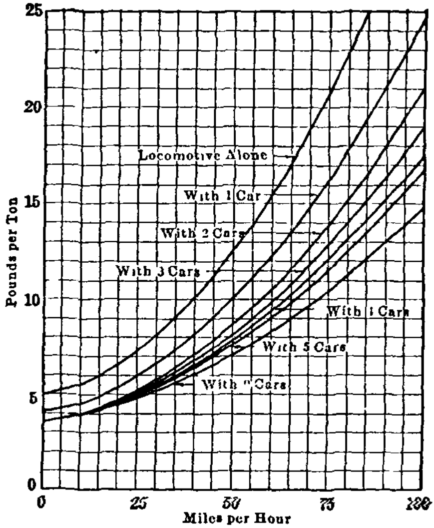


FIG. 21.—Train resistance, locomotive and train (Armstrong formula).

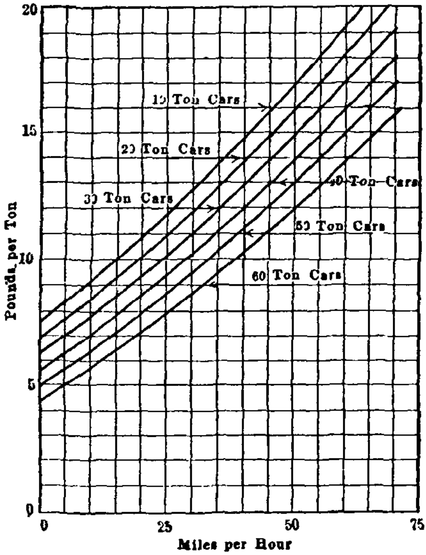


FIG. 22.—Train resistance, trailing cars (Blood formula)

open air train resistance values should be increased 25 per cent to meet subway or tunnel conditions, but no such general rule can be applied with assurance to the many differing designs of subways and tunnels, some examples of which are shown in Figs. 46 to 61,

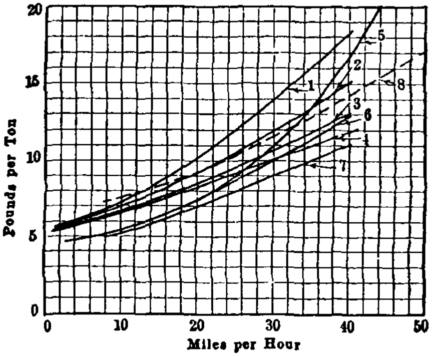


FIG. 23.—Train resistance in subways and tunnels.

1. Interborough Rapid Transit test—3-car train—110 tons
2. Interborough Rapid Transit test—5-car train—180 tons
3. Interborough Rapid Transit test—8-car train—300 tons
4. Interborough Rapid Transit—10-car train—estimated from above
5. Boston—Cambridge subway test—3-car train—165 tons
6. Hudson-Manhattan tests in 4000 ft. tube—3 ton train—104 tons
7. Hudson-Manhattan tests in 2000 ft. tube—3 ton train—104 tons
8. Above tests modified to meet Brooklyn Rapid Transit subway and tunnel conditions

inclusive, Section I, pages 72 to 75, inclusive. In the absence of any general formula such as those available for open air conditions, recourse must be had to the results of tests under conditions as nearly as possible similar to those in hand, whenever such tests have been or can be made. Fig. 23 shows the results of several investigations on this subject.

Train Resistance of Freight Train. Due to the relatively small cross-section of cars in proportion to length of train, the total air resistance of freight trains is made up of proportionately more side friction and less end effect than that of passenger trains. The lower speeds further reduce the relative effect of air resistance, so that bearing and rolling friction enter into freight train resistance in a larger measure than into that of passenger trains. For such reasons, the usual train resistance formulas are rarely applied to long and slow speed freight trains. Fig. 24 (from Bulletin No. 43 of the Univ. of Ill. Eng. Exp. Sta.) shows the results of a series of dynamometer car tests on freight trains, expressed as train resistance in pounds per ton. These are the most commonly used and generally accepted as most reliable data on train resistance of freight trains, but it must be remembered that as the tests were made by a dynamometer car, the train resistance curves and formulas deduced therefrom *do not include the head on portion of air resistance.* The

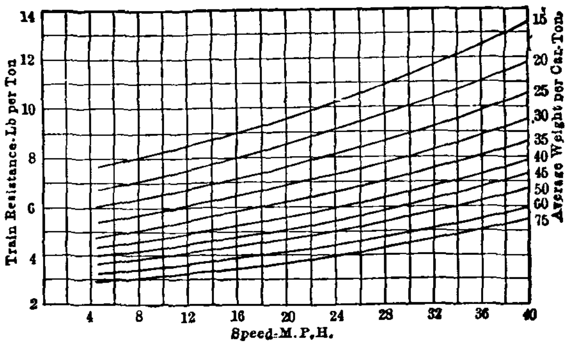


FIG. 24.—Train resistance of freight trains.

following formulas are empirical, yielding values whose maximum variation from those given by Fig. 24 is 0.5 per cent when used within the speed limits of Fig. 24.

FREIGHT TRAIN RESISTANCE FORMULAS

When $W = 15$ tons; $f = 7.15 + 0.085\,S + 0.00175\,S^2$
$W = 20$ tons; $f = 6.30 + 0.087\,S + 0.00126\,S^2$
$W = 25$ tons; $f = 5.60 + 0.077\,S + 0.00116\,S^2$

$W = 30$ tons; $f = 5.02 + 0.066\,S + 0.00116\,S^2$
$W = 35$ tons; $f = 4.49 + 0.060\,S + 0.00108\,S^2$
$W = 40$ tons; $f = 4.15 + 0.041\,S + 0.00134\,S^2$

$W = 45$ tons; $f = 3.82 + 0.031\,S + 0.00140\,S^2$
$W = 50$ tons; $f = 3.56 + 0.024\,S + 0.00140\,S^2$
$W = 55$ tons; $f = 3.38 + 0.016\,S + 0.00142\,S^2$

$W = 60$ tons; $f = 3.19 + 0.016\,S + 0.00132\,S^2$
$W = 65$ tons; $f = 3.06 + 0.014\,S + 0.00130\,S^2$
$W = 70$ tons; $f = 2.92 + 0.021\,S + 0.00111\,S^2$
$W = 75$ tons; $f = 2.87 + 0.019\,S + 0.00113\,S^2$

The following formula gives an approximation to the values of train resistance given by Fig. 24 (the maximum difference being 9.5 per cent which occurs at $S = 21$ and $W = 55$).

$$f = \frac{S + 39.6 - 0.031W}{4.08 + 0.152W}$$

in which f = train resistance, pounds per ton
W = weight of car, tons
S = speed of train, miles per hour.

These curves (Fig. 24) may be applied to predict the probable total train resistance of entire freight trains which are either homogeneous or mixed as regards individual car weights and which have been in motion for some time, when the air temperature is above 30 deg. F. and the velocity of the wind is not more than 20 miles per hour. Due to variation in make-up or external conditions, some trains may have a train resistance about 9 per cent in excess of that given by Fig. 24, but this is of importance only in rating the motive power for speeds under 15 miles per hour.

The above results were obtained from tests of 32 ordinary freight trains in *regular service of such make-up as naturally* resulted from the traffic conditions in the Champaign yards of the Illinois Central Railroad. The chief characteristics of these trains were as follows:

	Minimum	Maximum
Total weight of train, tons	747	2908
Average weight of cars composing the train, tons	16.12	69.92
Number of cars in the train	26	89
Train length, feet	1120	3480

The trains whose average car weights were less than 20 tons or more than 60 tons were composed of cars of nearly uniform weight; while those whose average car weights were between 20 and 60 tons were either homogeneous or mixed as regards the weights of the individual cars. Presumably, the majority of the cars had journals conforming to the specifications of the Master Car Builders' Association which for some years have required that the size of freight car journals be either 3¾ by 7 in., 4¼ by 8 in., 5 by 9 in., or 5½ by 10 in., depending upon the car capacity. All the cars had four wheel trucks and it is safe to assume that all the car wheels were 33 in. in diameter.

The track on which these tests were made is on the Chicago division of the main line of the Illinois Central Railroad. It extends from Gilman to Mattoon, Ill., a distance of 91 miles. The maximum grade against north-bound traffic was 29 ft. per mile, and against south-bound traffic, 31.9 ft. per mile, and in the 91 miles there was 7850 ft. of curved track. The track was well constructed and well maintained, and probably was such as one might expect to find on main lines of first class railroads. About 94 per cent of the track was of 85-lb. A.S.C.E. section rails laid in about 1900, and the remainder was of 75-lb. A.S.C.E. section rails laid in 1894 and 1895. It was laid on oak ties spaced 20 in. center to center. About 83 of the 91 miles were ballasted with broken limestone and the rest, which was in station grounds, was ballasted with screenings or cinders. None of the data used in the construction of the curves was taken before the train had been in motion at least 10 miles. The speed during the tests ranged from 5 to 35 miles per hour, the air temperature from 34 deg. F. to 82 deg. F. and the approximate average wind velocity during all but one test was less than 20 miles per hour. The direction of the wind relative to that of the track varied through 360 deg. during the tests. Each train resistance was reduced to train resistance on level track by correcting for grade. The tests were made by means of a dynamometer car and only take into consideration the train resistance of the part of the train behind the locomotive tender.

Relation Between Vestibule Shapes and Train Resistance. Fig. 25 gives approximate speed air resistance curves for a single

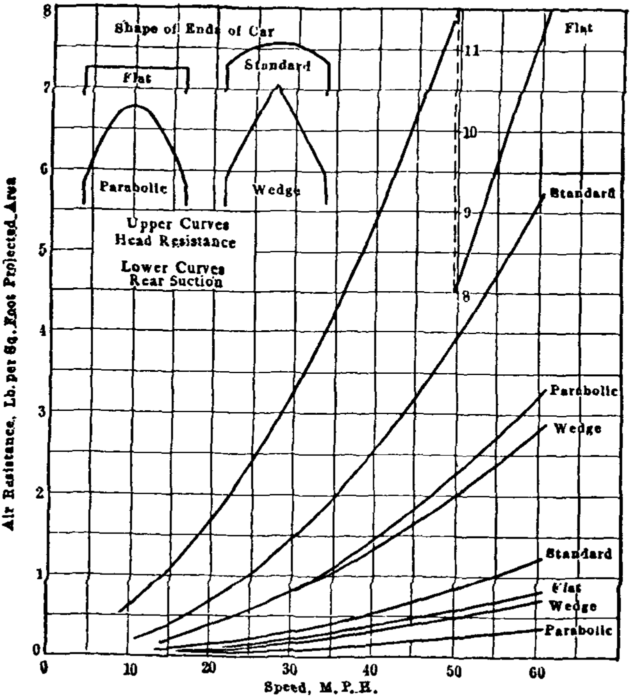


FIG. 25.—Air resistance with various shapes of vestibules.

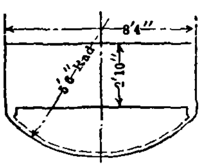


FIG. 26.—Standard vestibule.

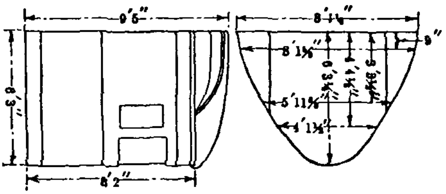


FIG. 27.—Parabolic vestibule.

interurban car when tested with vestibules of various shapes at speeds up to 60 miles per hour. These curves are from data given *in the report of the Electric Railway Test Commission, 1904*, and show the end air resistance which may be expected under ordinary conditions of service on a level track, especially the variation in resistance due to various shapes of car ends. The tests on which these curves are based were performed with a specially constructed car which was operated on a tangent track of 80-lb. T-rails laid on gravel-ballasted oak ties. The general method of making the tests was to make determinations of speed, electromotive force applied to the car and current taken by the car while running at practically constant speed. From the average values of voltage and current and the current efficiency curve for the motor equipment, the power required to balance train resistance at that speed was determined, and by means of dynamometers the end resistances were segregated.

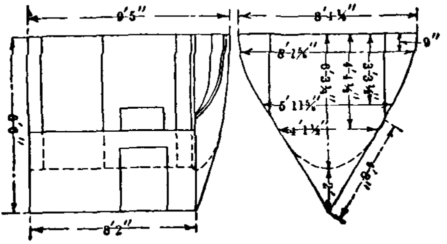


FIG 28.—Parabolic wedge vestibule.

The car body was of an interurban type 32 ft. long without vestibules, the foundation was a pressed steel flat car of 100,000 lb. capacity, the trucks were Baldwin locomotive, M.C.B. interurban type, the motors were four Westinghouse No. 85, the gear ratio was 27 : 47. The total weight of the car was approximately 38 tons. The types of vestibules used consisted of a "standard," "parabolic," "parabolic wedge" and "flat." The dimensions of these are shown by Figs. 26, 27, 28 and 29, respectively.

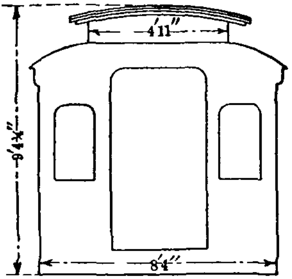


FIG. 29.—Cross-section of car.

Wind Pressure. Wind pressure on a body varies with the velocity of the wind and the shape of the body presented to the wind. For convenience it is given in pounds per square foot of area projected on a plane perpendicular to the direction of the wind. (Wind used in this connection indicates relative motion of air and a body in the air.) The following formula has been found to show wind pressures agreeing with experimental results:

$$P = dV^2$$

in which P = pressure on a plane surface normal to the direction of the wind, pounds per square foot

V = actual velocity of the wind, miles per hour. (See Wind Velocity, p. 141.)

d = wind coefficient. (See p. 141.)

Wind Velocity. When wind velocity is tested by hemispherical cup anemometer

$$V = \log^{-1}[0.509 + 0.9012 \log v]$$

in which V = true wind velocity

v = actual velocity of cup centers.

Velocity given out by the Weather Bureau is obtained on the assumption that

$$V = 3v.$$

This gives a velocity higher than the actual velocity. By the following table the true wind velocity may be obtained from the velocity given out by the Weather Bureau:

Reported velocity	True velocity
10	9 6
20	17.8
30	25.7
40	33.3
50	40.8
60	48.0
70	55.2
80	62.2
90	69.2
100	76.2

Wind Coefficient for Plane Surface Normal to the Direction of the Wind. Following are values of wind coefficient determined by various methods:

Authority	Wind coefficient (*d*
Weather Bureau	0 004
W. J. Davis	0 004
Martin	0 004
Longley	0 0036
Smeaton	0.005

The value of 0.004 is probably the closest approximation.

Wind Pressure on Other than Plane Surfaces. (From a paper by Professor Kernot before the Australasian Association for the Advancement of Science, 1893.) The following determinations were made by placing small models in an approximately steady jet of 10 in. by 12 in. cross-section.

Modulus. At any given wind velocity, the ratio of the wind pressure on a given body to the pressure on a plane surface normal to the direction of the wind and having an area of cross-section equal to the area of projection of the body on a plane normal to the direction of the wind is, for convenience, called the *modulus*.

Cube. The pressure on a cube was as nearly as could be measured the same whether the direction of the wind was parallel to a side or a diagonal and was 0.9 the pressure on a square card equal in size to a face of the cube.

Rectangular Blocks. x = length measured in the direction of the wind, y and z = dimensions in other directions.

Where $x = 2y = 2z$	modulus = 0.8
Where $x = 3y = 3z$	modulus = 0.7
Where $y = 2x = 2z$	modulus = 0.9
Where $y = 3x = 3z$	modulus = 0.9

A block representing a tower and having a height equal to three times its width of base gave a modulus of 0.9 when the direction of the wind was normal to one face. When the direction of the wind was the same as that of a diagonal the effect was, as nearly as could be measured, the same.

Pyramid. A pyramid of square base, having a height equal to about three times its base gave a modulus of 0.8 when a side was presented to the wind. When one angle was presented to the wind the total pressure was increased by 25 per cent.

Cylinder. Cylinders having the elements of their curved surfaces normal to the direction of the wind gave a modulus of 0.52.

Octagonal Prism. The pressure on an octagonal prism was 10 per cent greater than on the circumscribing cylinder.

Cone. For a cone having a height equal to three times the diameter of the base, the modulus was 0 50.

Sphere. For a sphere, the modulus was 0.36.

Hemispherical Cup. For a hemispherical cup (such as is used on Robinson's anemometer): when the convexity was to the wind the modulus was 0.36; when the concavity was to the wind the modulus was 1 15.

Retaining Surfaces. When a surface parallel to the direction of the wind was brought nearly into contact with a cylinder or sphere, the pressure on the latter bodies was increased by about 20 per cent, owing to the checking of the lateral escape of air.

Sheltering Surfaces. When a 9-in. disk was used as a sheltering surface and a 6-in. disk was placed 2 in. in front of it, the latter received only two-thirds the pressure it endured if the larger disk was removed. This reduction in pressure was perceptible, though to a less extent, at all distances up to 9 in.

Temperature Effects on Train Resistance. Train resistance increases with an increase of journal lubricant viscosity; thus when

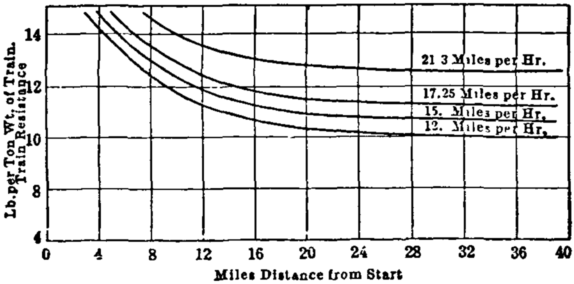


FIG. 30—Decrease of freight train resistance as train progresses (Univ. Ill. tests)

the viscosity of the journal lubricant is increased by the lowering of atmospheric temperature an increased train resistance results. Tests have indicated that at the atmospheric temperatures of ordinary railway operation, train resistance decreases as the journal lubricant becomes less viscous with rising journal temperature, other conditions being constant. When a constant journal temperature is reached on a moving train, the train resistance is at a minimum for the then prevailing track conditions, speed of train and temperature of the atmosphere. Figs. 30 and 31, from Bulletin No. 59 of the University of Illinois Experiment Station, were plotted from data secured from 32 tests. The tests were made in 1910 with trains on the Illinois Central Railroad as they came in regular service Fig 30 indicates that the mean resistance at speeds of from 12 to 35 miles per hour, and atmospheric temperatures from 30 to 42 deg. F., became a minimum when the train had been in motion for about 35 miles. It was found that in warm weather the minimum train resistance for a similar train was reached when the train had been in motion from 8 to 10 miles. Fig. 31 gives a comparison of minimum train resistance values for the two atmospheric temperatures and shows that the minimum train resistance in cold weather is approximately 25 per cent greater than in warm weather.

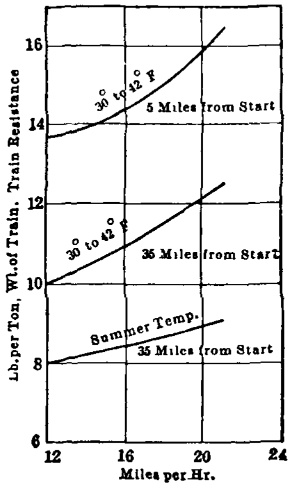


FIG. 31.—Effect of temperature on train resistance (Univ. of Ill. tests).

Fig. 32, as the result of a large number of observations by A. W. Baumgarten, Electrical Engineer of the Chicago and Joliet Electric Railway (*El. Ry Jour*, 1922), shows a typical example of the variation in energy consumption which follows variation in temperature. The tests previously quoted were made on freight trains, while those on which Fig. 32 is based were made in single passenger car operation, and show an energy requirement at freezing temperature about 12 per cent greater than at summer temperature. M. B. Rosevear, Superintendent of Distribution, Public Service Railway (*El. Ry. Jour.*, 1918), shows double this increase between the same temperatures, but, as he states, his cars were electrically heated, the heaters requiring more than 10 per cent of the total energy used by the cars; this explains the apparent discrepancy.

Starting Resistance. All of the train resistance formulas which have been given herein indicate a minimum value for train resistance at the minimum or zero speed. It is a fact, however, that at the instant of starting the tractive effort required to start the car or train from rest is very considerably greater than that needed to keep it in motion at low speed. Such "friction of rest" or "starting resistance" is of importance only in determining the peak power requirement of starting, and in the design of starting resistors for control. Tests made on a single car at Purdue University (*El.

Ry. Jour., 1915) showed a train resistance at the instant of starting equal to four times that at 5 miles per hour; this was reduced one half as soon as the car began to move. It is not unusual that the starting resistance amounts to 40 pounds per ton, which with fair to good conditions may be taken as the maximum; the minimum probably is about 10 pounds per ton. These limits of 10 and 40 pounds per ton are recommended for use in the 1921 Manual of the *Am Ry. Eng. Assn.*, depending on loading, temperature, character and condition of the track and train.

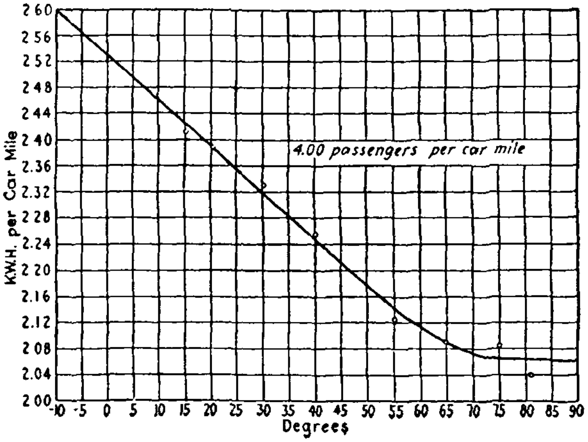


FIG. 32.—Variation of energy consumption with temperature.

Curve Resistance. (See pages 55 and 58 for designation and measurement of track curves.) Curve resistance may be defined as the resultant force, due to track curvature, which opposes the motion of a train at a constant speed on a curved track. It is equal to the tractive effort at the driving wheel treads necessary to keep a train moving at a constant speed on a circular curve of level track in still air, in excess of that necessary to keep the train moving at the same speed on tangent level track of the same character in still air. Curve resistance is commonly expressed in pounds per ton weight of train per degree of track curvature. For convenience it is also often expressed in terms of grade, that is, in terms of that per cent grade whose grade resistance would be equal to the curve resistance under consideration. Many values of curve resistance have been determined by experiment, but the variables concerned are such that no rational method for the general calculation of it has yet been determined. In ordinary cases, the value of curve resistance is small compared with train resistance and grade resistance; also, but a small portion of the total track of any usual railway is curved; therefore, in ordinary cases an accurate determination of curve resistance is of minor importance. Curve resistance is due to the increased slippage between wheel (tread and flange) and rail and increased friction in the moving parts of the train on the curve. Its value depends primarily on the coefficient of friction between wheel and rail, length of truck wheel base, gage, flexibility and condition of the track and the condition of the rail surface. The value of curve resistance on moderate curves is nearly proportional to the degree of curvature. On curves of short radius it is less per degree of curvature than on curves of greater radius. On curves of very short radius, such as are usually found in city streets, conditions are such that no definite statement of the value of curve resistance can be given.

Grade Compensation. In order that the combined up grade effect and curve resistance on a curve may be equal to the grade effect on tangent track having a grade equal to that originally on the curve, the grade is often reduced by the amount whose grade effect would be equal to the curve resistance. This reduction of grade on curves is called "grade compensation."

Determination of Curve Resistance (and Grade Compensation). The Committee on Economics of Railway Location (1910) of the Am. Ry. Eng. and Maint. of Way Assn. reported tests at North Mountain Cut-off and Mt. Airy Grade on the Baltimore and Ohio R. R. to determine the effect of curve compensation. The trains used in these tests were made up of locomotive, dynamometer car, 30 and 36 steel hopper cars (empty and loaded) and caboose. On portions of the grade compensated at the rate of 0.03 per cent per degree of curvature the combined up grade effect and curve resistance was greater than on tangent track, while on portions compensated at the rate of 0.04 per cent per degree of track curvature the combined up grade effect and curve resistance was less than on tangent track. Assuming that the mean of these rates, 0.035 per cent per degree of track curvature, was the correct rate of compensation, the curve resistance was 0.7 lb. per ton weight of train per degree of track curvature, and this is the coefficient most generally used in curve resistance calculations, although it is increased by some engineers to as much as 1.0 pound per ton per degree. The 1921 Manual of the Am. Ry. Eng. Assn. recommends that grades be compensated 0.03 to 0.05 per cent per degree of track curvature, depending upon relative lengths of curve and train, upon location of curve with respect to beginning of grade, upon superelevation of outer rail, and upon train speeds. For electric railway conditions, the following formula may be used:

$$c = 0.7\,D$$

where c = curve resistance, pounds per ton (of 2000 lb.)
D = track curvature, degrees

This is on the assumption that each degree of track curvature is the equivalent of 0.035 per cent of up grade. It should be noted that this formula does not apply to short radius curves with no superelevation of the outer rail such as those found in city streets and usually rated as "special work."

Grade Effect. The component of force along a line parallel to the center line of the track, due to the action of gravity on a train, is called grade effect. Grade effect is commonly expressed in pounds per ton weight of train. If the direction of grade effect is opposite to the direction of motion of the train (that is, if the train is on an up grade) the tractive effort will be decreased by the value of the grade effect. If the direction of grade effect is the same as the direction of motion of the train (that is, if the train is on a down grade) the tractive effort will be increased by the value of the grade effect.

Approximate Value of Grade Effect. The following formula will give results slightly in excess of the true values, but close enough for all ordinary traction work. The error is greater with heavier grades, but reaches only 0.5 per cent for a 10 per cent grade.

$$G = 20\,n$$

in which G = grade effect, pounds per ton (of 2000 lb.)
n = number of per cent of track grade.

This formula may be stated in the form of a rule: Grade effect is equal to 20 lb. per ton weight of train per per cent of grade.

Actual Value of Grade Effect. The precise value of grade effect may be expressed by the formula

$$G = \frac{2000\,n}{\sqrt{10{,}000 + n^2}}$$

in which the symbols are the same as in the preceding paragraph.

It may be noted that the above rule for obtaining the *approximate* value of grade effect yields the *precise* value of grade effect if the per cent grade used in applying the rule is the approximate per cent grade obtained according to "Approximate Value of Grade," below.

Actual Grade. The actual grade of a track is its rate of uniform rise or fall. It is equal to the difference in elevation between two adjacent points on the grade divided by the horizontal distance between them, *i.e.*, the tangent of the angle of inclination of the track to the horizontal. Grade is generally expressed in per cent.

Approximate Value of Grade. As it is easier to measure the distance along the track than along the horizontal, this substitution is often made. This introduces no serious error in the case of the small angles of inclination encountered in the usual railroad work as the sine thus substituted is practically equal to the tangent.

Per Cent of Grade Determined by the Use of Tape Line. Fig. 33 (by G. M. Eaton, *Elec. Journal*, 1911) shows a method of finding the per cent of grade by use of a tape line. The tape is reefed through the end ring or eye, and a pin is stuck through the tape at a point beyond the 12-ft. mark such that when the pin just clears the ring the 12-ft. mark will lie in the center plane of the ring. The tape is creased at the 4-ft. and 9-ft. marks and is fastened to the reel in such a way that the latter may be used as a plumb bob. If possible, the ballast is scooped out to receive the reel hanging freely below the rail, as at D. The tape is tied to the rail at A, the 4-ft. mark, and is suspended by a string or wire at B with the portion AB drawn straight. The portion of the tape from the ring to A is drawn straight and the suspension of B and C are adjusted so that suspended tape and pin clear the ring. The number of per cent grade is then approximately equal to $2x$ in which x is measured in inches. That is, every half inch between the 12-ft. mark and the surface of the rail at G indicates approximately 1 per cent of track grade.

Average Grade. The average grade of a section of track connecting two given points is the ratio of the difference in elevation of these two points to their distance apart, measured in the same unit. It is usually expressed in per cent.

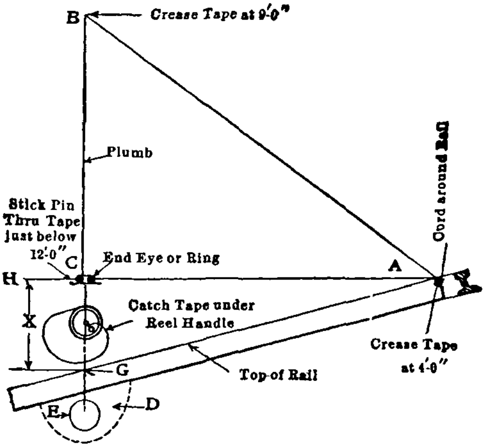


FIG. 33.—Method of finding approximate value of grade by use of tape line.

Ruling Grade. The ruling grade of a road or section of road is that grade which limits the weight and length of train to be propelled over that road or section by a given motive power. Its value is fixed by the traffic and economic conditions encountered on the particular section under consideration. The maximum weight of train to be propelled by a given motive power and limited by the ruling grade is that weight of train which the given motive power would move up a continuous grade equal in per cent to the ruling grade, at some predetermined constant speed. The grade of the greatest per cent on a section need not always be the ruling grade for that section. The length of the grade, the rates of those grades preceding it, and the speed of the train on entering the grade are of as much importance as the rate of the grade, because a grade which may be approached at considerable speed by a given train may be easily passed over while a longer grade of much lower rate may be sufficient to stall the same train.

Momentum Grade. A grade whose operation is made possible by the momentum of a train which approaches the grade at a considerable velocity is called a momentum grade (or velocity grade). Grades of much greater rate than that of the ruling grade may be operated in this way in order to save the expense of their reduction. The argument against the very extensive use of the momentum grade is the possibility of some unforeseen condition such as the presence of a new crossing or a new stopping place which would so reduce the speed of the train that the momentum of the train would be insufficient to help it over the grade. The possibility that such a condition will arise to limit train movement is more remote on an old road where stopping places are fairly well established than on a new road in a rapidly growing community. This point is of importance in fixing the ruling grade for a new road or in reducing the rate on an old one. In some cases the necessity for reducing or removing a grade may be foreseen, but the time of its occurrence may be so remote that it will pay to delay this reduction or removal till the change is made necessary by developing conditions.

Virtual Profile. The electric railway engineer usually finds it convenient to study the relation of grades, trains and equipment together directly on the speed-time and distance-time curves (see Run Curves) for the train and equipment operating in the required direction on a typical run between whose limits lies the particular section under investigation. Unless the train is operated at constant speed (as by three-phase induction motors) care must be exercised that the speed of the train shall become neither dangerously high at the foot of a grade nor so low at the top of a grade that there is a probability that the train may be stalled by a strong head wind or adverse conditions of track, or both. As an aid in adjusting speeds, locating points where power should be cut off, brakes applied, or regeneration resorted to, or in studying the shortening or reduction of grades for the satisfactory operation of a given equipment and train weight over a given section, a virtual profile is sometimes plotted. The virtual profile depends upon the following six items: (1) actual profile of the section considered, (2) track curves in the section, (3) equipment (motors and gears), (4) train weight, (5) direction of motion on the given section, (6) known speed at the beginning of the section. It is only possible to construct a virtual profile for a section at whose beginning the speed of the train is zero or some other definitely known value.

Construction of the Virtual Profile. To construct the virtual profile, the actual profile for the track section is first plotted with elevations as ordinates; then the virtual profile is the locus of all points, the ordinate of each of which is equal to the sum of the corresponding ordinate of the actual profile and the kinetic energy head (see p. 149) (to the same scale) of the given equipment and train, when at that location and traveling in the direction considered. A virtual profile thus constructed will generally be curved. For most purposes, however, it is sufficient to plot only the points of the virtual profile corresponding to the ends of each grade (and curve) and connect the successive points thus plotted by straight lines (see Fig. 34). The virtual profile for a given train and section of track touches the actual profile where the train is at rest, it diverges from the actual profile where the train is accelerating (velocity increasing), it is parallel to the actual profile where the velocity of the train is constant, and it converges toward the actual profile where the train is retarding.

Kinetic Energy Head. The kinetic energy head (also known as the velocity head) for a train at any speed is the height to which the kinetic energy of the train at that speed would lift the train against the force of gravity alone. It is equal to the height through which the train acted on by the force of gravity alone would have to fall from rest to acquire a kinetic energy equal to that of the train at the particular velocity considered. With the falling mass there is no effect of rotating parts, therefore in order to acquire an amount of energy equal to that of the train at any instant a mass equal to that of the train and acted upon by the force of gravity alone must fall

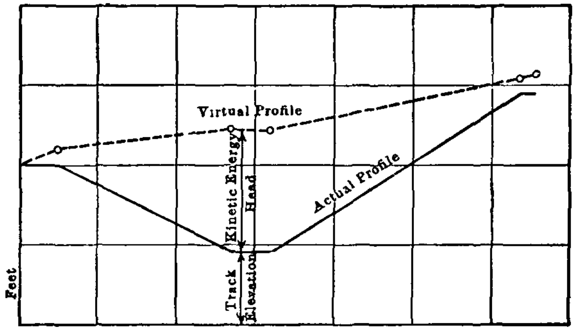


FIG. 34.—Sample virtual profile.

through a greater height than would be necessary in acquiring a speed equal to that of the train at that instant. The following derivation of the formula for kinetic energy head is further explanatory of the term.

At any instant $E = \dfrac{mv^2}{2k} = \dfrac{mv_1^2}{2}$

in which
E = kinetic energy of train at speed S
m = mass of train
v = speed of train, feet per second
k = ratio of linear inertia to total inertia of train (see page 150)
v_1 = velocity of the falling mass m, feet per second

therefore

$v_1^2 = \dfrac{v^2}{k}$ which when substituted in $h = \dfrac{v_1^2}{2g}$ (for a body falling from rest) gives

$$h = \frac{v^2}{2gk} = \left(\frac{0.03342}{k}\right) S^2$$

in which
h = kinetic energy head, feet
S = speed of train miles per hour
k = same as above.

Acceleration

Linear Acceleration. When acceleration is expressed in miles per hour per second, and the force producing it in pounds per ton, the familiar formula for acceleration becomes

$$I = \frac{2000}{32.2} \times \frac{5280}{3600} \times A$$

$$= 91.1\,A = \frac{A}{0.01098}$$

or $A = 0.01098\,I$

where A = rate of linear acceleration, miles per hour per second

I = force required for such acceleration, pounds per ton (of 2000 lb.).

Ratio of Force Required for Linear Acceleration to Force Required for Total Acceleration. As the total effective inertia of rotating parts is greater than their inertia when moved as a mass in a straight line, allowance must be made for this fact in the consideration of train acceleration. This is particularly true in the case of electric motor cars the geared armatures of which rotate at considerably higher speed than the car wheels. The ratio of the "linear inertia" to the "total inertia," including that of rotating parts, may be expressed by K, in the expression:

$$\frac{I}{F} = K$$

in which I = force required for linear acceleration, pounds per ton weight of train

F = total force required to accelerate the train, pounds per ton weight of train

K = ratio of linear inertia to total inertia of train.

$$K = \frac{I}{F} = \frac{W}{W + n_w W_w \left(\frac{r_w}{R_w}\right)^2 + n_a W_a \left(\frac{r_a R_a e}{R_a R_w}\right)^2}$$

in which W = weight of car complete, car or locomotive

W_w = weight of car wheel

W_a = weight of motor armature

n_w = number of car wheels

n_a = number of motor armatures

R_w = radius of car wheel at tread

R_a = radius of motor armature

r_w = radius of gyration, car wheel

r_a = radius of gyration, motor armature

e = gear ratio

F = total effective inertia ($= I + i$)

I = linear inertia

i = rotational inertia

$\left(\frac{r_w}{R_w}\right)^2$ = approximately 0.6 for the average car wheel

$\left(\frac{r_a}{R_a}\right)^2$ = approximately 0.5 for the average railway armature.

Substituting these latter two values the formula becomes:

$$K = \frac{W}{W + 0.6n_wW_w + 0.5n_aW_a\left(\frac{R_ae}{R_w}\right)^2}.$$

The value of K for a train made up of units (motor cars, motor cars and trail cars, or locomotives and trail cars) is equal to the sum of the values obtained by multiplying the value of K for each individual motor car, locomotive, or trail car by the ratio of the weight of that particular car or locomotive to the weight of the whole train.

Ratio of Linear Inertia to Total Inertia in Practice.

For electric locomotive and heavy freight train,	$K = 0.95$
For electric locomotive and high-speed passenger train,	$K = 0.935$
For high-speed electric motor car,	$K = 0.935$ to 0.91
For low-speed electric motor car,	$K = 0.91$ to 0.85.

Formula for Train Acceleration. The formula for the linear acceleration of a car or train including rotating parts then becomes

$$A = 0.01098\ K\ F \text{ (symbols as before)}$$

Inasmuch as an average value of K for electric motor cars is about 91.1 (which makes $0.01098\ K = 0.01$), it is common practice to assume such a value, and for approximate work to use the acceleration formula as

$$A = 0.01\ F$$

The force producing acceleration is made up of the tractive effort of the motors plus the effect of down grade and reduced by the combined effect of train resistance, curve resistance, up grade effect and braking, or such of those elements as may exist at the moment. Inasmuch as the constant in the acceleration formula has been deduced with the accelerating force F expressed in pounds per ton weight of train, each of its elements should be used in the same terms. As a formula, the above becomes

$$F = P - f - c \pm G - B$$

and the complete formula for train acceleration is

$$A = 0.01098\ K\ (P - f - c \pm G - B)$$

or for approximate work

$$A = 0.01\ (P - f - c \pm G - B)$$

where A = rate of acceleration, miles per hour per second
K = ratio of linear inertia to total inertia
F = net force for acceleration, pounds per ton (of 2000 lb.)
P = tractive effort of motors, pounds per ton
$= \frac{Tn}{W}$
T = tractive effort each motor, pounds
n = number of motors each developing T
W = weight of train, tons (of 2000 lb.)
f = train resistance, pounds per ton (see pp. 127 to 144)
c = curve resistance, pounds per ton (see p. 144)
G = grade effect, pounds per ton (see p. 146)
= positive on down grades, negative on up grades
B = braking effort, pounds per ton

Motor tractive effort (P) will be present as a definite value in the formula while power is being used; it equals zero when power is off, as during coasting; it may be obtained through T from the motor characteristic curve, and will vary in amount depending upon the current or speed. Train resistance (f) always will be present in the formula; it will vary in amount for a given equipment depending upon the speed. Curve resistance (c) equals zero on straight track; and grade effect (G) equals zero on level track. Grade effect (G) is entirely independent of speed, while curve resistance (c) and braking effort (B) are practically so. Braking effort (B) equals zero except when brakes are applied. In ordinary operation both P and B rarely appear in the formula at the same time, as brakes are seldom applied while power is on. Constant speed is the equivalent of zero acceleration, or when the conditions are such that $P - f - c \pm G - B = O$. For tangent level track, constant speed is had when $P = f$ (no braking), or in other words, when the speed is such that the train resistance is just equal to the motor tractive effort. When the formula gives a negative value for acceleration, a decreasing speed is indicated; this usually will occur when coasting ($P = O$) or braking, and always will result when the combination of motor tractive effort and down grade effect (if either or both exist) are insufficient to balance the resistance factors f, c, $-G$ and B, or such of them as exist.

Graphical Representation of Acceleration. The slope of any increment of a speed time curve plotted between time in seconds as abscissas and speed in miles per hour as ordinates is numerically equal to the rate of acceleration in miles per hour per second.

Relations between Acceleration, Speed, Time and Distance.

Where A = rate of acceleration, miles per hour per second

S = speed, miles per hour

t = time, seconds

d = distance, miles

then $t = \frac{S}{A}$ and $d = \frac{S^2}{7200A}$

Speed at End of Straight Line Acceleration. To obtain the approximate speed at the end of straight line acceleration, use may be made of the following formula, which, as shown on page 244, gives the approximate relation between the speed and tractive effort of a direct current railway motor:

$$\frac{S_A}{S_M} = \left(\frac{T_M}{T_A}\right)^{1/3}$$

or

$$S_A = S_M\left(\frac{T_M}{T_A}\right)^{1/3}$$

where S_A = speed at end of straight line acceleration

S_M = free running speed

T_A = tractive effort at end of straight line acceleration

T_M = tractive effort at free running speed.

For example, assume a 50-ton car geared for a maximum speed of 60 miles per hour, and an acceleration of 0.8 miles per hour per second. The train resistance at 60 miles per hour (see Fig. 13) is 26 lb. per ton, which at maximum speed is the same as the tractive effort. During acceleration the tractive effort must be approximately ($0.8 \times 100 =$) 80 lb. per ton for acceleration, plus an average of say 16 lb. per ton for train resistance, or 96 lb. per ton total. The speed at the end of straight line acceleration will then be

$$S = 60 \times \left(\frac{26}{96}\right)^{1/3} = 38.8 \text{ miles per hour.}$$

Usual Rates of Acceleration in Practice.

Steam locomotives, freight service	0.1 to 0.2 mi. per hr. per sec.
Steam locomotives, passenger service	0.2 to 0.5 mi. per hr. per sec.
Electric locomotives, passenger service	0.3 to 0.6 mi. per hr. per sec.
Electric motor cars, interurban service	0.8 to 1.3 mi. per hr. per sec.
Electric motor cars, city service	1.3 to 1.8 mi. per hr. per sec.
Electric motor cars, rapid transit service	1.5 to 2.0 mi. per hr. per sec.
Light weight safety cars	1.5 to 2.5 mi. per hr. per sec.
Highest practicable rates	2.5 to 3.0 mi. per hr. per sec.

The maximum possible rate of acceleration is limited by the available tractive effort which is dependent not only upon the motor equipment and weight of train but also upon the coefficient of adhesion between wheel and rail.

Comfortable Rates of Acceleration. It should be remembered that the effect of acceleration on the comfort of passengers is not so much a measure of the rate of acceleration as of the rate of change of accelerating rate. Referring to Fig. 35, the rates of acceleration and of braking as shown by the full lines may be uncomfortable to passengers, due to the abrupt start, application of brakes, and stop. However, if the starting resistance on the first controller position be so proportioned as to give the start shown by the dotted line at A, and the brakes be gradually applied and released so as to replace angles by curves as shown by the dotted lines at B and C, very high rates of acceleration and braking may be employed without causing discomfort to passengers. The inherently high values of train resistance at the instant of starting (see p. 143) tend to round out the initial start at A.

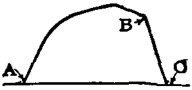


FIG. 35.

Effect on Draft Gear and Other Car Equipment. The above remarks, relative to comfort of passengers, apply equally to shocks to draft gear in train operation and to other parts of the car equipment. To reduce these shocks it is necessary to keep down the rate of change of accelerating rate.

Economical Rates of Acceleration. The most economical rate of acceleration will depend upon (1) the frequency of service, (2) train weight and (3) the capacity of the equipment. If, with the same schedule speed, the acceleration be increased, there will result a slight saving in energy due to the lessened train resistance at the lower maximum speed attained. There will also be a very important saving in energy due to (1) the possibility of a longer coast, allowing braking to begin at a lower speed (the losses in braking being proportional to the square of the speed at which braking begins), and (2) with direct current equipments a saving in rheostatic losses, which are inversely proportional to the rate

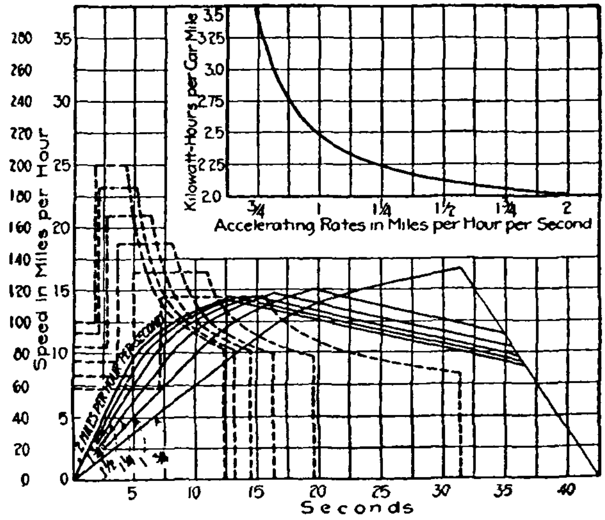


FIG. 36.—Variation of energy consumption with acceleration.

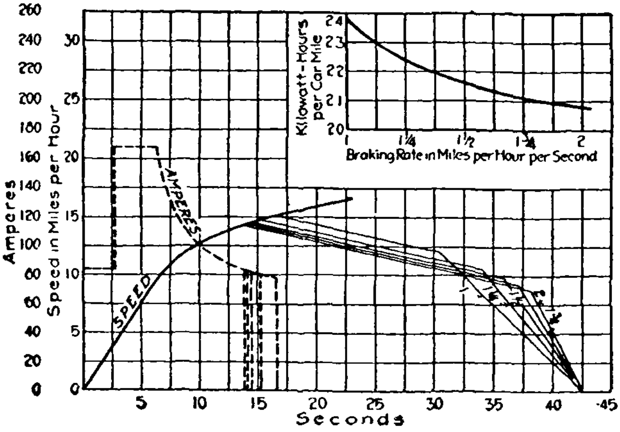


FIG. 37.—Variation of energy consumption with braking rates.

of acceleration. Such savings are effected each time a stop is made, and therefore are much more important in frequent stop service, not only as regards energy consumption, but also with respect to speed. As illustrative of the relation between rates of acceleration and braking on energy consumption, Figs. 36 and 37 have been reproduced from a paper by C. W. Squier (*Elec. Ry. Jour.*, 1918). The curves are based on a study of a 23½ ton car at a schedule speed of 8½ miles per hour, with a 620 ft. run and 7 second stop Fig. 36 shows the effect of changing the rate of acceleration, the braking rate remaining at 1½ miles per hour per second. Fig. 37 shows the effect of changing the rate of braking, while the accelerating rate remains constant at 1½ miles per hour per second. It will be noted that as the rate of acceleration is increased, the relative energy saving becomes less, and this will be found generally true. In this case, had advantage been taken of the possibility of increase in schedule speed (8 per cent between 1.0 and 2.0 miles per hour per second), the energy saving would have been less (only 3 per cent), but the combined savings of platform wages (due to the reduction in car hours) and power cost would have amounted to more than 6 per cent. Formulas for energy losses in rheostatic control are given on page 201, and for losses in braking on page 195.

Relation Between Acceleration and Maximum Speed. As noted in the section on Run Curves, the dotted line in Fig. 44 (p. 164) is the locus of the maximum speeds attained during a run of 1 mile, assuming coasting, braking retardation and train resistance constant. This figure, as well as Fig. 46 (p. 165), illustrates the fact that for a given run the maximum speed usually is reduced by increasing the rate of acceleration.

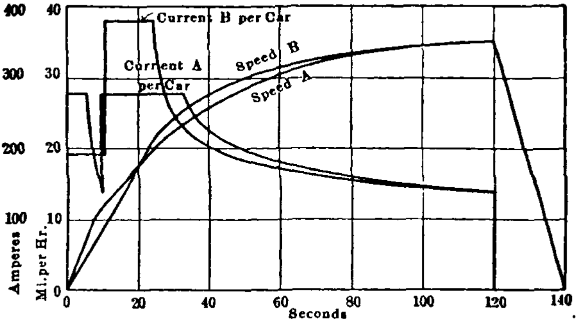


FIG. 38.—Reduction of maximum current by use of lower rate of acceleration in multiple than in series.

Special Accelerating Method. Where the service is infrequent, as in the case of some interurban railways, the acceleration is often limited by the low tension distribution and substation capacity. This limitation may, however, be greatly reduced with the retention of a fair average rate of acceleration and its attending benefits *by using a high rate of acceleration during series acceleration and a* considerably less rate during multiple acceleration. The effects of such a procedure are shown by the speed-time curves and corresponding current curves in Fig. 38. In case *A*, constant current per car is used throughout both series and parallel connections of the motors; this is done by allowing the controller to rest in full series position until the value of current is one-half of what it was during the series acceleration. In case *B*, with a constant rate of straight line acceleration, the value of the current per car when the motors are in parallel is double that when they are in series; in other words, the current per motor is kept constant. In case *A*, for this particular run the rates of straight line acceleration are approximately 1.5 and 0 6 mile per hour per second. In case *B*, the straight line acceleration is approximately 0.9 mile per hour per second throughout. Practically the same energy is used in both cases but the maximum value of current in case *A* is only about 73 per cent of that in case *B*. Another method of using two rates *of acceleration in the series and parallel* positions of the controller is an intermediate one, where the controller is so operated that the current per car with motors in parallel is greater than when in series, but less than double, as in case *B*. The maximum current is then somewhere between the values in cases *A* and *B*. An article by F. E. Wynne in the *Electric Journal* states that for hilly roads this third method is found more satisfactory than the method of case *A*. He also states that on runs of $\frac{3}{4}$ mile or less, the acceleration with constant current per motor (case *B*) is less economical of energy than either of the other methods which are about on a par with each other, but on longer runs the energy consumption is practically the same with all three methods.

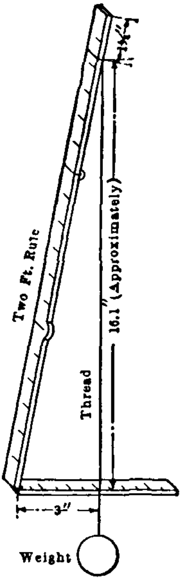


Fig. 39.—Improvised accelerometer.

Improvised Accelerometer. A readily improvised accelerometer is shown in Fig. 39. *It consists of an ordinary 2-ft. four-fold* pocket rule, a piece of thread and a weight. Eighteen inches of the rule is opened out straight and the thread is tied to this part at a point $1\frac{5}{8}$ in from the end; the weight is tied to the other end of the thread, and the remaining 6 in. of the rule is opened and leveled (extending in the direction of motion) till the plumb line cuts the 3-in point of the base thus formed. About 16.1 in. of thread then hangs between the point of support and the base; consequently, each increment of 1 ft. per second, per second acceleration or retardation will be indicated by a corresponding deflection increment of approximately $\frac{1}{2}$ in. at the point at which the thread and the base line intersect. If the thread be attached at a point 6⅝ in. from the upper end, and the upper part of the rule bent further over until the plumb thread again cuts the 3-in point, the deflection of each ½ in. due to acceleration or retardation will indicate 1 mile per hour per second.

Coefficient of Adhesion. As used in railway work the coefficient of adhesion is the coefficient of friction between driving-wheel tread and rail. It is equal to the ratio of the maximum possible tractive effort (neglecting torque limit of motor equipment) at the driving-wheel tread, to the normal pressure (called effective weight on driver) between rail and driving-wheel tread. It is usually expressed in per cent of the effective weight on the driver. The results of tests to determine the coefficient of adhesion vary considerably. This variation is probably due to the difficulty in ascertaining, describing and duplicating the exact condition of rail and wheel surfaces

It is common steam locomotive practice to assume 0.22 to 0.25 as the coefficient of adhesion with clean very dry rails. Because of torque uniformity throughout a revolution of a driving wheel higher maximum values are possible in electric railway practice, but in order to take full advantage of this fact all sudden fluctuations in torque such as might be brought about by improper arrangement of resistance taps or improper controller manipulation must be avoided.

Coefficient of adhesion is a quite variable quantity depending upon the condition of the rail and wheel. Neglecting the use of sand, a clean dry or very wet rail gives the highest coefficient of adhesion. Instances have been noted where the value of the coefficient of adhesion with a thoroughly wet rail has been nearly that for a perfectly dry rail, while with the presence of moisture in amounts scarcely perceptible to the unaided eye it has been found to be very low. Sand, while detrimental in its abrasive effect, improves poor adhesion and by its use very high values may be obtained with dry rail. It is, however, unsafe to depend upon sand to improve adhesion with dry rail because the sand is likely to be blown from in front of the wheel. When the wheels are slipping the coefficient of kinetic friction between driving wheel tread and rail is less than 0.10.

Approximate Values of Coefficient of Adhesion

Clean dry rail	0 25 to 0 30;	with sand 0 35 to 0 40
Clean thoroughly wet rail	0 18 to 0 20;	with sand 0 22 to 0 25
Greasy and moist rail	0 15 to 0 18,	with sand 0 22 to 0 25
Sleet on rail	0 15	with sand 0 20
Light snow on rail	0 10	with sand 0 15

It is evident that the coefficient of adhesion is the factor which, with a given weight on the driving wheels, very definitely limits the tractive effort of a car or locomotive, and therefore also limits the rate of acceleration or the grade which may be surmounted with a given weight of train, or the weight of train which may be handled over a given grade or accelerated at a given rate. Conversely, with a given weight of train and rate of acceleration or per cent of grade, the coefficient of adhesion determines the required weight on the driving wheels. The coefficient of adhesion further determines the maximum possible braking rate

In the consideration of any such problems where the coefficient of adhesion is the limiting factor relative to rates of acceleration or braking, due allowance must be made for transfer of weight between axles in the direction opposite to that of positive acceleration.

Weight Transfer Due to Acceleration. Due to the facts that effective traction is applied at the points of contact between driving wheels and rails, and that both the center of gravity and the drawbar of the motor car or locomotive are located considerably above the rail, there will be a transfer of the weight of the car or locomotive from the forward to the rear wheels during acceleration, and at any other time when positive drawbar pull is being exerted on trailing cars. Such weight transfer is similar to, but opposite in direction to that which occurs in braking, as discussed by R. A Parke, A I E.E., 1902, and abstracted at page 444. Formulas for such weight transfer due to acceleration and drawbar pull of trailers are as follows:

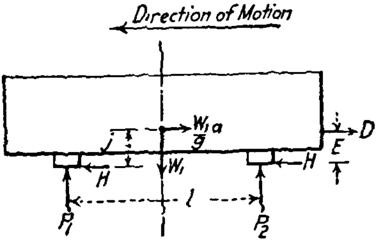


FIG 40 —Weight transfer in acceleration Car body.

Case of One Car or Locomotive Alone with no trailing cars and hence no drawbar pull.

External Forces Acting on Car Body in Accelerating. (Fig 40.)

$$H = \frac{W_1 a}{2g}$$

$$P_2 = \frac{W_1}{2} + \frac{W_1 aj}{gl}$$

$$P_1 = \frac{W_1}{2} - \frac{W_1 aj}{gl}$$

in which H = horizontal retarding force on each truck center pin, pounds (the same for each truck because the same number and size of motors per truck, similarly controlled and providing the same tractive effort per truck)

P_2 = pressure between body and truck rear center plates, pounds

P_1 = pressure between body and truck forward center plates, pounds

W_1 = weight of car body, pounds (Center of gravity being in a vertical axis midway between truck centers)

j = height of center of gravity of body above center plate surface, inches

l = distance between center pins, inches

g = acceleration due to gravity, feet per second per second = 32 2

a = rate of acceleration, feet per second per second = (miles per hour per second) × 1.467.

External Forces Acting on Forward Truck in Accelerating. (Fig. 41)

$$R_1 = \frac{W_2}{2} + \frac{P_1}{2} - \frac{Hh}{b} - \frac{W_2ad}{gb}$$

$$R_2 = \frac{W_2}{2} + \frac{P_1}{2} + \frac{Hh}{b} + \frac{W_2ad}{gb}$$

$$T_1 = qR_1 \qquad T_2 = qR_2$$

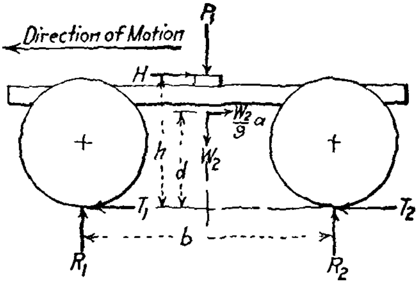


FIG. 41.—Weight transfer in acceleration. Trucks.

in which R_2 = total pressure between rail and rear pair of wheels of forward truck, pounds
R_1 = total pressure between rail and front pair of wheels of forward truck, pounds
T_2 = total maximum tractive effort available between rail and rear pair of wheels, pounds
T_1 = total maximum tractive effort available between rail and front pair of wheels of front truck, pounds
W_2 = weight of each truck, pounds (Center of gravity of truck being in vertical axis midway between wheels)
W = total weight of car, pounds = $W_1 + 2W_2$
h = height of truck center plate surface above rail
b = wheel base of truck
d = height of center of gravity of truck above rail
q = coefficient of adhesion between wheel and rail (0 25 may be used for this).

For significance of other symbols see page 158.

The above formulas do not take into consideration weight transfer due to the inertia of rotating parts (wheels, axles and armatures). This effect may vary, depending upon the relative weights and relative arrangements of such rotating parts, the effect is small, however, and in all practical cases, the formulas as shown will give results within the limits of error in the assumptions necessary with respect to location of centers of gravity and coefficient of adhesion.

Case of Car or Locomotive with Trailer or Trailing Cars. (Figs. 40 and 41)

Drawbar pull D will be applied at the coupler which is a distance m above the line of application of the forces H

$$H = \frac{W_1 a}{2g} + \frac{D}{2}$$

$$P_2 = \frac{W_1}{2} + \frac{W_1 aj}{gl} + \frac{Dm}{l}$$

$$P_1 = \frac{W_1}{2} - \frac{W_1 aj}{gl} - \frac{Dm}{l}$$

$$R_1 = \frac{W_2}{2} + \frac{P_1}{2} - \frac{Hh}{b} - \frac{W_2 ad}{gb}$$

$$R_2 = \frac{W_2}{2} + \frac{P_1}{2} + \frac{Hh}{b} + \frac{W_2 ad}{gb}$$

$$T_1 = qR_1 \qquad T_2 = qR_2$$

in which D = drawbar pull in pounds.

m = height of drawbar pull or coupler above center plate surface, inches.

The significance of other symbols, the methods of deriving equations and the considerations neglected are the same as in the preceding case.

Run Curves

Speed is the desired end in the solution of most railway problems. Revenue usually is in some proportion to the miles operated, while many important expenses are more nearly proportional to time. Within certain limits, therefore, efforts should be made to increase miles operated in a given time, or to reduce the time for operation over a given distance, either process resulting in increased speed. Under some conditions, however, the attainment of increased speed is costly, and under any conditions a critical point is reached where the cost of a further increase in speed is not justified. It is some phase of this problem that is involved in nearly every engineering study of train movement. Factors must be considered such as size and headway of cars or trains, frequency and duration of stops, track grades and curves, train resistance, acceleration and braking rates, coefficient of adhesion, signalling, motor capacities, gear ratios, control and starting resistance design, power and energy requirements, feeder layout and power supply.

Speed Determination by Counting Poles or Rail Joints. The approximate speed of a car may be determined by counting the number of evenly spaced poles or rail joints passed in a given time. The speed in miles per hour is equal to the number of poles or rail joints passed in 0 682*d* seconds, where *d* is the spacing of the poles or rail joints in feet. Thus, approximate miles per hour speed =

number of poles passed in 68	seconds (100 ft. spacing)
number of poles passed in 75	seconds (110 ft. spacing)
number of poles passed in 85	seconds (125 ft. spacing)
number of joints passed in 20	seconds (30 ft. rails)
number of joints passed in 22½	seconds (33 ft. rails)
number of joints passed in 41	seconds (60 ft. rails)

The Speed-Time Curve is the convenient and useful medium for the study of the interrelation of the many elements suggested above. Plotted in ordinates of speed against time, its slope at any point is a measure of acceleration, the latter being the resultant of the potential, kinetic or frictional forces which tend to aid or oppose the motion of the car or train. A knowledge of such forces as are applicable to the given problem enables the engineer, through the acceleration formula, to plot the speed-time curve, and through the latter to make further determinations relative to distance, power, energy, losses, and the like. The detail of such a study, and the exactness of the resulting determinations may depend upon the degree of accuracy desired, or upon the limitations of the available data. Preliminary estimates from rough data need not nor cannot be made with as much detail as to minor factors as is possible and often desirable in specifying final equipment in costly installations. In ordinary street railway service where succeeding trips are made with stops varying in number and location, an average length, time and grade may be taken and "typical" run curves usually are employed for the solution of the problems usually encountered. In rapid transit or heavy traction service where stops are at definite fixed stations, the predetermination of motor or signal requirements may involve a consideration of the actual stops, profile and alinement, with the resultant run curves plotted in considerable detail from one end of the route to the other, in each direction and for each class of service. The methods employed in the construction of the speed-time curve vary considerably with the requirements of the case in hand. Where but few typical runs are to be considered with the same equipment, or a study is to be made of the effect of a change in motors, gear ratio, or accelerating rate, for instance, the "step-by-step" *method is most often employed*, with succeeding calculations of the acceleration formula applied directly to the speed-time curve. Where a definite equipment is to be considered as operating over runs of varying length with definitely fixed stops, grades and curves, it may be advisable to adopt some such method as that proposed by C. O. Mailloux, including the preparation of working charts or templets as ultimate time-savers. Or, where data is most meager, preliminary estimates may be made by the use of "straight line" curves, or charts or tables of more or less general application.

Data Required for Speed-Time Curves. The data listed below is required for the construction of speed time curves, the exactness depending upon the degree of accuracy desired. Letters to the left refer to one of three commonly used methods as listed below, a capital letter indicating imperative need, lower case indicating data desirable for exactness but not essential for close approximations.

A Straight line approximation—no definite motor under contemplation—typical run. See pp. 163–167.
B Step-by-step method with motor characteristic—typical run. See pp. 167–176.
C Detailed run with definite stops, grades and curves, and with greatest accuracy. See pp. 176–184.

ABC Number of motor cars in train
ABC Number of trail cars in train
ABC Weight of motor car or locomotive, exclusive of load
ABC Weight of trail car, exclusive of load
abC Weight of locomotive on drivers
ABC Weight of average load, motor and trail cars
bc Weight of maximum load, motor and trail cars
bC Cross-sectional area of locomotive and cars
BC Number of motors per motor car or locomotive
A Rate of acceleration
BC Characteristic curve of motor
bC Line voltage at train, average, maximum and minimum
bc Electrical resistance of motors
bc Weight of motor armature
bc Diameter of motor armature
BC Diameter driving wheels
bc Weight driving wheels
bc Diameter and weight of trailing wheels
BC Gear ratio of motors
bC Profile and alinement of track
ABC Length of run
ABC Time of run
ABC *Number of stops*
bC Location of stops
aBC Duration of stops
c Layover at ends of line
C Speed limitations
BC Braking rate

Determination of Typical Run. In selecting a typical run it is advisable to divide the line into general sections, for instance, city, suburban and interurban, and then use the approximate average length of run in a section for the length of the typical run for that section. If there are steep grades on the line there should also be a division into grade sections and a typical run selected for each of these. The average equivalent grade, in per cent, may be obtained by the following formula, which assumes only one-half the gravitational energy on down grades to be saved, the remainder being consumed in braking. Other assumptions may be found necessary in certain cases, for instance, in city work, more than half of this energy may be wasted in braking; in high speed limited interurban service on private right of way, less.

$$G = \frac{100}{l}(H_1 - \tfrac{1}{2} H_2)$$

where G = average equivalent grade, per cent
l = distance, feet
H_1 = summation of rises, feet
H_2 = summation of falls, feet.

In determining the average length or time of a typical run, the total distance or time should be divided by the total number of equivalent stops. The term "equivalent stops" includes slowdowns in addition to actual stops, as it is necessary to allow for the former as well as for the latter. In the absence of data permitting a more accurate determination, it is usual to make each slowdown where brakes are applied equivalent to one-half a stop in city service, or one-third stop in suburban or interurban service. The number of equivalent stops is then the number of actual stops plus one-half (or one-third) the number of slowdowns.

Wherever the frequency of stops, maximum speeds, or grades are materially different on various sections of a line, the distances, stops, running times and grades should be segregated, and separate typical runs made up for each such section.

"Straight Line" Approximation of Speed-Time Curves. Several authors have proposed "general" solutions of speed-time problems, based upon either an assumed general motor characteristic curve or an assumption of uniform rates of acceleration and retardation. As uniform rates of acceleration are represented by straight lines in the speed-time curve, the general solution based on the latter assumption has been called the "straight line method." It is most convenient for quick and rough approximation, and involves no reference to motor characteristic curves. The following description, with Figs. 42 to 46, inclusive, is from the Standard Handbook, 1922, section 16, paragraphs 39 to 47, inclusive.

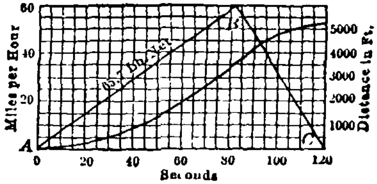


FIG. 42.—Typical straight line speed and distance curves (no coasting).

The speed-time curve is shown in Fig. 42 in its simplest form, acceleration being carried on at constant rate up to the point of applying brakes, which are so applied as to give a constant rate of braking. The area enclosed within the triangle ABC is proportional to the distance traveled, the distance covered up to any instant being shown by the distance-time curve. Thus, with the constants chosen in Fig. 42, a maximum speed of 60 miles per hour is required to obtain an average speed of 30 miles per hour, that is, covering a distance of 5280 ft. in 120 seconds.

In practical operation it is not possible to choose the rate of acceleration and braking with such nicety as shown in Fig. 42, a greater or less period of coasting being required. Introducing coasting gives rise to the form of speed-time curve shown in Fig. 43, showing three friction rates: $f = 0$, 15 and 30 lb. per ton respectively. With no friction the speed-time curve $ABCD$ is constructed, the speed being constant at 40 miles per hour during the coasting period. With 15 lb. per ton friction, the speed-time curve $AEGD$ is formed, and with 30 lb. per ton friction the speed-time curve $AFHD$. The introduction of friction occasions a falling off of speed during the coasting period proportional to the friction value taken, which for the sake of simplicity is here assumed to be constant at all speeds.

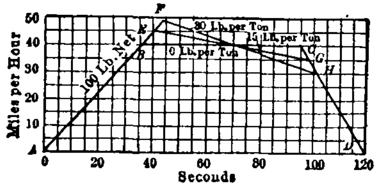


FIG. 43.—Typical straight line speed-time curves (various coasting resistances).

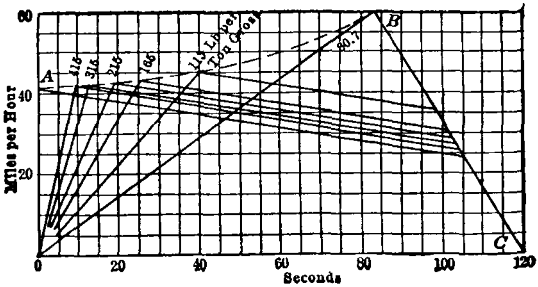


FIG. 44.—Typical straight line speed-time curves (various rates of acceleration).

The speed-time curves shown in Figs. 42 and 43 both indicate the completion of the run of 5280 ft. in 120 seconds, although in one case the rate of acceleration was that produced by 65.7 lb. per ton, and in the other case by 100 lb. per ton. These curves are of equal area, as the distance in each case is 5280 ft., and it thus becomes possible to produce any number of speed-time curves for a given distance and elapsed time by varying the rate of acceleration with consequent variation in time of coasting.

A more extended set of curves is given in Fig. 44, for the same distance of 1 mile covered in 120 seconds, the rate of acceleration varying from 0.713 mile per hour per second as a minimum to an infinite number of miles per hour per second as a maximum. A train resistance value of 15 lb. per ton is assumed constant at all speeds and the dotted curve AB is the locus of the maximum speeds reached with the different rates of acceleration. The highest maximum speed required is obtained with no coasting, and the minimum speed is obtained with an infinite rate of acceleration. The tractive efforts corresponding to the different accelerating rates are given as including 15 lb. per ton train resistance, hence the net tractive effort values corresponding to the rates of acceleration indicated are 15 lb. per ton less than the figures given.

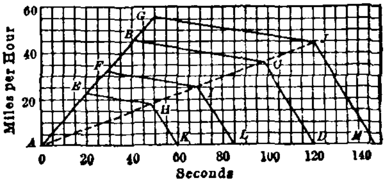


FIG. 45.—Similar straight line speed-time curves (various distances).

Instead of plotting similar curves for distances other than 5280 ft., advantage may be taken of the fact that the area enclosed by the speed-time curve is proportional to the distance travelled and the coordinates are proportional to the square root of the enclosed area. It is convenient therefore to plot a full series of curves for one distance, preferably one stop per mile, that is, a distance of 5280 ft. run, and apply the results so obtained to any other distance by using a factor expressing the relation of the square roots of the distance traveled. This is shown in Fig. 45, where $ABCD$ represents an area of 1 mile, or one stop per mile; $AFIL$ two stops per mile with a factor of $\sqrt{0.5} = 0.707$; $AEHK$ four stops per mile with a factor of $\sqrt{0.25} = 0.5$, and $AGJM$ one stop in 1½ miles with a factor of $\sqrt{1.5} = 1.225$.

Referring to Fig. 44, it is obvious that a similar sheet could be prepared for any elapsed time other than 120 seconds, using the same train resistance and braking values of 15 and 150 lb per ton respectively. Fig. 46 is so constructed to show the time limits imposed by 15 lb. per ton train resistance, and 150 lb. per ton braking effort for any length of run and any rate of acceleration. The dotted curves indicate the locus of the several maximum speeds reached with different accelerating rates for a run made in a given elapsed time; thus the dotted curve terminating at 80.7 lb. per ton is a reproduction of the similar dotted curve, AB, given in Fig. 44, and shows the maximum speed reached with any rate of acceleration for a run of 5280 ft. in 120 seconds with 15 lb. per ton train resistance and 150 lb. braking effort. Similarly, the dotted curve terminating at 100.4 lb. per ton shows the limiting maximum speeds reached with any rate of acceleration when a run of 5280 ft. is accomplished in 110 seconds with the same values of train resistance, braking, etc.

The line CD shows the slope of a coasting line at the rate of 15 lb. per ton train friction. Thus in a run completed in 120 seconds, the minimum accelerating rate corresponds to 80.7 lb. per ton (gross) with no coasting, braking commencing as soon as acceleration ceases. If a higher rate of acceleration than 80.7 lb. per ton be used for a cycle completed in 120 seconds, for example 132 lb. per ton (gross), coasting must be introduced between the accelerating and braking lines. This coasting line may be plotted, with 132 lb. per ton acceleration, by drawing a line parallel to the line CD (Fig. 46) starting at intersection of accelerating line, 132 lb. per ton, and dotted line to 80.7, and terminating at intersection with braking line ending at 120 seconds.

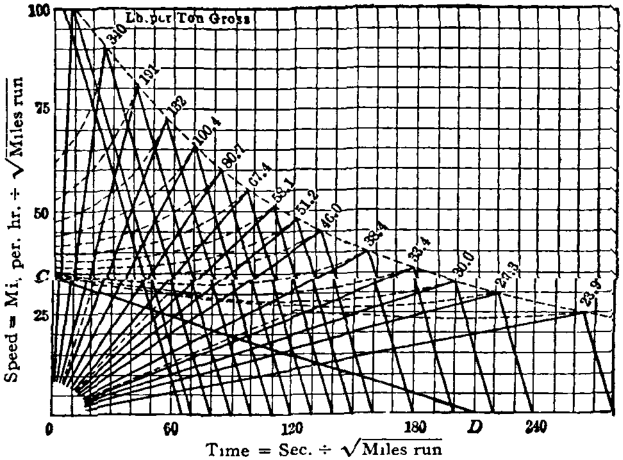


FIG. 46.—General straight line speed-time curves (train resistance, 15 lb. per ton; braking resistance, 150 lb. per ton).

By the use of Fig. 46 it becomes possible to determine the time required to make a run over any distance with any rate of acceleration, provided the train resistance is 15 lb. per ton and braking corresponds to 150 lb. per ton retarding effort.

Example: Given a distance of 8000 ft., train resistance 15 lb. per ton, braking effort 150 lb. per ton, gross tractive effort 67.4 lb. per ton (including 15 lb. per ton train resistance), what is the minimum time required to perform the run and what maximum speed is reached?

Solution: From Fig. 46, minimum elapsed time with 67.4 lb. tractive effort is 130 seconds with no coasting.

$$\text{Ratio of distances} = \sqrt{\frac{8000}{5280}} = 1.23.$$

Hence for 8000 ft. time of run = 130 × 1.23 = 160 seconds.
Maximum speed for 5280 ft. = 55.6 miles per hour.
Hence for 8000 ft. speed = 55.6 × 1.23 = 68.5 miles per hour.

In actual practice, a certain amount of coasting is necessary; hence, the run of 8000 ft. would be made in somewhat more than the minimum possible limit of 160 seconds, or else the tractive effort should be increased to allow for some higher rate of acceleration which would permit of some coasting.

The values chosen for train resistance and braking effort in Fig. 46 (15 lb. and 150 lb., respectively) are conservative and practical operating values. The maximum speed during a service run is influenced but little by the type of motive power and its characteristics, and will differ but slightly from that indicated by Fig. 46. This chart, therefore, constitutes a set of fundamental data by means of which it becomes possible to approximate the several data in any acceleration problem.

Step-by-Step Method, Speed-Time Curves

In this method the formula for acceleration is applied successively to the various conditions of speed, tractive effort, train resistance, curve resistance, grade effects and braking, as such conditions change and present themselves throughout the run. During successive increments of speed or time, the acceleration rates are assumed to be constant, and are plotted as slopes. It is only necessary that such increments be made sufficiently small to obtain great accuracy in plotting. It should be remembered, however, that such formulas *as those for train resistance, which enter into the acceleration formula*, are based upon certain assumptions, and the measure of probability of error in such formulas is the measure of maximum possible accuracy in the resulting speed-time curves.

Definite motor characteristics are usually used in the construction of speed-time curves. However, as the characteristics of railway motors are very much alike, the characteristic curve of any motor of the proper type and capacity may be used in preliminary work without fear of serious error. In some cases it even may be advisable to disregard any definite motor, and use a "general" motor characteristic such as described on page 242, and following.

In the following, the plotting of a speed-time curve is described and illustrated by the solution of a definite problem. A method is shown where the curve is plotted in steps after a calculation of the acceleration rate for each step, and this is followed by a convenient graphical solution. In each case a typical run has been considered.

Step-by-step method of plotting the speed-time curve for a given car or train starting from rest and running a given distance to a stop, at a given schedule speed. It is desirable to divide the speed-time curve into the following principal parts which are given in the order in which they usually are constructed:

(1) Acceleration with motor current controlled, called for convenience, Straight Line Acceleration. (*oa*, Fig. 47.)

(2) Acceleration beyond Straight Line Acceleration, or "acceleration on the motor curve." (*ab*, Fig. 47.)

(3) Stand-still during stop, allowed for in the schedule speed. (*de*, Fig. 47.)

(4) Braking. (*cd*, Fig. 47.)

(5) Coasting. (*bc*, Fig. 47.)

Scales. The speed-time curve shows the speed of the train at any instant during the run and usually is plotted between time in seconds as abscissas and miles per hour as ordinates.

(1) *Acceleration with Motor Current Controlled (Straight Line Acceleration) oa.*

$$\textit{Slope of oa} = A = \frac{\text{miles per hour}}{\text{seconds}}, \text{ drawn to scale.}$$

in which A = assumed acceleration, miles per hour per second, or the acceleration produced by the assumed maximum current per motor, found as below under (2).

Speed at a. The approximate speed of the train at a, Fig. 47, is determined from the motor characteristic curve, and is either the speed corresponding to the assumed value of current per motor during straight line acceleration, or to the tractive effort T determined in accordance with the following:

$$T = \frac{W}{n}\left(\frac{A}{0.01098K} + f + C \pm G\right)$$

in which T = tractive effort at driving wheel tread for one motor, pounds

W = weight of train, tons

n = number of motors

A = rate of straight line acceleration, miles per hour per second

K = ratio of linear inertia to total inertia of train (See page 150)

f = average train resistance of the train between the initial speed and an assumed probable approximate speed at a (Fig. 47), pounds per ton weight of train

C = curve resistance, pounds per ton weight of train

G = grade effect, pounds per ton weight of train. (Use (+) before this value for up grade and (−) for down grade.)

(2) *Acceleration Beyond Straight Line Acceleration (ab).* The curve *ab* is approximated by a succession of straight lines (*ag*, *gh*, etc., Fig. 47).

Slope to scale of each of these lines =

$$A = 0.01098K(P - f - C \pm G)$$

in which, for any part such as *ag*,

$$P = \frac{Tn}{W} = \text{tractive effort, pounds per ton weight of train.}$$

P, T, and f are taken as the mean of those values between the speeds a and g. For this and other parts of the speed-time curve beyond the straight line acceleration portion, T is obtained from the motor tractive effort characteristic curve. The sign of G is (−) for up grade and (+) for down grade. The point b will be determined by the final location of the coasting line bc.

Speed Increment. The amount of the speed increment between a and g will depend on the allowable error in approximation. For the same allowable error, this increment may be greater in the steeper than in the flatter portions of the curve.

(3) *Stop Period, de.* The time of run, in seconds, is

$$oe = \frac{\text{miles from start to stop} \times 3600}{\text{miles per hour schedule speed}}$$

and $$od = oe - (\text{stop period in seconds})$$

(4) *Braking Line, cd.* The slope to scale of the braking line cd is equal to the assumed rate of braking retardation. The point c is determined by the final location of the coasting line bc.

(5) *Coasting Line, bc.* The average slope to scale of the coasting line bc is

$$A = 0.01098K(-f - C \pm G)$$

in which f = approximate average train resistance between the probable approximate assumed speeds at b and c, pounds per ton weight of train.

K, C and G are the same as before.

The line bc is placed so that area ($oabcd$) is equivalent to the distance to be covered by the run (see page 184).

Theoretically, the coasting line is curved, as the retarding forces decrease as the speed decreases; practically no great error is involved by assuming the retarding forces constant; these are often taken as equal to the train resistance at the speed at which power is cut off, assuming that the decreasing train resistance is offset by the neglect of motor friction and windage.

See pages 171 and 175 for more definite location of coasting line.

As an **example** to illustrate the application of the formulas and method described above, assume the following conditions:

Weight of car, 18 tons
Diameter of wheels, 30 inches; weight, 400 lb.; number, 8
Motor equipment, two GE-247 motors; gear ratio $^{63}/_{15}$
Weight of armature 503 lb.; diameter $10\frac{3}{8}$ inches
Average line voltage, 600
Average starting current per motor, 70 amp.
Braking rate, 1.5 mi. per hr. per sec.
Average distance between stops, 0.46 mile
Schedule speed, 17.4 mi. per hr.
Average duration of stop, 5 seconds.

Required to construct the speed-time curve for a complete run, neglecting track curves and grades.

Ratio of Linear to Total Inertia. The value of K is 0.924, calculated from the above data by the formula given on page 150. In the absence of such data relative to wheel and armature weights and dimensions, a value for K may be assumed by inspection of the table on page 151, or for less accurate work, the constant 0.01 may be substituted for $0.01098K$ in the calculations of acceleration rates.

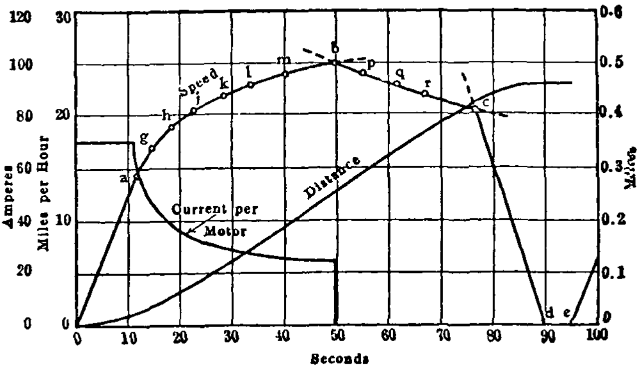


FIG. 47.—Typical run curves For data, see p. 169; calculations below

Section or point	Speed, m p h	I per motor	T lb.	P lb.	f lb.	A m p h p s	Distance, miles
o to *a*			1230	136 7	12 5	1 260	
a	14 2	70 0					0 022
a to *g*	avg 15 6		910	101 0	14 5	0 878	
g	17 0	47 0					0 036
g to *h*	avg 18 0		600	66 7	15 3	0 522	
h	19 0	38 0					0 055
h to *j*	avg 19 7		460	51.1	15 8	0 358	
j	20 5	33 0					0 078
j to *k*	avg 21 2		380	42 2	16 4	0 262	
k	22 0	29 5					0 112
k to *l*	avg 22 5		330	36 7	17 0	0 200	
l	23 0	27 5					0 144
l to *m*	avg 23 5		290	32 2	17 5	0 149	
m	24 0	26 0					0 188
m to *b*	avg 24 5		250	27 8	18 0	0 099	
b	25 0	24 5					0 254
b to *p*	avg 24 5		0	0	18 0	−0 183	
p	24 0	0 0					0 290
p to *q*	avg 23 5		0	0	17 5	−0 178	
q	23 0	0 0					0 329
q to *r*	avg 22 5		0	0	17 0	−0 173	
r	22 0	0 0					0 366
r to *c*	avg 21 2		0	0	16 4	−0 166	
c	20 5	0 0					0 421
c to *d*			0	0		−1 500	
d	0	0 0					0 460

(1) *Straight Line Acceleration, oa.* The speed of the car at *a*, Fig 47, is 14 2 mi per hr at 70 amp , as read from the characteristic curve of the motor, which is shown in Section IV as Fig 23, page 240.

$$\text{The slope of } oa = A = 0\ 01098K\left(\frac{Tn}{W} - f\right)$$

$$= 0\ 01098 \times 0\ 924\left(\frac{1230 \times 2}{18} - 12.5\right)$$

$$= 1\ 26 \text{ mi per hr per sec.}$$

where $T = 1230$ lb. at 70 amp., as read from the characteristic curve of the motor, and where $f = 12.5$ lb. per ton average train resistance between 0 and 14.2 mi. per hr., as interpolated with sufficient accuracy for this example from Fig. 13, page 131. On a sheet of cross-section paper, lay off appropriate units of time as abscissas and units of miles per hour as ordinates, as in Fig. 47, and draw the line *oa* with a slope of 1.26 mi. per hr. per sec., terminating at 14.2 miles per hour and $\frac{14.2}{1.26} = 11.3$ seconds.

(2) *Acceleration beyond Straight Line Acceleration, ab.* Assume an increment of speed of 2.8 miles per hour. Then between 14.2 and 17.0 miles per hour, $A = 0.01098 \times 0.924\left(\frac{910 \times 2}{18} - 14.5\right)$ $= 0.878$ mi. per hr. per sec., where T and f are determined as above at the average speed of $\frac{14.2 \times 17.0}{2} = 15.6$ miles per hour. Draw *ag* having a slope of 0.878 *mi.* per hr. per sec., and *terminating at 17.0* miles per hour. Calculate the accelerating rates for succeeding increments, and draw in a similar manner, continuing the construction of the acceleration line to a point somewhat beyond the point at which it is probable that coasting will begin.

(3) *Stop Period, de.* The time of run, in seconds, is

$$oe = \frac{\text{miles from start to stop} \times 3600}{\text{miles per hour schedule speed}} = \frac{0.460 \times 3600}{17.4} = 95$$

seconds

and $od = oe - de = 95 - 5 = 90$ seconds.

(4) *Braking Line, cd.* Draw *cd* of indefinite length, starting at 90 seconds, zero speed, and having a slope of -1.50 mi. per hr. per sec., the given braking rate.

(5) *Coasting Line, bc.* On a separate sheet of similar cross-section paper, lay off axes with units of time and miles per hour exactly like the ones first constructed and, beginning at a speed somewhat greater than the probable maximum, calculate and draw the coasting line for proper decrements of speed down to a speed somewhat lower than that at which it is probable that braking will begin. The decelerating rate A for each decrement is calculated exactly as for accelerating increments, *a* to *b*. Place the second sheet carrying the coasting line beneath the first sheet carrying the accelerating and braking lines, and with base lines coinciding, slide it horizontally to a position where the coasting line, *bc*, is so placed that the area (*oabcd*) $\approx$ length of run $= 0.46$ mile. Trace the coasting line on the first sheet, intersecting the accelerating line at *b* and the braking line at *c*. Several trials for the position of *bc* may be required. The area may be determined by the use of a planimeter or by counting the squares. The number of squares, of the size shown in Fig. 47, needed under the complete curve $= a =$ $\frac{d \times 3600}{ts} = \frac{0.46 \times 3600}{10 \times 5} = 33.12$, as derived from the formula for d on page 184.

In the actual construction of the speed-time curve, a much larger diagram than Fig. 47 should be drawn, the size depending upon the degree of accuracy required.

Chart for Speed-time Curve Calculations. Fig. 48 is illustrative of the form of chart used by S. B. Cooper to locate points on the motor acceleration and coasting portions of a speed-time curve. To the right and left of the point O, Fig. 48, are laid off positive and negative values of tractive effort. These may be in pounds per motor as shown in Fig. 48, or in pounds per ton as in the preceding discussion. To a suitable vertical scale of miles per hour, curves are plotted showing the relation between pounds tractive effort

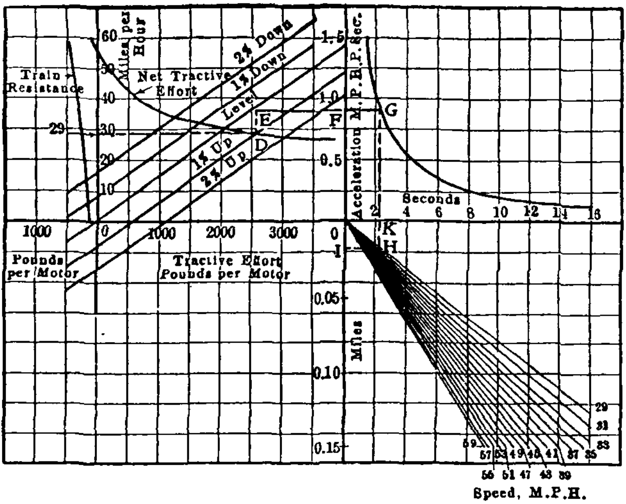


FIG. 48.—Sample chart for plotting speed-time curve, step-by-step method.

and speed, for "net tractive effort" and "train resistance." To a second vertical scale of miles per hour per second, straight lines are plotted corresponding to level track and various up and down grades, showing the relation between net tractive effort and acceleration for the various grade conditions, by the acceleration formula. In the upper right quarter of the chart, and to the same scale of miles per hour per second, various reciprocal curves are plotted, showing the relation between acceleration and time, for various increments of speed (see "Chart of Reciprocals," page 179). In Fig. 48, only one such curve is shown, that for an increment of two miles per hour. In the lower right quarter of the chart, and to the same scale of time, a group of radial lines are plotted, showing the relation between time and distance for various speeds. It is obvious that the "net tractive effort" and "train resistance" curves apply only to one definite motor equipment and car weight; the other portions of the chart apply to any equipment. The use of the chart is illustrated by the dotted line which shows for this equipment the time and distance required to accelerate from 28 to 30 miles per hour. Starting, as shown, at the average speed of 29 miles per hour, proceed horizontally to the net tractive effort curve at *D*, then vertically to the proper grade line at *E*, then horizontally to the acceleration scale at *F* and the reciprocal curve which indicates the proper speed increment (in this case, 2 m.p.h.) at *G*, then down to the time scale at *K* and the proper average speed line (in this case, 29 m.p.h.) at *H*, then to the left to the distance scale at *I*. The points *F*, *K* and *I* indicate, respectively, a rate of acceleration of 0.9 m.p.h.p.s., an increment of time of 2.2 seconds, and an increment of distance of 0.018 mile. The chart may be used for the coasting portion of the speed-time curve by using the "train resistance" curve instead of the "net tractive effort" curve, the remainder of the procedure being the same as above. When a negative value for acceleration is indicated by a location of the points *E* and *F* below the base line, this ordinate value may be laid off above the base, and time and distance determined as before described.

Graphical Method of Plotting Speed-time Curves. The following graphical method of plotting speed-time curves is adapted from Bulletin No. 90 of the University of Illinois, by A. M. Buck.

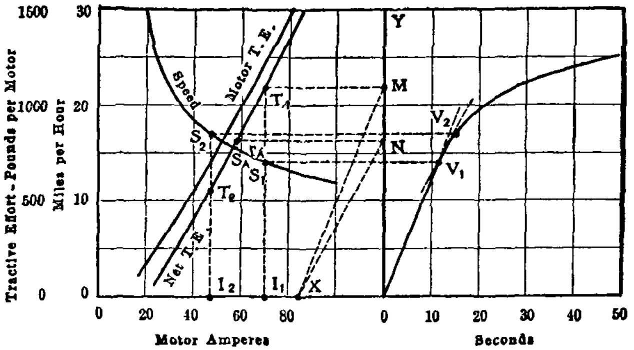


FIG. 49.—Graphical method of plotting speed-time curves.

Lay off units of current as abscissas and units of pounds tractive effort and miles per hour speed as ordinates, the latter to the scale desired for the final curve (as shown in Fig. 49). Draw the current-speed and current-tractive effort curves of the motor under consideration. Calculate and draw the net current-tractive effort curve. The net tractive effort for any current may be calculated as follows:

$$T_1 = T - \frac{fW}{n}$$

where T_1 = net tractive effort per motor, pounds
T = gross tractive effort per motor, pounds

f = train resistance at speed corresponding to assumed current, pounds per ton
W = weight of train, tons
n = number of motors in train.

At the right of the curves just drawn, erect the perpendicular OY, and lay off to the right the horizontal scale of seconds for the speed-time curve. Locate the point X to the left of O at a distance corresponding to t seconds on the time scale, determined as follows:

$$t = \frac{SW_M}{0.01098KF}$$

or, approximately:

$$t = \frac{100SW_M}{F}$$

where S = any speed, miles per hour
F = the tractive effort, pounds, the ordinate value of which is the same as that of S
W_M = weight of train, tons per motor
K = ratio of linear to total inertia (see page 150).

To construct the straight line portion of the speed-time curve, OV_1, erect a perpendicular at I_1, the average starting current, cutting the speed curve at S_1 and the tractive effort curve at T_1. Draw T_1M parallel to the horizontal axis, draw MX, and then OV_1 parallel to MX and terminating on a horizontal line from S_1 at V_1.

Should the data for the problem include the initial accelerating rate, draw XM first, at the proper slope to represent such rate. Then draw the horizontal MT_1 and the perpendicular T_1I_1, the latter indicating the starting current as I_1 and the speed at the end of straight line acceleration as S_1. OV_1 is then drawn as described above.

To construct the next increment of the speed-time curve, assume a small increment of speed, giving the new speed S_2. Determine the average net tractive effort T_A between S_1 and S_2 by erecting a perpendicular at $S_A = \frac{S_1 + S_2}{2}$. Draw T_AN and NX. Draw a line through V_1 parallel to NX, cutting a horizontal line from S_2 at V_2. Then V_1V_2 is the new increment of the speed-time curve. By assuming further increments of speed and continuing the process outlined above, a complete accelerating speed-time curve may be constructed, the accuracy of which may be made as great as desired by proper choice of the speed increments.

For the coasting portion of the curve, the ordinates representing train resistance, which are plotted down from the gross tractive effort curve, may be stepped off with dividers and transferred to the line OY (produced below the base) to determine the corresponding retardation. Speed increments may be taken as before, and the coasting curve plotted. For the braking curve an ordinate corresponding to the braking force must be obtained and added to the train resistance. In this manner the entire speed-time curve may be determined.

As Fig. 49 is drawn, it applies to level track conditions. It is evident, however, that other "net tractive effort" curves may be drawn corresponding to the average equivalent grade or to various per cents of up and down grades, so that by choosing the proper net curve for reference, the slope of a section of the speed-time curve may be determined for any grade. Fig. 49 is nothing more or less than a graphical solution of the formula for acceleration.

Location of Coasting Line when Brakes are to be Applied at a Given Speed. Construct the acceleration portion of the speed-time curve and its corresponding distance-time curve, and carry

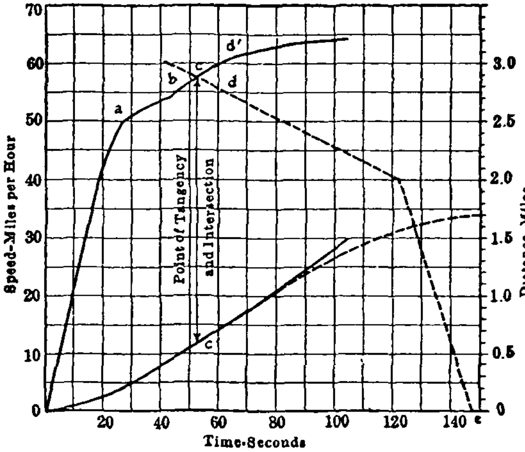


FIG. 50.—Method of joining retardation and acceleration curves.

both to a time somewhat beyond that at which it is assumed coasting will begin. These curves should be plotted on tracing paper or cloth, and will be similar to those shown by the solid lines in Fig. 50. (For Fig. 50 grades, see page 182.) Next, on a second sheet of paper, and to the same scales, begin at the end of the run and plot the braking and coasting speed-time curves up to a speed somewhat in excess of that where it is assumed coasting will begin. The distance-time curve for braking and coasting should also be plotted on the second sheet, beginning at the proper distance ordinate for the end of the run (1.7 miles in Fig. 50). The curves on the second sheet will appear as shown by the dotted lines in Fig. 50. Then place the second sheet under the first, and keeping the base lines coinciding, slide one under the other until the two speed-time curves cut one another on the same vertical ordinate as that of the point of tangency of the two distance-time curves (*cc'*, Fig. 50). The speed-time and distance-time curve to the right of this ordinate on the second sheet (dotted lines, Fig. 50) may now be traced on the first sheet and the curves are complete for the run, and show not only the exact time at which power should be cut off in order to coast down to the given speed for applying brakes, but also the total time required for the run under the assumed conditions.

Speed-time Curve for Train over Given Section of Track with Varying Grades and Alinement. When it is required to plot a speed-time curve for a train over a given track, taking into consideration the effect of definite track grades and curvature, it is convenient to divide the total distance into consecutively numbered sections, each of which is uniform throughout its length with respect to track grades and curvature. A tabulation should then be made, with the following headings (compass points are relative only).

I. Consecutive number of section (beginning at west end).
II. Length of section (feet or miles).
III. Percentage of track grade ("+" for down, "−" for up grade in east-bound direction).
IV. Degree of track curvature.
V. Track curvature expressed as "equivalent grade" (as explained below: always with "−" sign).
VI. Net equivalent grade, east-bound.
VII. Net equivalent grade, west-bound.

As the resistance due to track curves and the effect of grades are functions only of the degree of curvature and the percentage of grade, respectively, the degree of track curvature (Col. IV) may be expressed in terms of "equivalent grade" (Col. V) as follows:

$$g' = \frac{nc}{20} = \frac{0.7n}{20} = 0.035\ n$$

in which n = number of degrees of track curvature
c = track curve resistance, pounds per ton per degree of curvature. (See p. 145.)
g' = "equivalent grade," per cent, which would cause a resistance equal to the curve n.

If the values in Cols. III and V have been given "+" or "−" signs as above, their algebraic sums will give the proper values for Col. VI; reversing the signs in Col. III and adding the values in Col. V will give the proper values for Col. VII.

Using Cols. VI and VII values (multiplied by 20) for G', the formula for acceleration becomes

$$A = 0.01098K(P - f \pm G')$$

with but three variables instead of four, as before.

The speed-time curve may then be plotted as before described, but the distance-time curve must be plotted at the same time. The distance-time curve, in connection with the distance values as shown in Col. II of the tabulation, will show the time ordinate at which the acceleration formula must change by the substitution of a new value for the equivalent grade, G'. It is advisable to use the method described on p. 175 to locate the coasting and braking curve. The completed curve will be similar to those shown in Figs. 50, 55 or 58.

Chart of Acceleration Coefficients. Where a considerable number of speed-time curves are to be plotted for the same equipment, the "chart of acceleration coefficients" as proposed by C. O. Mailloux (A.I.E.E., 1902) becomes convenient. This chart, a sample of which is shown by Fig. 51, is plotted with rates of acceleration as ordinates and miles per hour speed as abscissas. Curve A_p is the solution of $A_p = 0.01098KP$ with $P = \frac{Tn}{W}$ *for the particular equipment in hand,* values of tractive effort being taken for various speeds from the motor characteristic curve. Curve A_p may be plotted between the limits of maximum possible motor current and maximum safe motor armature speed. The dotted line A_P corresponds to the current per motor (constant or not, as assumed) during the period in which it is under control by rheostat or otherwise. Curve A_f is the solution of $A_f = 0.01098K\ (-f)$ with f the value of train resistance for the particular equipment at various speeds. Curve A_f' is the same as Curve A_f reproduced above the X axis. Curve A_F is the solution of $A_F = 0.01098K\ (P - f) = 0.01098KP - 0.01098Kf$, and therefore, its ordinates are the algebraic sum of the ordinates of Curves A_p and A_f, or the ordinate distance (which may be measured with a pair of dividers) between curves A_p and A_f'. Lines are drawn above and below and parallel to the X axis, at ordinate distances therefrom equivalent to solutions of $A_G = 0.01098K(\pm G)$, and marked, as at the right of Fig. 51, with the per cent grade to which they correspond.

If track curvature be expressed as "equivalent grade," as explained on p. 176, the chart may be used for any solution of the acceleration formula $A = 0.01098K(P - f - c \pm G)$ for the particular equipment in hand. For the rate of acceleration, with motor current applied, at any speed and on any (equivalent) grade, take the ordinate value (on that speed ordinate) between curve A_F and the line for that particular grade. For the rate of acceleration during coasting, take the ordinate value between A_f and the grade line. Ordinate values measured below A_F or A_f are positive, above are negative. The speed abscissa on which curve A_F cuts any grade line indicates the speed at which $A = O$, or the maximum uniform speed which will be maintained on that grade, with current on. Likewise, the speed abscissa at which A_f cuts any grade line indicates the maximum uniform speed at which coasting will be maintained on that grade. The following solutions have been indicated on Fig. 51:

Rate of acceleration, power on, at 33 miles per hour, on 2 per cent down grade = 1.32 miles per hour per second (length of ordinate *ab*).

Rate of retardation, coasting, at 37 miles per hour on 1½ per cent up grade = 0.46 miles per hour per second (length of ordinate *cd*).

Maximum constant speed, current on, on 0.5 per cent up grade = 49.5 miles per hour (*e*).

Maximum coasting speed on 1 per cent down grade = 44.0 miles per hour (*j*).

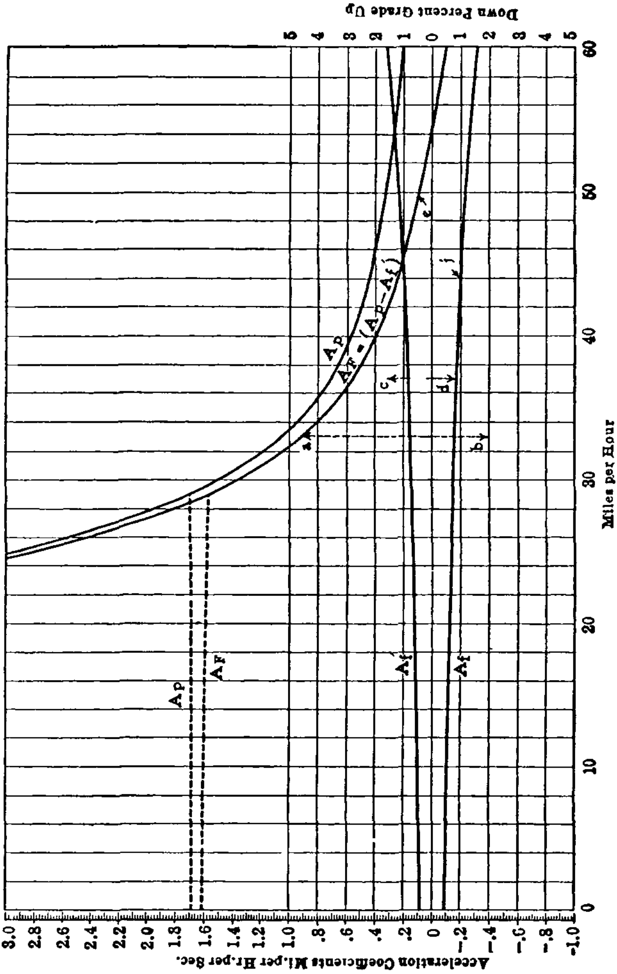


FIG. 51.—Typical chart of acceleration coefficients.

In the practical use of the chart, the ordinate value between curve A_F or A_f and a grade line is most conveniently measured by a pair of dividers, transferring the measurement to the A scale at the left of the chart. If some care be exercised in the original construction of the chart, and it be made to a sufficiently large scale, rates of acceleration may be very rapidly determined to an accuracy of $\frac{1}{100}$ mile per hour per second

Chart of Reciprocals. For use in connection with the above described chart of acceleration coefficients, Mr Mailloux proposes a "chart of reciprocals," a sample of which is shown in Fig. 52. Rates of acceleration are plotted as ordinates (to the same scale

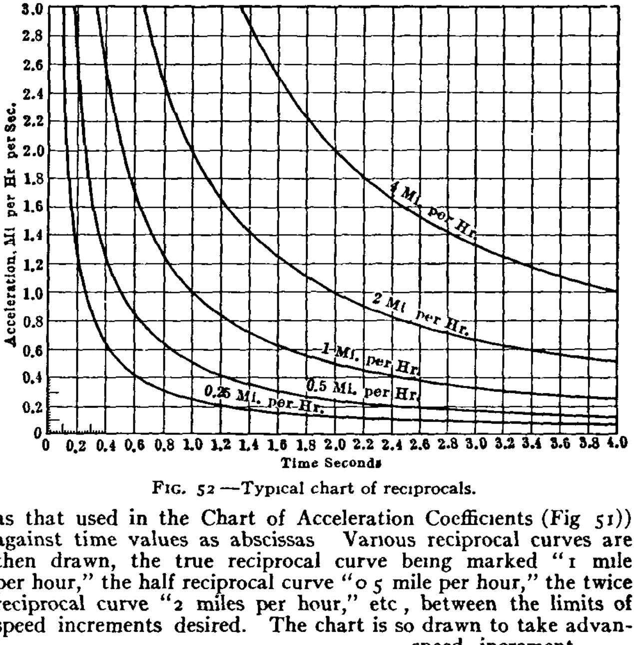


FIG. 52—Typical chart of reciprocals.

as that used in the Chart of Acceleration Coefficients (Fig 51)) against time values as abscissas Various reciprocal curves are then drawn, the true reciprocal curve being marked "1 mile per hour," the half reciprocal curve "0 5 mile per hour," the twice reciprocal curve "2 miles per hour," etc, between the limits of speed increments desired. The chart is so drawn to take advantage of the relation: time increment $= \dfrac{\text{speed increment}}{\text{acceleration}}$ and shows the time required to make a given increment of speed at a given rate of acceleration. In its practical use, the length of ordinate representing the rate of acceleration is taken off the chart of acceleration coefficients with a pair of dividers, one point of which is then moved along the X axis of the chart of reciprocals until the other point cuts the required speed increment curve; the abscissa on which this occurs indicates the time increment directly. Thus the coordinates of the terminating point of a speed-time curve increment are definitely located, obviating the possible errors likely to result from

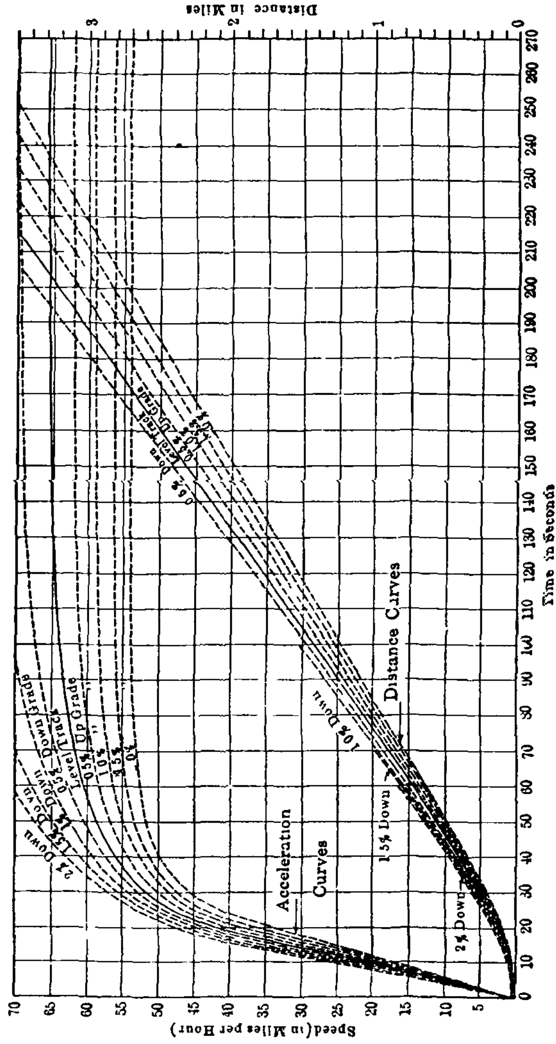


Fig. 53.—Typical chart of accelerations.

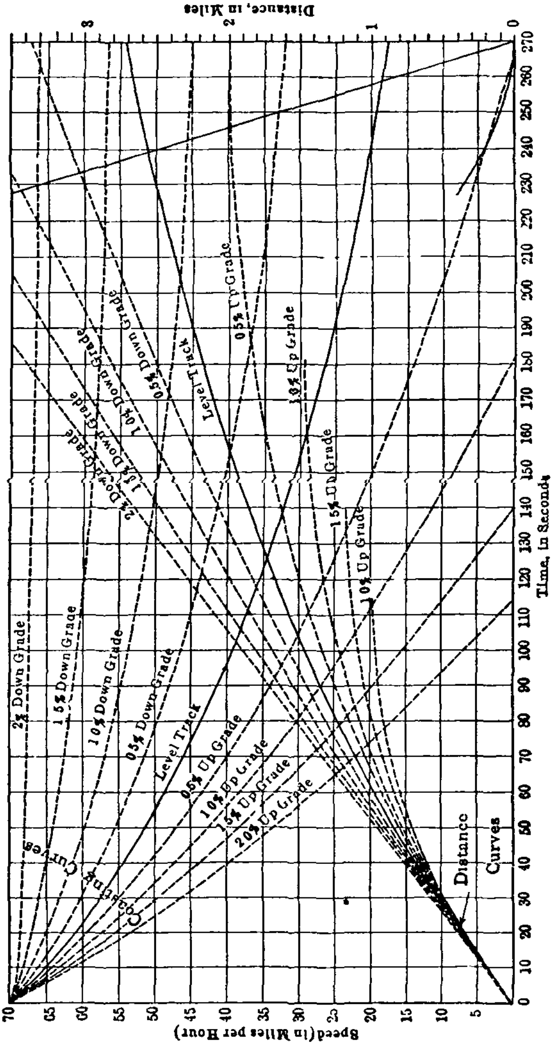


FIG. 54 —Typical chart of retardations

drawing this increment at a slope corresponding to a rate of acceleration. Obviously, the chart of reciprocals is a general one and one such chart may be used in connection with any number of charts of acceleration coefficients, provided only that all are drawn to the same scale of rates of acceleration.

Tracing Method of Construction of Speed-time Curves. Where speed-time curves are to be drawn for a considerable number of runs with the same equipment, especially with varying track grades and lengths of run, a most convenient method is one described by C. O. Mailloux (Trans. A.I.E.E., 1902). A given motor starting current having been assumed, a number of accelerating speed-time curves are drawn, for level track and various percentages of up and down grades; these are drawn to the same scales as have been chosen for the final curves; corresponding distance-time curves are also drawn on the same axes (Fig. 53). On a second sheet, to the same scales, coasting speed-time curves and distance-time curves are drawn, for level track and various grades (Fig. 54). Some care should be exercised in constructing these two charts accurately. Curves on both charts should be drawn between the limits of the maximum grades to be encountered, and a sufficient number of both accelerating and coasting curves should be drawn so that the curves for intermediate grades may be readily interpolated with sufficient accuracy when tracings are being made.

Having tabulated the uniform sections of track, as suggested on p. 176, the final speed-time curve is drawn on tracing paper, over the proper one of the two charts described. To trace the speed-time curve for any track section, move the chart under the tracing paper (with time axes coinciding) until the speed-time curve on the chart for the (equivalent) grade of the track section falls on the point of speed and time known at the beginning of that section. Trace this curve to the point where its corresponding distance-time curve indicates that the length of the track section has been covered, and repeat for the various track sections tabulated. It is advisable that the retardation portions of each run be plotted backward from the stop, as described on p. 175, and here also the same method of tracing may be used.

The final distance-time curve may also be traced from these charts, if, after the speed-time curve section has been drawn for a given track section, the chart be moved under the tracing along a vertical ordinate, keeping time axes parallel, until the proper distance-time curve coincides with the terminal point on the end of the curve as drawn for the preceding section; then trace for the proper distance.

The run curve shown in Fig. 50 was constructed by this tracing method from the charts shown in Figs. 53 and 54. In the data assumed, and given below, the breaks in grades have been made much sharper than are practicable, simply in order to bring out in a small figure the effects of changes in grade on the speed-time curve. The data assumed for Fig. 50 are as follows:

Section	Length	Grade
o–a	0 20	level
a–b	0.25	1 per cent up
b–d	0.30	½ per cent down
d–e	0.95	level

Braking to begin at 40 miles per hour.

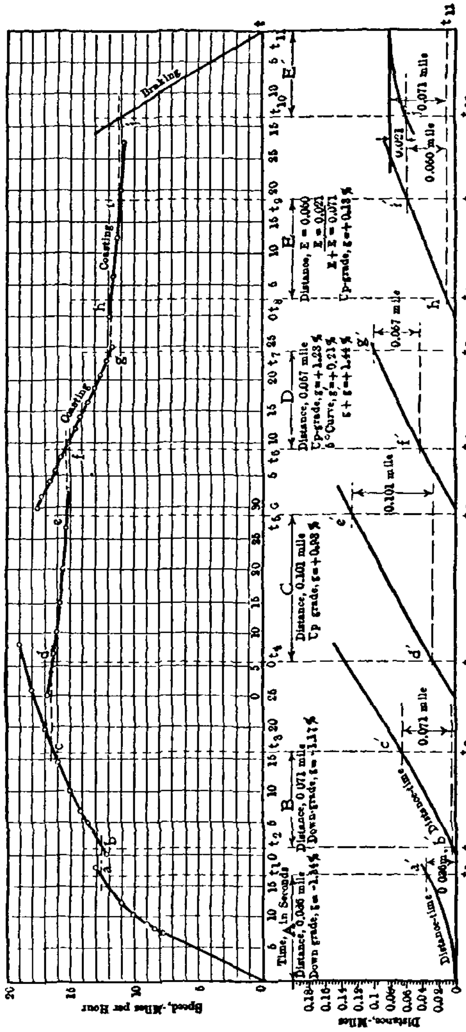


FIG. 55.—Sectionalized speed-time curves, illustrating use of chart of accelerations and chart of retardations.

The speed-time curve shown sectionalized in Fig. 55 illustrates the use of charts similar to Figs. 53 and 54 as described above. The sections *oa*, *bc*, etc., are of speed-time curves from a Chart of Accelerations similar to Fig. 53, and the sections *oa′*, *b′c′*, etc., are of distance-time curves corresponding thereto. In the final tracing, *b* will be joined at *a*, *d* at *c*, etc., in the speed-time curve, and *b′* at *a′*, *d′* at *c′*, etc., in the distance-time curve.

Distance-time Curve. The distance traveled by a train during any period of time is proportional to the area bounded by the speed-time curve for the train during that period of time, the ordinates through the extremities of that period, and the axis of time. This distance is shown by the distance-time curve (Figs. 47 and 58) plotted between elapsed time in seconds and feet or miles, on the same axis of time as that used for the speed-time curve.

Determination of Distance from Speed-time Curve. A method of obtaining the distance traveled during any elapsed time is as follows: The portion of the speed-time curve for this period of elapsed time is divided into increments of length depending upon the degree of accuracy to which the speed-time curve is drawn and the degree of accuracy desired. The distance increment corresponding to each of these speed-time increments is calculated as below, and all the distance increments for any elapsed time are then added together.

$$d = \frac{S_m(t_2 - t_1)}{3600}$$

in which d = distance increment, corresponding to any increment of the speed-time curve, miles

S_m = mean speed of train during that increment, miles per hour

$(t_2 - t_1)$ = time elapsed while that increment accrued, seconds.

The same result may be attained by the use of a planimeter, integrating the area above described, when

$$d = \frac{ats}{3600}$$

where d = distance increment, as above, miles

t = number of seconds per unit of ordinate

s = number of miles per hour per unit of abscissa

a = area, in square units.

Units may be inches, centimeters, etc., as desired, but t, s, and a must all refer to the same unit.

Graphical Construction of Distance-time Curve. The following graphical method for the construction of the distance-time curve is adapted from a description by J. G. Pertsch in the *Sibley Journal of Engineering* in 1910. The process will be described for one short section of the distance-time curve, *jk* (Fig. 56), assuming that the speed-time curve is complete and that the portion *Oj* of the distance-time curve has been drawn. Through various points *a*, *b*, *c*, etc., on the speed-time curve (including especially the points of maximum and minimum speed) draw lines parallel to *OX* and intercepting *OY* at *d*, *e*, *f*, etc. Draw lines *gj*, *hk*, etc., perpendicular to OX, and at such positions that the small triangles formed on each side of such lines by the speed-time curve and the horizontal lines previously drawn, will be equal in area. (Note that triangle m should equal triangle n, and $v = w$, but that triangles n and v are not necessarily equal.) Locate point P, to the left of O, and on OX prolonged, as follows:

Let d = any distance, miles

s = the speed, miles per hour, whose ordinate value on the speed scale is the same as that of d on the distance scale

$$t = \frac{d \times 3600}{s}, \text{ in seconds}$$

OP = value of t, on time scale.

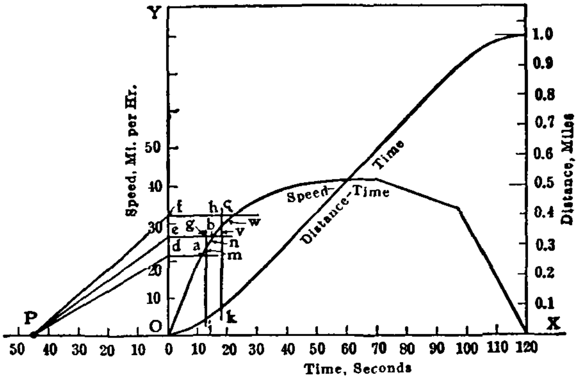


FIG. 56.—Graphical construction of distance-time curve.

Then draw the various lines Pd, Pe, Pf, etc. The average slope of the distance-time curve between the perpendiculars gj and hk will then be the same as Pe, and the portion jk may be drawn as a straight line parallel to Pe and joining the portion Oj previously drawn. The final distance-time curve will be a smooth curve drawn tangent to the short straight line sections such as jk.

Current-time Curves. A current-time curve is drawn to indicate the current per motor, current per car, or current per train at any instant. It is plotted between time values as abscissas and current in amperes as ordinates, and for convenience is constructed on the same time axis with the speed-time and distance-time curves. (For typical examples see Figs. 38 and 47.)

Current Curve for One Motor.

(a) *During the Straight Line Acceleration Period.* During straight line acceleration the average value of current will remain practically constant at the value, shown by the motor characteristic curve, necessary to give the required tractive effort for that rate of acceleration, and it usually is so plotted (as constant). Actually, the current as well as the accelerating rate will fluctuate above and below the average, inasmuch as the controlling resistance is cut out in finite steps, but it rarely serves any purpose to introduce into the curves the refinement of following these fluctuations.

(*b*) *During Acceleration Beyond the Straight Line Acceleration Period.* From the time full voltage is applied to the motor until the current is cut off, the value of the current at any instant is given by the motor characteristic curve at the speed for that instant.

Current Per Car. (Two- and four-motor equipments.) To plot the current per car the value of current at any instant (for similar motor equipments) is found from the value of current per motor at that instant, as follows:

(*a*) *Two-motor Equipment* (Series parallel control)
Series (current per car) = (current per motor)
Parallel (current per car) = 2 × (current per motor)

(*b*) *Four-motor Equipment*
Series (current per car) = (current per motor)
Series-parallel (current per car) = 2 × (current per motor)
Parallel (current per car) = 4 × (current per motor)

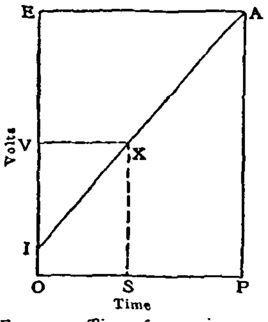


FIG. 57.—Time for series-parallel swing.

Time of Change from Series to Parallel. In series-parallel control, the swing from series to parallel should be made at the time when *the counter electromotive force plus* the drop in one motor equals one half the line voltage. When the current is being kept constant through straight line acceleration, the voltage across each motor rises in a straight line from the instant of starting when it is equal to the *IR* drop, to the end of straight line acceleration when it is equal to the line voltage. The relative times in series and in parallel may be determined as in Fig. 57,

where OE and PA = line volts
OV = ½ line volts
OI = IR drop in motor
OP and EA = time for straight line acceleration
OS and VX = time for series operation of motors
X = point where line IA cuts VX.

Current Per Train. In plotting the current per train having more than one motor car the value of the current per train at any instant is equal to the sum of the current for all the motor cars at that instant.

Potential Curve. The line potential at the car, or potential across motor terminals, is generally plotted between time in seconds as abscissas and volts as ordinates on the same time axis as is used for the current curve. The value of the potential at the car at any instant depends upon the potential of the generating station and substation, the regulation of the generator, the efficiency and

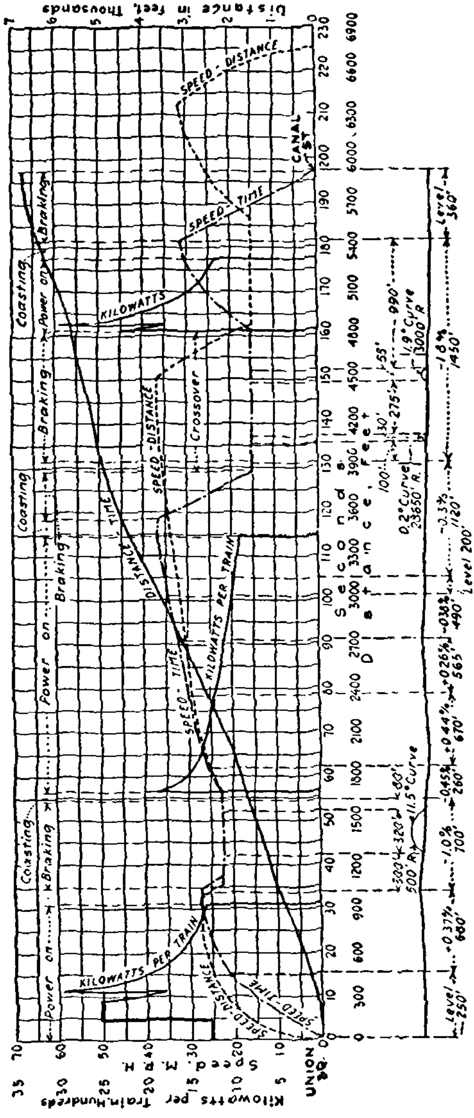


FIG. 5.—Speed-distance curves derived from speed-time and distance-time curves.

regulation of transformation, transmission, conversion and distribution, and load on the system Its value may be obtained practically only by test. For preliminary work an average voltage at the car usually is assumed to be constant; the corresponding curve is a straight line parallel to the axis of time.

Power and Power Factor Curves. The power curve may be plotted on the same time axis with other curves as above, and its instantaneous ordinate (usually to a scale of kilowatts) is at any point proportional to the product of potential and current per motor, car or train at that instant. In the case of alternating-current equipments, the power factor must, of course, be used in the determination of the power ordinate values, and in such cases, the power factor curve usually is also plotted on the same time axis, the instantaneous values being determined in connection with the motor characteristic curve, where the power factor of the motor is shown plotted against the current values.

Speed-distance Curves. Most run curves are plotted with time as abscissas, as previously described, but in some cases, such as when signal locations are being determined, the speed-distance curve, with distance as abscissas, is of great value. The construction of speed-distance curves is preceded by that of speed-time and distance-time curves, the combination of the data from which enables the speed-distance curves to be plotted, point by point. Fig. 58 shows speed-time, distance-time and speed-distance curves for a given run. It will be noted that constant rates of acceleration and braking, shown by straight lines in the speed-time curve, are represented by parabolic curves in the speed-distance curve.

Traction Power Requirements

Value of Power Input Required by Train While Moving at Constant Speed. The approximate value of power input required by a train moving at a constant speed may be obtained by the formula:

$$P = \frac{2FWS}{1000U} = \frac{FWS}{500U}$$

in which P = approximate value of power input required by train, kw.

S = constant speed of train, miles per hour

F = *total constant train, track curve and grade resistance* $(f + C \pm G)$ for the train while running at speed S, pounds per ton weight of train

W = total weight of train, tons

U = efficiency of motor equipment at the proper speed, *decimally expressed.* This may be taken from the characteristic curve for the motor equipment, or from table, page 190

$$2 = \text{approximately } \frac{5280 \times 746}{60 \times 33000}$$

NOTE: Since the value of F is always an approximation, the use of 2 instead of the more nearly exact value 1.99 for the constant is justified in this and similar formulas herein.

Value of Power Input Required by Train While Moving at Constant Speed on Tangent Level Track. When the train is moving at a constant speed on a tangent level track C and G are zero and the formula is

$$P = \frac{2fWS}{1000U} = \frac{fWS}{500U}$$

in which f = constant train resistance for the train while running at speed S, pounds per ton weight of train
P, W, S and U = same as given in paragraph immediately preceding.

The following tables show the power input required for motor-car trains and for locomotive passenger train operation, at constant speeds on tangent level track. The tables are based on the above formula for power input and the Armstrong formula for train resistance. The efficiency of geared motors at full speed is taken as 75 per cent for the motor-car trains; that of direct-current gearless motors as 90 per cent for the locomotives. Cross-section of cars is taken as listed on page 129.

TRAIN INPUT FOR CONSTANT-SPEED RUNNING ON TANGENT LEVEL TRACK, MOTOR-CAR SERVICE

(Input values expressed in kilowatts)

Train weight	Speed (miles per hr.)									
	10	20	30	40	50	60	70	80	90	100
20 ton car	6.5	16.2	32.0	56.7	93.5					...
30 ton car	8.0	19.5	38.4	67.3	109.0	167				...
40 ton car	9.4	23.1	44.1	76.2	124.0	188	276			...
50 ton car	10.4	25.6	49.2	84.8	137.0	210	305	430	584	.
60 ton car	11.5	27.9	52.8	90.2	144.0	218	316	442	599	792
2–20 ton cars	9.3	22.4	42.5	72.5	116.0					..
2–30 ton cars	11.5	27.4	51.4	87.0	137.0	206				..
2–40 ton cars	13.2	31.6	59.0	99.3	156.0	234	336			..
2–50 ton cars	14.8	35.5	66.3	111.0	175.0	261	374	520	699	.
2–60 ton cars .	16.3	38.8	71.7	119.0	185.0	274	390	540	720	945
3–20 ton cars	11.4	27.2	50.9	84.1	136.0					.
3–30 ton cars	14.0	33.3	61.8	103.0	162.0	240				.
3–40 ton cars	16.3	38.7	71.5	119.0	185.0	271	391			.
3–50 ton cars	18.4	43.7	80.6	134.0	206.0	308	437	602	805	.
3–60 ton cars	20.1	48.0	87.4	144.0	222.0	326	460	635	925	1,092
5–20 ton cars	14.8	35.3	65.4	109.0	171.0					..
5–30 ton cars	18.3	43.5	80.0	133.0	205.0	303				.
5–40 ton cars	21.3	50.8	93.2	154.0	237.0	348	493			...
5–50 ton cars	26.2	61.9	112.0	183.0	279.0	406	568	773	1,026	.
5–60 ton cars	31.3	72.8	130.0	208.0	312.0	448	6[illegible]2	835	1,100	1,415

Train Input for Constant-speed Running on Tangent Level Track, Locomotive Passenger Service

(Input values expressed in kilowatts)

Gross train weight	Speed (miles per hr.)									
	10	20	30	40	50	60	70	80	90	100
200 tons	17.5	41.2	75.8	124	196	277	398	530	710	920
300 tons	26.1	60.5	108.0	172	258	369	511	689	905	1,160
400 tons	32.0	79.0	139.0	219	330	462	635	840	1,100	1,405
500 tons	41.0	99.0	170.0	268	395	563	755	995	1,295	1,645
600 tons	50.0	118.0	203.0	315	464	645	875	1,150	1,489	1,890
700 tons	59.0	135.0	235.0	363	530	735	996	1,305	1,682	2,132
800 tons	69.0	154.0	267.0	410	596	828	1,117	1,460	1,878	2,375
900 tons	77.0	173.0	300.0	459	663	920	1,238	1,615	2,070	2,620
1,000 tons	85.0	193.0	330.0	507	730	1,011	1,358	1,771	2,261	2,860

Value of power input required by train during straight line acceleration for direct-current equipments with rheostatic control,

$$P = \frac{F_a W S}{500 U}$$

in which P = approximate average of power input required during straight line acceleration, kw.

F_a = gross tractive effort of motors during straight line acceleration, pounds per ton weight of train (see page 151)

W = total weight of train, tons

S = speed at instant when rheostat is entirely cut out, miles per hour (see page 168)

U = efficiency of motor equipment, at a load corresponding to the required gross tractive effort per motor, *decimally expressed.* See motor characteristic or table below.

Note that for the three cases of (1) parallel, (2) motors two in series and (3) motors four in series, the speed S should correspond to the instant when rheostat is entirely cut out with such motor combination. As a rough approximation, the power input in cases (2) and (3) are about ½ and ¼, respectively, that in case (1).

Approximate Efficiency of Direct-current Railway Motors

Capacity, h.p.	40	60	80	100	125	150	200	250	*250	*500	700
Max. Efficiency, per cent...	84	86	87	88	89	89	89	89	91	93	90.5
Efficiency at full speed, per cent	68	70	72	73	74	74	75	75	90	92	78.0

* Gearless.

The maximum efficiency of railway motors of the geared type occurs during maximum output, hence the values so quoted should be used for calculation of the power required to accelerate a car or train. An exception to this rule may be taken in locomotive work, where it is customary to force the motors to nearly their maximum rated output even after the train has reached its normal maximum speed.

The efficiency when the car is at full speed is generally lower with motors of the geared type, owing largely to the losses in gears and also in the magnetic circuit of the motors themselves. This lower efficiency at higher speeds does not hold true of motors of the gearless type.

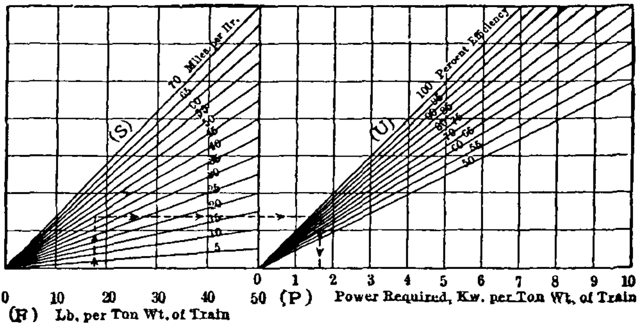


FIG. 59.—Power required per ton weight of train.

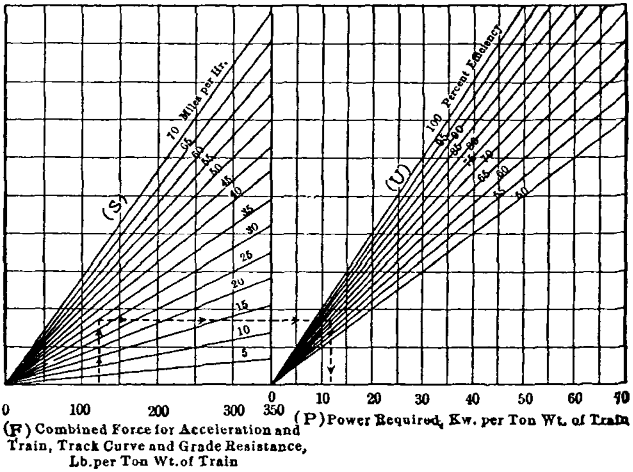


FIG. 60.—Power required per ton weight of train.

Graphical Solution of Power Equations. Figs. 59 and 60 give a general graphical solution of the equation $P = \frac{FS}{500U}$ for the amount of power required per ton weight of train. Fig. 59 is for values of F up to 50 lb. per ton. For speeds greater than 70 miles per hour use the curve for speed = $\frac{\text{(given speed)}}{\text{(a constant)}}$ and multiply the resulting kw. per ton weight of train by that constant. Fig. 60 is to be used in determining the power necessary while the train is accelerating or in other cases where the value of F is greater than 50 lb per ton weight of train.

Power Required at Substation. The average or instantaneous value of power required at the substation bus bars is equal to the average or instantaneous value, respectively, of power required at the trains taking power from that substation divided by the average efficiency (decimally expressed) of the distribution and working conductor systems connecting the trains to the substation. In approximating the power required at a given time on a given section over which the trains have regular running times and all definite stopping places, such as turnouts and stations, the train sheet or graphical time table (see p. 120) may be used to advantage as it will give the number and location of trains on the section at any time. This information together with the run curves for the trains and runs included in this section will make possible the closest approximation to the value of power required at the substation bus bars at any instant. Where trains are not operated on regular running times with definite stopping places, so close an approximation to the value of power required at any instant is impossible, but the average value of power required during any period may be approximated to a degree of accuracy depending upon the regularity, frequency and uniformity of operation over this section. The average value of power per train required may be determined from the run curve for an average train and typical run on this section. Multiplying this average value by the number of trains on the section during the given period will give the approximate average value of power required at the substation throughout that period. Any change in the headway, speed or weight of trains will tend to change the power requirements of the given section. For sections on which such changes are brought about periodically, as in putting on extra cars and adding trailers during rush hours, the power requirements may be calculated for each of the different conditions of operation.

A rough general rule is to install one kilowatt in rated substation capacity for every two horsepower in car equipments drawing more than one-half their total current from that substation, and one kilowatt per four horsepower for those cars which are taking less than one-half current from that substation.

Power Required at Generating Station. In the operation of a system without substations the value of power required at the generating station bus bars may be determined in the manner just outlined for the determination of power required at the substation bus bars. The average or instantaneous value of power at the generating station bus bars required by a substation is equal to the corresponding average or instantaneous value, respectively, of power required at the substation bus bars divided by the combined efficiency (decimally expressed) of substation, transmission from generating station to substation and generating station transformation. The average or instantaneous value of power required at the generating station bus bars by several substations is equal to the sum of the corresponding average or instantaneous power values, respectively, required at the generating station bus bars by all the substations.

A rough general rule is to allow one kilowatt of generator capacity for every two horsepower of rated motor capacity in car equipments. This rule is based on the fact that the continuous current capacity of a railway motor is about one-half of its rated one-hour capacity and that the average efficiency of a complete well-designed system is about 70 per cent or 75 per cent from power house to car, so that if a motor is operated with a fair margin, one horsepower input to the car equipment is equivalent to one kilowatt output from the generator.

Load Curves. The load curve for a substation or generating station may be approximated by plotting the instantaneous or average values of power required at the bus bars of the station against elapsed time and connecting adjacent points so plotted. If the load curves for the substations are first constructed they may be convenient in applying the method outlined in the second preceding paragraph for determining the power required at the generating station. A load curve for an existing load may be automatically drawn by a graphic recording wattmeter. A curve so drawn gives instantaneous values of power, and from it average values of power may be determined in the manner outlined for determining the average value of power required by a train.

Energy Consumption

Energy Consumption for Traction. The net energy available for traction remaining after the control, wiring, motor and mechanical transmission losses in the train have been supplied is consumed in the two following processes: first, balancing train, track grade and curve resistance; second, heating brake shoes and car wheels while braking. The first continues while the train moves, the second only while the train is in motion and brakes are applied.

Net Traction Energy Consumed in Moving Train When Brakes Are Not Applied.

$$e_r = \frac{5280 \times 745.6}{60 \times 33{,}000} FWL$$

$$= 1.99\, FWL = \text{approximately } 2FWL$$

in which

e_r = energy necessary to balance train, track curve and grade resistances while the train travels the distance considered, watt-hours

F = total average train, track curve and grade resistance $(f \pm G + C)$ for train while traveling the given section, pounds per ton weight of train

L = distance traveled, miles
W = weight of train, tons.

NOTE: Since the value of F is always an approximation, the use of 2 instead of the more nearly exact value 1.99 for the constant is justified in this and similar formulas herein.

Net Traction Energy Consumed While Train Is Coasting or While Moving with Brakes Applied. The value of the total net traction energy consumed by a train in passing with power off over a section between two given points is equal to the difference between the total values of kinetic energy of the train at these two points, that is

$$E_B = E_1 - E_2$$

in which

E_B = net traction energy consumed while brakes are applied or train is coasting, watt-hours
E_1 = total kinetic energy of train at beginning of section considered. (See E_n below)
E_2 = total kinetic energy of train at end of section considered. (See E_n below)
$E_n = 2000Wh$

in which

E_n = total kinetic energy of train at any instant n, foot-pounds
W = total weight of train, tons
h = kinetic energy head for train at the instant n, feet.

Expressing kinetic energy head in terms of speed of train (see p. 149) and reducing E_n to watt-hours,

$$E_n = \frac{2000W \times (5280)^2 S_n^2}{32.2 \times 2 \times (3600)^2 K \times 2655.4}$$

$$= \frac{WS_n^2}{39.75\,K}$$

$$= \frac{0.0252\,WS_n^2}{K}$$

in which E_n = total kinetic energy of train at any instant n, watt-hours
S_n = speed of train at that instant n, miles per hour
K = ratio of linear inertia to total inertia of train. (See p. 150)

Substituting E_n in the formula for E_B,

$$E_B = \frac{WS_1^2}{39.75K} - \frac{WS_2^2}{39.75K}$$

$$= \frac{W}{39.75K}(S_1^2 - S_2^2) \text{ or } = \frac{0.252W}{K}(S_1^2 - S_2^2)$$

in which W = total weight of train, tons
S_1 = speed of train at beginning of section considered, miles per hour
S_2 = speed of train at end of section considered, miles per hour.

Net Traction Energy Consumed in Heating Brake-shoes and Car Wheels in the Process of Braking.

$$E_b = E_B - e_F$$

$$= \frac{W}{39.75K}(S_3^2 - S_4^2) - 2FWL_b$$

or

$$= \frac{0.0252W}{K}(S_3^2 - S_4^2) - 2FWL_b$$

in which

W = total weight of train, tons

K = ratio of linear inertia to total inertia. (See p. 150)

S_3 = speed of train at beginning of braking section considered, miles per hour

S_4 = speed of train at end of braking section considered, miles per hour. When train is brought to a standstill by braking, S_4 becomes zero

L_b = length of braking section, miles

F = total average train, track curve and grade resistance $(f \pm G + C)$ for train while traveling the braking section, pounds per ton weight of train.

Approximate Formula for Energy Consumed in Heating Brake-shoes and Car Wheels in the Process of Braking. It is often assumed that while brakes are being applied the kinetic energy of rotation of the rotating parts just balances the train resistance, track curve, and grade resistances (if the latter be small). This assumption gives rise to the following approximate formula:

$$E_b = \frac{W}{40}(S_3^2 - S_4^2) \text{ or } 0.025W(S_3^2 - S_4^2)$$

When brakes are applied until train comes to a standstill, the formula becomes $E_b = \frac{WS_3^2}{40}$ or $0.025\ WS_3^2$ in which the significance of the symbols is the same as in the preceding paragraph. Fig. 61 gives a graphical solution of this formula.

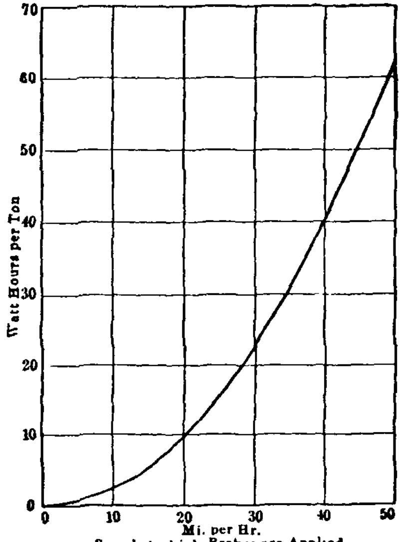


FIG. 61.—Losses in braking.

Total Net Traction Energy that must be Available for Traction while Train is Running with Power on Between Two Points. The value of this energy is equal to the sum obtained by adding the difference between the values of the kinetic energy of the train at these two points to the net traction energy consumed by the train in passing between the two points. It is expressed by the following formula:

$$e_p = \frac{W}{39.75K}(S_6^2 - S_5^2) + 2FWL$$
$$= \frac{0.0252W}{K}(S_6^2 - S_5^2) + 2FWL$$

in which

W = total weight of train, tons
K = ratio of linear inertia to total inertia of train. (See p. 150)
S_5 = velocity of the train at the beginning of the section over which the train is considered to run with power on, miles per hour. The value of S_5 is zero if train starts from rest at this point
S_6 = the velocity of the train at the end of the section over which the train is considered to run with power on, miles per hour
F = total average train, track curve and grade resistance $(f \pm G + C)$ for train while traveling the section between the two given points, pounds per ton weight of train
L = length of section between the two given points, miles. This is the length considered over which the train runs with power on.

Total Net Traction Energy Consumed during any Run.

Case 1. Train coasts to standstill, brakes not applied.

Energy consumed = e_P (See p. 193)

Case 2. Brakes applied during any part or number of parts of run.

Energy consumed, kilowatt hours = $e_P + E_B$

in which

e_P = energy consumed while brakes are *not* applied, kilowatt hours. (See p. 193.)

Where several sections over which the train runs without braking occur during the run, e_P is calculated for each of these sections and these values of e_P are added together to give the value of e_P for use in this formula.

E_B = energy consumed while the brakes are applied, kilowatt hours. (See p. 194.)

Where several sections over which the train runs with brakes applied occur during the run, E_B is calculated for each of these sections and all those values of E_B are added together to give the value of E_B.

Approximation of Energy Consumption. For preliminary determinations, approximate energy consumption may be obtained by use of the general energy output charts described below, or by

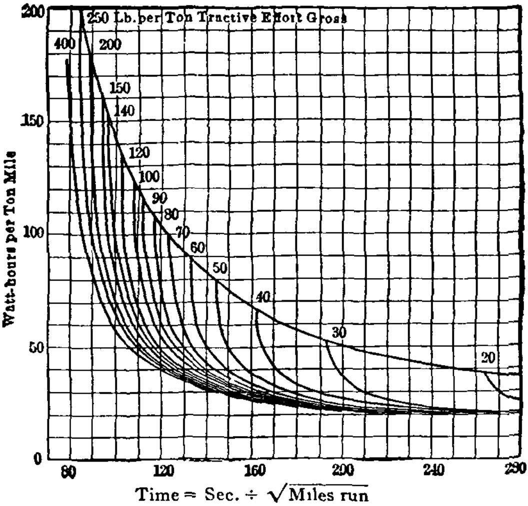


FIG. 62.—General energy output chart (train resistance, 10 lb. per ton).

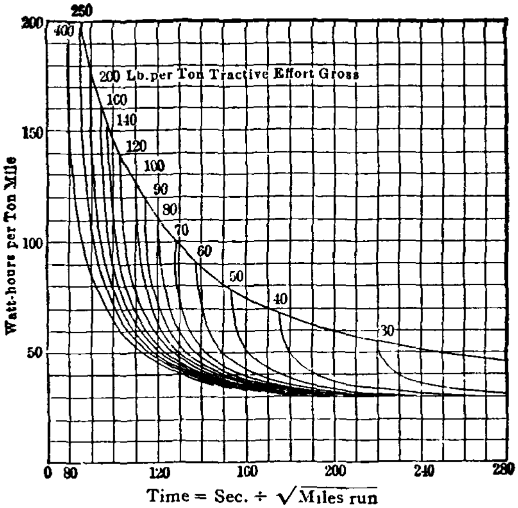


FIG. 63.—General energy output chart (train resistance, 15 lb. per ton)

use of the general operating characteristics shown in Section IV, Figs. 1–4, inclusive, and described on page 224 and following

General Energy Output Chart. The general energy consumption charts, Figs. 62 and 63, are from the Standard Handbook and are based on speed-time curves in Fig 46, and the above formula for total net traction energy. A constant braking rate of 150 lb. per ton weight of train is assumed in both, and a constant train resistance of 10 and 15 lb. per ton weight of train is used in the curves of Figs 62 and 63 respectively. These curves give approximate values of energy *output* watt-hours per ton mile, for a 1-mile run at various gross tractive efforts for acceleration and times required to make the run; these values must be divided by the efficiency of acceleration (see below) to get energy input. In using the general energy output charts the approximate value of watt-hours per ton mile is given by the ordinate of the point on the proper gross tractive effort curve, whose abscissa is equal to T_1 in the following formula. The curve selected should correspond to the gross tractive effort during straight line acceleration.

$$T_1 = \frac{T}{\sqrt{L}}$$

in which T = time to make run under consideration, seconds
L = length of run under consideration, miles

Efficiency of Acceleration. The efficiency of acceleration is the ratio of the net energy available for traction (output) to the gross energy delivered to the train in furnishing that net energy. It is usually expressed as a per cent. It depends upon the efficiency of control, efficiency of car wiring, motor efficiency, efficiency of mechanical transmission, and the per cent of straight line acceleration (see p 199) *obtaining during the particular run under consideration.*

Efficiency of Acceleration with Series Motors and Three-phase Induction Motors. The following table from data given in the Standard Handbook gives the per cent efficiency of acceleration at various per cents straight line acceleration for direct current series motors, also for three-phase induction motors (connected in parallel)

Per cent straight line acceleration	Rheostatic control, per cent.	Series parallel, per cent.	Four series, two series, parallel, per cent.	Three-phase induction motors, parallel, per cent.
100	43	56 0	60 5	40
90	46	59 0	62 5	43
80	49	62 0	65.0	46
70	52	64 0	67 0	49
60	56	67 0	69 0	52
50	58	69 0	71 0	55
40	62	71 0	72 5	58
30	65	72 5	73 5	61
20	68	73 5	74 0	64
10	72	74 5	74 5	72
0	75	75 0	75 0	75

Efficiency of Acceleration with Single-phase Commutating Motor Equipments. The efficiency of acceleration for trains equipped with single-phase commutating motors may be taken at 70 to 75 per cent. This value is a close approximation for the class of service to which such equipments are most often used, *i e*, an express or semi-express service on private right of way, with necessarily small per cent of straight line acceleration.

Per Cent Straight Line Acceleration. The per cent straight line acceleration of a train on a given run is the ratio of the time during which the motors take constant current to the total time current is supplied to the motors, expressed in per cent. The per cent straight line acceleration for runs of greater than a mile in length may be less than 10; on shorter runs it is greater and may even exceed 75 for very short runs. It may be taken directly from the speed-time curves for any particular case.

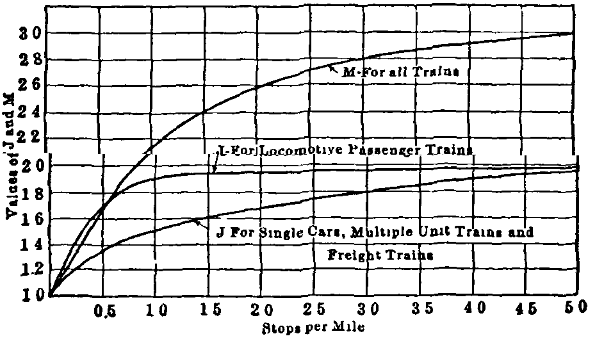


FIG. 64.—Empirical constants for energy approximation (Del Mar-Woodbury).

Approximate Determination of Energy Consumption. The following method and constants for determining the approximate amount of energy consumed in train movement are by W. A. Del Mar and D. C. Woodbury.

Significance of symbols.

S = average running speed, excluding stops, miles per hour
S_m = maximum speed, miles per hour
L = distance between stops, miles
L_p = distance traveled with power on, miles
$N = \frac{1}{L}$ = number of stops per mile including one terminal
f = average train resistance, in pounds per ton (say that corresponding to a speed of 10 to 20 per cent greater than the average speed)
G = average equivalent grade, per cent (see page 163)
D = average curvature, degrees

U = average motor and controller efficiencies as a decimal, which is usually about 0.7

$$J = \frac{S_m}{S} \text{ (see Fig. 64)}$$

$$M = \frac{L}{L_p} \text{ (see Fig. 64)}$$

OUTPUT AT WHEEL RIM AND INPUT TO CAR IN WATT HOURS PER TON MILE

	Actual energy output at wheel rims of cars	Approximate electrical energy input to cars
Due to kinetic energy	$\frac{S_m^2}{36.2L}$	$\frac{J^2NS^2}{25}$
Due to train resistance.....	$\frac{1.99fL_p}{L}$	$\frac{2.85f}{M}$
Due to grades..........	$\frac{39.8GL_p}{L}$	$\frac{57G}{M}$
Due to curves	$\frac{1.99DL_p}{L}$	$\frac{2.85f}{M}$
Total.............	Sum of above	Sum of above

In this table the first column gives the mechanical energy required at the wheel rims. In the second column an allowance of 70 per cent for efficiency of motors, gears and control has been made. (For values of J and M see Fig. 64.) In cases where S_m and L_p are known, more accurate results can be obtained by using them in the formula of the first column.

Example: A multiple-unit train makes an average speed, while running, of 30 miles per hour, making a stop every 2 miles. The average train resistance is 7 lb. per ton, the average equivalent grade 0.05 per cent, and the average curvature 0.75 deg.

Then $S = 30$
$f = 7$
$G = 0.05$
$D = 0.75$
$N = 0.5$
$\left.\begin{array}{l} J = 1.36 \\ M = 1.67 \end{array}\right\}$ from Fig. 64.

Then, in watt-hours per ton mile,

$$\text{Kinetic energy} = \frac{1.36^2 \times 0.5 \times 30^2}{25} = 33.3$$

$$\text{Train resistance energy} = \frac{2.85 \times 7}{1.67} = 12.0$$

$$\text{Energy for grades} = \frac{57 \times 0.05}{1.67} = 1.7$$

$$\text{Energy for curves} = \frac{2.85 \times 0.75}{1.67} = 1.3$$

Total energy input at train = 48.3

Energy Consumed in Rheostat. The energy consumed in the series motor control rheostat and appearing as heat therein depends upon (1) the voltage drop across the rheostat, (2) the current flowing through the rheostat and (3) the time during which this current continues to flow.

Below are given approximate formulas for calculating the energy consumed in the series motor control rheostat during start. Cases 1 to 3, inclusive, are for equipments arranged to give a uniform rate of straight line acceleration. Case 4 is for equipments arranged to give two rates of acceleration, viz., one rate of acceleration while half the motors are in series with the other half, and another rate of acceleration while all the motors are in parallel. (See p. 155.)

Since the resistance is arranged in steps, each of a finite value, the value of the current will fluctuate above and below the average as the controller is advanced and the motor speeds up. When the acceleration approximates a constant rate, the mean of these fluctuating values of current will approximate the value of current necessary to accelerate the train at the constant rate.

For convenience in using these formulas, they are given for rheostat energy consumption *per motor.* The second formula under Case 2 is an exception. It gives an approximation to the loss per ton weight of train.

Significance of Symbols.

e_R = energy consumed in rheostat during one straight line acceleration period, watt-hours *per motor*

T = duration of straight line acceleration, seconds

I = constant or average current *per motor* during period of straight line acceleration, amperes

E = working conductor (trolley or third rail) potential at the motor car or locomotive, volts

R = resistance of the motor, ohms. (See pp. 254–259.)

Case 1. *One or more motors, rheostatic control.* All motors in parallel throughout straight line acceleration period.

$$e_R = \frac{IT(E - IR)}{2 \times 3600}$$

Case 2. *Two- and four-motor equipments, series-parallel.* Two-motor equipments having motors in series throughout first part of straight line acceleration period and in parallel throughout the remainder of straight line acceleration period. Four-motor equipments having two groups of two motors in parallel in series throughout the first part of the straight line acceleration period and all motors in parallel throughout the remainder of the straight line acceleration period.

$$e_R = \frac{IT(E^2 - 2EIR + 2I^2R^2)}{4 \times 3600\,(E - IR)}$$

or (approximate formula), $L = \dfrac{AVT}{3600W}$

in which

L = approximate loss in watt-hours per ton
A = current *per car* with series position of controller
V = ½ line volts − drop in motor
T = time of straight line acceleration in seconds
W = weight of car in tons.

Case 3. *Four motors, series, series-parallel, parallel.* All motors in series throughout first part of straight line acceleration period, two groups of two motors in parallel in series throughout the second part, and all motors in parallel throughout the remainder of the straight line acceleration period.

$$e_R = \frac{IT(3E^2 - 4EIR + 8I^2R^2)}{16 \times 3600(E - IR)}$$

Case 4. *Two rates of straight line acceleration* (see p. 155), *i.e.*, two values of constant current per motor during straight line acceleration. Arrangement of motors same as Case 2.

Significance of Symbols, Case 4.

e_R = energy consumed in rheostat during one straight line acceleration period, watt-hours *per motor*
T_1 = duration of first rate of straight line acceleration, seconds. (Series connection)
T_2 = duration of second rate of straight line acceleration, seconds. (Parallel connection)
I_1 = constant current *per motor* during period (T_1) of first rate of straight line acceleration, amperes (series connection)
I_2 = constant current *per motor* during period (T_2) of second rate of straight line acceleration, amperes (parallel connection)
E = working conductor (trolley or third rail) potential at the motor car or locomotive, volts
R = resistance of the motor, ohms. (See pp. 254–259.)

$$e_R = \frac{I_1T_1(E - 2I_1R) + I_2T_2E}{4 \times 3600}$$

Energy Consumption Reduced by Use of Grade at Approach to Station. A considerable saving of energy may be brought about by causing a train to lift itself, at a station, to an elevation greater than its elevation between stations. By this process much of the energy that would be dissipated as heat in brake-shoes and car wheels in bringing the train to a stop while maintaining a practical schedule speed on a 'evel track is converted into potential energy available to aid in accelerating the train while it departs from the station. This method of conserving energy is especially applicable to operation on elevated railways and subways, where the track grades are to a greater extent under the control of the engineer than is generally the case with the surface railway. Fig. 65 shows the speed-time curve and energy-consumption curve for a 154-ton train on a typical average run of the Manhattan Elevated system, N. Y., and the corresponding data for the same train, with lighter motors as explained further on, running the same distance on a track having a 3 per cent up grade at the approach to the station, and a 3 per cent down grade at the departure from the station. It is to be noted that the train operates at the same schedule speed in both cases, but in traversing the run having the grades it consumes much less energy than it does in traversing the all level run. The rate of straight line acceleration is the same in both cases. In this example, the 3 per cent grade at departure from the station furnishes a force of approximately 9240 lb. of the 18,620 lb. (if we neglect rotational inertia) necessary to accelerate the train at the required rate of 1.33 miles per hour per second. The motors are then called upon to furnish an accelerating force of approximately 18,620 − 9240 = 9380 lb. or about half that necessary on the all level run, so they need have only about 50 per cent the capacity required for the all

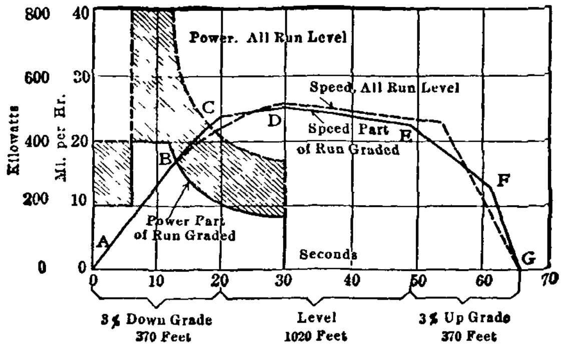


FIG. 65.—Example of energy saving by use of grades at station stops.

level run. In constructing the power curve, Fig. 65, for the train operating on the run having the grades, it was assumed that the capacity of the motors and consequently their torque at any speed are one-half those for the all level run. A comparison of the power-time areas indicates that in this case operation on the run having the grades demands only about half the energy necessary for operation on the all level run.

Regenerative Braking. It has been shown that the energy loss in heating brake shoes and wheels forms a considerable part of the total required for traction. Inasmuch as the electric motor is reversible, that is, can give out electric power when mechanically driven as a generator, it is possible, with the proper control, to brake a motor car or electric locomotive electrically, or make use of "regenerative braking," with reduced expense of brake shoe and wheel maintenance, besides returning to the line a considerable portion of the energy stored in the train during acceleration. Opposed to the advantages of regenerative braking, especially as applied to single car and rapid transit operation, are several facts. Some arrangement of shunt winding or separate excitation must be applied to the common direct current railway motor, which is of the series type, to enable it to act as a generator. Further, the thermal capacity of the motor must be increased due to the extra motor losses entailed in electric braking and the less proportionate amount of time with no load which may be utilized for cooling. The additional weight of the excitation and control equipment often will further increase the required capacity of the motors, and will add considerably to the first cost. Even in frequent stop service, not much more than 15 to 20 per cent of the gross input can be saved by regenerative braking, which may be more than balanced by the first cost and cost of maintenance of the necessary increased motor capacity and auxiliary apparatus. As any system of regenerative braking depends upon the counter e.m.f. of the revolving armature, it is evident that when approaching zero speed there will be no torque developed by the motor, and hence, regenerative braking must be used in connnection with air brakes, with resultant decreased economy. For street car and rapid transit service, including interurban service, it can be said that there is small attraction in regenerative braking, not because it is not possible, but because the added complications and expense bring no adequate return in the cost of the energy saved.

Regenerative braking used in connection with locomotives for heavy passenger or freight service on mountain-grade sections offers many advantages far outweighing in importance the possibility of a small amount of energy saved. One of the disadvantages attending the operation of long heavy trains on mountain grades is the danger incurred in braking trains on down grades. More accidents occur when trains are running down grade than when operating up grade, due to the possibility of overheated brake shoes and car wheels and also breaking apart of long trains when brakes are released momentarily for the purpose of recharging the train pipe. In such service, a regenerative system of control offers safer means of holding trains on down grades than exists with air brake control, and its claims in this direction make it worthy of very careful consideration for this class of service. The advantages have been listed as follows: saving of approximately 15 per cent of total power required; elimination of brake shoe and wheel wear and brake rigging troubles with material reduction in maintenance charges; removal of difficulties encountered in operation of long heavy freight and passenger trains on long grades due to inherent operating characteristics of air brakes; reduction in wear of track on grades and severe curves; increased safety to passengers and train crews due to duplicate braking systems; increased comfort to passengers and reduced wear on equipment due to constant speed on grades and uniform braking when slowing down for curves and stops.

Direct Current Regeneration. The system of regeneration used with direct current railway motors by the Westinghouse Electric & Manufacturing Company is diagrammatically shown as in Fig. 66. This figure shows the armature of the main motor, *A*, the main motor field, *F*, the balancing or stabilizing resistor, *R*, and the exciter, *E*. Inherently, a series direct current generator has no stability of voltage, and some external means is necessary to control it when regenerating. In the figure, the exciter, E, furnishes current which flows in the path indicated by the dotted arrows. This current excites the main motor field and causes the armature, A, to generate a voltage opposed to the line voltage. The voltage of E can be regulated either by regulating its field strength or by varying resistance in series with its armature. When the voltage of A overcomes the line, a current starts to flow from the ground through R and A to the line. Thus R carries both the main regenerated current and the exciting current.

The stabilizing effect comes from the resistance R, since any variation in the regenerated current will vary the drop across the resistance and thus, assuming the voltage of the exciter, E, to be fairly constant, will have an immediate effect on the field of the main motor to counteract such a variation. For instance, if a train is regenerating down grade with the exciter voltage set so as to give the proper excitation to the main field to furnish the necessary braking effort, and the line voltage should suddenly drop, the regenerated current would immediately tend to increase. However, the increased current which tends to flow increases the voltage drop across the resistance R, and directly reduces the exciting current, thus maintaining a balanced condition as before. It is important in a direct-current regenerating system to have this regulating influence as nearly direct as possible, since changes in line voltage may at times be so great and so sudden that unless the regulation is simultaneous with the variation, a condition of unbalance between the armature and field of the motor may be sufficiently great to cause flashing at the commutator.

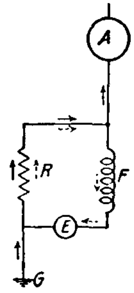


FIG. 66.—Direct current regenerative breaking.

The excitation for the main motors is supplied by the separate generator, E, driven either by a motor or from an idle locomotive axle through gearing. The regulation of the braking is then obtained by varying the exciter field strength or by varying the amount of resistance in the stabilizer circuit.

Fig. 67 shows the relation between speed, retarding effort and amperes regenerated, plotted on the basis of constant line voltage. Miles per hour and amperes regenerated are both plotted as ordinates against retarding effort as abscissas. The curves shown are for a 3000 volt direct-current locomotive which has six 1500 volt driving motors that regenerate in three different speed combinations. The performance for the intermediate speed combination (of two parallel circuits of three motors each) is shown in Fig. 67. The numbers on the curves refer to the different control notches, the control in this case being accomplished by varying the stabilizing resistance in steps. As long as the controller is on a certain numbered notch, the correspondingly numbered speed and current curves indicate the regenerative performance. The kilowatts returned to the trolley may be found for any point by multiplying the amperes regenerated by the trolley voltage. It will be noted that the general shape of the speed curve is upward as the retarding effort increases. This is just the reverse of the series motoring speed characteristic of the same motor, and is necessary for a stable condition of regeneration. As an example, suppose that the locomotive is regenerating on a down grade at 20 miles per hour with the controller on notch No. 9. The speed curve shows that a retarding effort of 18,000 pounds is being exerted; the current curve shows that 200 amperes is being returned to the line, and this current, multiplied by the trolley voltage of 3000, indicates that the regenerated power is 600 kw. Should the grade become steeper and the speed increase, the retarding effort will also increase

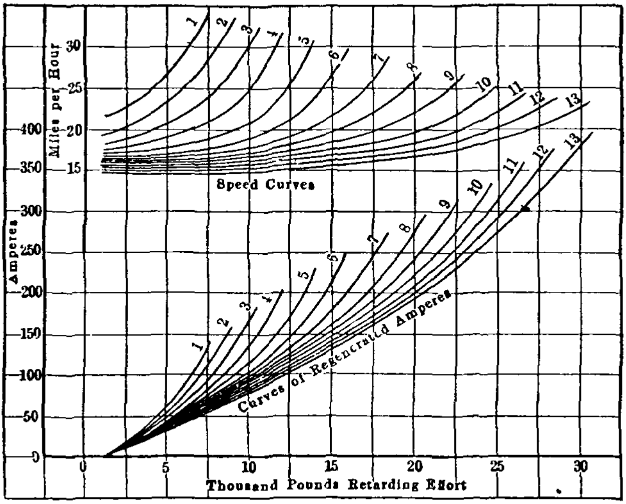


FIG. 67.—Relation between speed retarding effort and regenerated amperes on various control notches in direct current regenerative braking.

until a balanced speed is again reached. The shape of the curve is also an indication of the regeneration performance. A flat curve, or one which shows a gradual increase of speed with increase of retarding effort, indicates that ordinary changes in retarding effort will not necessitate a change of the controller handle from one notch to another, unless a change in speed is desired. A steep curve, however, or one which shows a rapid increase of speed with increase of retarding effort, indicates that any increase in retarding effort required will necessitate movement of the controller handle, if the same speed is to be maintained. The curves corresponding to the high-speed notches will be found to be steeper than those for the low-speed notches. This is because at the high-speed notch the value of the stabilizing resistance is the greatest, and with a large value of stabilizing resistance, small changes in retarding effort represent large changes in speed. This is due in turn to the fact that, with a large value of stabilizing resistance, a small change in regenerated or armature amperes produces a large change in field amperes.

In any direct current system of regeneration, it is not safe to allow the armature current to become too great in proportion to the field current, or the armature reaction may distort the field form to the extent of causing a flash-over. In Fig. 67 the upper end of the curves are discontinued at the point where the ratio of armature amperes to field amperes becomes 2.5 to 1. This ratio is determined from experience, and depends on the ratio of the armature turns to the field turns and on the individual flashing characteristics of the motor used.

If the per cent efficiency is desired for any point on any curve, it may easily be found as follows:

$$\text{Per cent Efficiency} = \frac{\text{output}}{\text{input}} = \frac{\text{Line volts} \times \text{amperes regenerated} \times 503}{\text{Miles per hour} \times \text{pounds retarding effort}}$$

The curves of Fig. 67, as mentioned, refer to only one of three *regeneration combinations. The higher speed combination makes* use of three parallel circuits of two armatures each from trolley to ground. The lower speed combination uses a single circuit of six armatures in series.

Alternating Current Regeneration. Very successful installations of regenerative electric braking have been made in connection with three-phase or split-phase electrifications using induction motors on the locomotives, such as the Great Northern Railroad's three-phase installation at the Cascade Tunnel, the Italian State Railways in Italy, and the split-phase electrification of the Norfolk & Western Railroad. The three-phase induction motor characteristics are such that it lends itself very readily to this method of control, as it is only necessary to operate the motor slightly above the synchronous speed to cause it to work as a generator; but it has the serious inherent disadvantage that electric braking cannot be obtained at any other speed. The regenerative curves of a 115 ton 6600 volt three-phase freight locomotive are shown in Fig. 68.

Effect of Variation in Line Voltage on D.C. Motor Operation. The speed of a series motor at any given current is directly proportional to the counter e.m.f., that is, to the impressed voltage less the drop in the motor itself. The line voltage impressed on a railway equipment has, therefore, an important bearing on the performance of that equipment. Voltage affects schedules, energy consumption and motor heating. The effects of low voltage are for the most part undesirable, and hence demand serious consideration.

The effect on schedule speed is the most readily observed feature of low voltage. It is well known that in a specific service a lower voltage involves a lower schedule speed, or a reduction in margin for making up lost time, or both. If service conditions are such that an equipment is able at a given voltage to make a certain length of run without coasting, it is found that an increase in voltage produces a comparatively small speed margin on very short runs, while on long runs for the same voltage increase the speed margin may be relatively great. This indicates that for the reasonable maintenance of schedules, it is necessary on suburban and interurban lines to maintain better line voltage conditions or apply equipments with greater leeway in speed than would be necessary on a city line. Since the maintenance of good voltage is relatively more difficult and expensive on interurban than on city lines, the alternative of higher speed equipment is usually adopted.

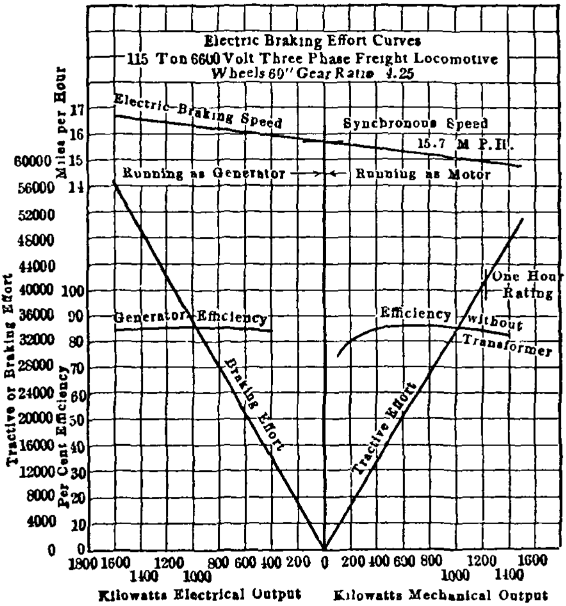


FIG. 68.—Performace of three-phase induction motors in regenerative braking.

On every run, the time during which current is drawn by the motors is made up of two distinct parts, the period of straight line acceleration and the period of running with the controller on full. The speed of the car before the controller is on the full parallel position depends on the rate of straight line acceleration, and is to a certain extent independent of the line voltage. The more marked effect of the change in voltage on the speed margin on longer runs is due to the fact that the motors are running at line voltage a larger part of the time.

The effect of voltage on the operation of car equipments will be examined from two standpoints: first, where the schedule can be adjusted to suit the voltage, and second, where the schedule must be maintained irrespective of voltage. In the first case, the schedule, energy consumption at the car and the motor heating will vary in the same direction as the line voltage. For example, as shown by Fig. 69, under certain assumed conditions and with a typical railway equipment, it has been found that on the basis of constant speed margin, a reduction in line voltage from 550 to 450 results in a reduction in schedule speed of 5 per cent for an average run of 1000 ft., and a reduction in schedule speed of 10

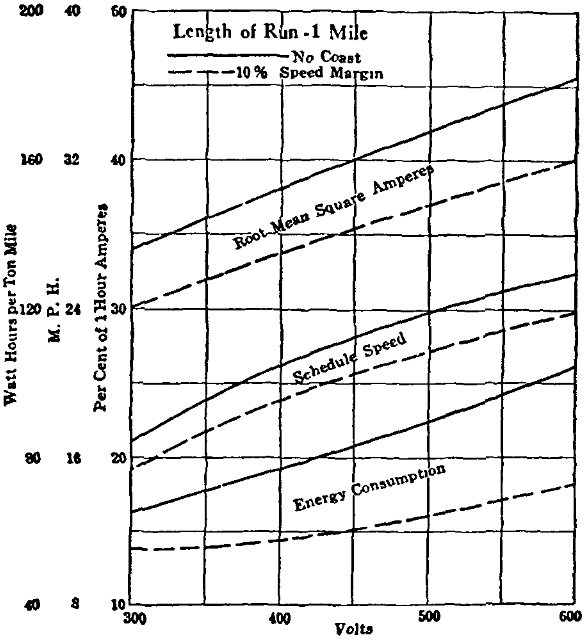


FIG 69.—Relation between voltage, energy consumption and schedule speed in a given run.

per cent for an average run of 1 mile. These percentages are approximately constant so long as the comparison is made on the basis of the same percentage of speed margin for both voltages. Under these same conditions, the heating current is reduced about 9 per cent, both on a 1000-ft. run and on a 1-mile run, and this percentage varies but little for different speed margins so long as the comparison is made on the basis of a definite speed margin. Energy consumption is reduced 17 per cent at the car on a 1000-ft. run, and 13 per cent at the car on a 1-mile run, the percentage being slightly greater when the comparison is based on runs without coasting than when based on runs with sufficient speed margin. This reduction in energy consumption at the car is due almost entirely to the reduction in schedule speed. If the lower voltage were due to drop in trolley and track circuits, the energy consumption measured at the substation would increase by 13 per cent on the 1000-ft. run and 5.7 per cent on the 1-mile run.

As an example of the conditions prevailing where the schedule must be maintained, irrespective of voltage, it has been found that an increase in generator voltage from 450 to 550 results in a 28 per cent increase in percentage speed margin on a 1000-ft. run and a 32 per cent increase on a ½-mile run. The same increase in voltage results in a 12 per cent decrease in energy consumption at the car on a 1000-ft. run and a 20 per cent decrease at the car on a ½-mile run. If the lower voltage is due to drop in the trolley and track circuit, then with the higher voltage at the car, the energy consumption measured at the substation decreases 28 per cent on the 1000-ft. run and 35 per cent on the ½-mile run. The energy consumption decreases as the voltage increases because with higher voltage the car accelerates more quickly, more coast is obtained, and the brakes are applied at a lower speed. The lower energy consumption is the result of less loss in the heating of wheels and brake shoes. Under any given set of conditions there is one voltage giving a minimum heating current, which will increase if the voltage is either raised or lowered.

Fig. 70 shows the variation in heating current, energy consumption and speed margin for several lengths of run from 0.1 mile to 2.5 miles, with voltages from 300 volts to 600 volts. For the four short runs, the schedule is such as can be made at 300 volts without coasting. The short runs are taken on that basis since the comparison can be made over the entire range of voltage chosen. On the two long runs, however, at higher voltages the car makes the distance and coasts to a standstill in less time than required for the run without coast at 300 volts. In order to obtain a wider range of comparison, the schedule is taken as that which can be made at 600 volts when coasting to a standstill without any braking.

While the effect on speed margin and energy consumption is marked, the effect on heating current is comparatively slight. The ability of a car to maintain schedules under varying track conditions as measured by the speed margin and the energy consumption are of primary importance to the operator. The maintenance of an equipment is affected to a certain extent by the temperature rise obtained on the motors in their average service. It is doubtful, however, if the degree to which motor heating is increased by reasonably increased voltage will ever be reflected in any maintenance reports, provided running time is kept the same. With higher voltage, the coasting time is longer, and brakes as a result are applied at a lower speed. This reduces the wheel and brake shoe wear and tends to counterbalance the theoretical increase in motor maintenance due to higher temperature.

Low voltage will often cause the overheating of a motor equipment on long grades. On account of the lower speed resulting from decreased voltage, the time on the grade may become sufficiently long to exceed the motor's thermal capacity, overheat the brushes, or otherwise produce poor commutating conditions. This is particularly true of locomotives, since they generally will be more heavily overloaded on grades and operate at lower speeds than motor cars.

With extremely high resistance in trolley and track circuit, it is sometimes possible to obtain a higher speed with the motors in

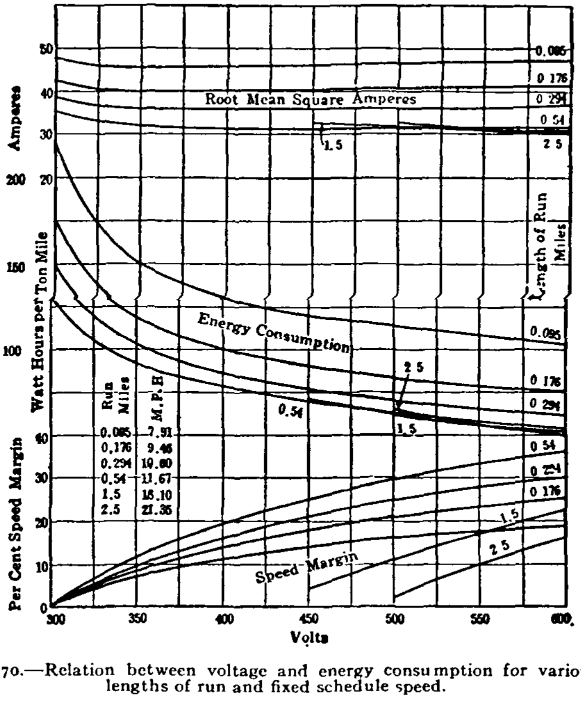


FIG. 70.—Relation between voltage and energy consumption for various lengths of run and fixed schedule speed.

series than with the motors in parallel. Assume as an example the following somewhat abnormal conditions: A single track section of road laid with 60-lb. rails and fed from one end by a single No. 0000 copper trolley wire. The total resistance per mile would be 0.305 ohm. If the substation voltage is 625 volts, and a car with quadruple motor equipment at the end of the line, 5 miles from the substation, is drawing 70 amp. per motor, the voltage at the car with motors in parallel is 198 volts. The voltage at the car with motors in series-parallel is 412 volts, and the voltage per motor

206. The voltage per motor is, therefore, in the example chosen, 8 volts more with the motors in series-parallel than with the motors in full parallel, and the speed of the car would be 5 per cent higher. With any higher current per motor the voltage drop would be greater, and the difference in the voltage per motor with motors in series-parallel and with the motors in full parallel would increase as the current per motor increases.

Effect of Coasting. With a given equipment on a run of a given distance in a given time from start to stop, an increase in the rate of acceleration, the rate of braking remaining the same, will (*a*) reduce the time of running on resistance, which reduces the resistance energy loss, and (*b*) reduce the speed at which braking begins, which reduces the energy lost in braking; the average speed remains the same, and the maximum speed is changed little if any, therefore the energy expended in overcoming train resistance is changed little if any. Similarly, an increase in the rate of braking, the rate of acceleration remaining the same, will reduce the speed at which braking begins and thus reduce the energy lost in braking. Either or both of the changes, in rate of acceleration or in rate of braking, will increase the coasting time, so that in general it may be said that within the physical limitations of the equipment, the energy consumption becomes less as the coasting time is increased.

Devices for Checking Motormen. It is upon the principle outlined above that the operation of the Coasting Time Recorder is based. The instrument records the time during which the car is coasting, and is one of the devices in common use for checking and comparing operation of motormen. Another such device is the Current Time Recorder, which records the time during which current is applied to the motors, the most economical run being assumed to be that in which the current-on time is the shortest, as such result presumably is obtained only by increasing the rates of acceleration and braking to the maximum. The comparative operation of motormen, as well as that of the car equipment, is more directly indicated by the readings of a watthour meter connected in the motor circuit. This directly measures the energy (which it is desired to save), and is further used by many operators as an indicator of inspection time, the various parts of the equipment being inspected on a kilowatt hour rather than on a mileage basis. The Economy watthour meter is of the mercury flotation type, reducing to a minimum the troubles incident to carrying such meters on moving cars. When used as an inspection indicator, it is equipped with auxiliary inspection dials, one of which may be used as an indicator for brakes and controllers, one for oiling, and one for general inspection. The auxiliary dials are arranged for resetting to zero after equipment inspections, and are equipped with markers which may be set to indicate proper inspection intervals.

The indications of any of the devices named, and particularly the watthour meter, serve as a check as to the condition of equipment, giving warning of such troubles as short-circuited motor fields, too close brake shoe adjustment, etc. The saving in energy resulting from the use of one or another of these devices has amounted to from 10 to 25 per cent of the total, on numerous roads.

Train Operation. The full realization of the advantages of multiple-unit operation can be obtained only on roads having ample substation capacity and a liberal feeder system, although the requirements of these items can be kept within very reasonable limits by the proper training of the motorman. Where multiple-unit trains are employed, it is usually not necessary to have either cars or equipment as large as if they were operated singly at all

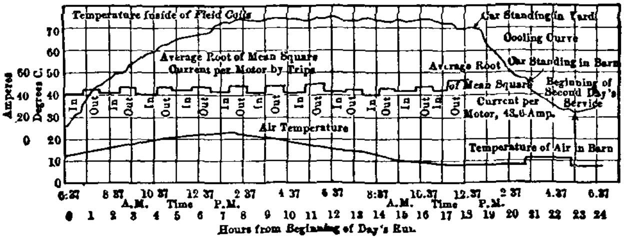


FIG. 71.—Motor temperature, single car.

times. The predecessor and principal competitor of the multiple-unit train is the combination of the motor car and trailer. If the motors are of sufficient capacity, and especially if two-car trains are to be operated for only short periods of time, trailer operation may be more economical than multiple-unit operation. Care should be taken that in such operation the motors are not overloaded since the excessive heating may do them damage. The

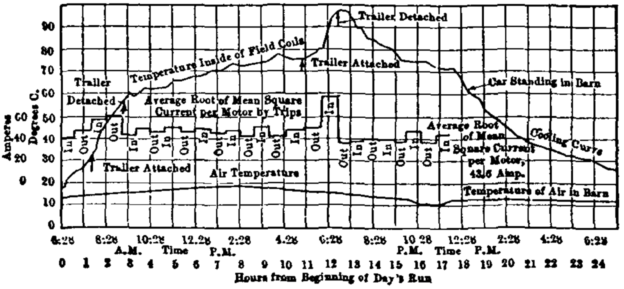


FIG. 72.—Motor temperature, with rush hour trailer.

effect on motors of pulling a trailer may be seen from the curves in Figs. 71 and 72 which are reproduced from a paper by Clarence Renshaw, Trans. A.I.E.E., 1903, Fig. 71 showing rise in temperature of motors with one car operating singly and Fig. 72 showing the rise in temperature with the car pulling a trailer during the rush hours. It will be noted that the presence of the trailer for one trip in the morning when the motors were fairly cool only caused the temperature to rise much more rapidly without causing it to reach any higher value, but in the afternoon when the motors had reached the temperature where they would otherwise have remained, the addition of a trailer for about 1¼ hours raised the temperature in this case approximately 25 deg. from 75 deg. C. The use of trailers for rush-hour service is also likely to cause delay, due to the reduction in speed on account of the added weight, and the inability of the trailer to move by itself causes further delay in backing it up, dropping it, and possibly shifting it at stub-end terminals. Otherwise, in certain classes of service trailer operation may be very successful without causing injury to the motor equipment, especially by taking advantage of the thermal capacity of the motors in attaching trailers to motor cars which have been out of service for 5 or 6 hours and leaving them attached for only a limited number of trips. It may be possible, also, to compensate for the extra load by a reduction in the number of stops which would ordinarily be made by the motor car without the trailer. With multiple-unit operation it is of course possible to apply motors to the cars more efficiently, both as to size and gear ratio, than if trailer operation had to be provided for, but on the other hand, the motor equipment on all cars, including those used as the second car, represents considerable dead investment when train operation is resorted to only infrequently. In rapid transit service where train operation is the rule instead of the exception, there is of course no question as to the advantage of multiple-unit operation, thus distributing the motors among the axles all along the train, reducing the capacity of the single-motor equipments, and making available for traction a much greater proportion or all of the weight of the entire train.

The Public Service Railway Company of New Jersey conducted a series of experiments with the view of determining the merits of multiple-unit train operation as applied to city service; whether the operation of two-car trains in city service was economical and desirable; and the combination and number of motors best suited for such operation. Comparative tests were made with six trains, the following description being taken from *Aera*, 1913: The trains tested comprised two single cars, one four-motor and one two-motor, one train of two four-motor cars, one train of one four-motor and one two-motor car, one train of two two-motor cars and one train of a four-motor car and trailer. Motors were Westinghouse No. 307, with HL control. Some of the operating results are as follows:

Class of train	Per cent of standard running time	Kilowatts per car mile, per cent	Watt-hours per ton mile, per cent
Four-motor single car	101.4	100.0	100.0
Two-motor single car	106.5	68.4	78.7
Two four-motor cars	100.8	96.0	100.0
One four-motor and one two-motor car.	102.9	81.5	92.6
Two two-motor cars	104.5	65.9	83.5
Four-motor car and trailer.	117.8	94.1	111.2

The table shows that the four-motor cars, either single or in trains, kept fairly close to the standard running time. A train made up of a four-motor car and a two-motor car lost a little time, but not as much as the two-motor cars, either single or connected. The four-motor car with trailer fell much further behind. The energy consumption figures are taken from typical rush-hour trips in the direction of travel, and it will be noted that the two-motor cars show a saving of about 30 per cent in energy consumption per car mile over the four-motor cars. This, because of the lighter weight of the two-motor cars, corresponds to about 20 per cent per ton mile. The six-motor train shows a saving of about 14 per cent per car mile, or about 8 per cent per ton mile, as compared with the eight-motor train. The motor car with trailer shows a slight saving on a car-mile basis, but over 10 per cent more energy consumption per ton mile than a train of two similar motor cars.

The six-motor train seems to be worthy of special consideration for this class of service. Its running time was very nearly as good as that of the eight-motor train or of the single four-motor car, and at the same time it showed a saving of 7.4 per cent in energy consumption per ton mile and about twice as much per car mile. It has none of the disadvantages of trailer operation and it would seem to be entirely feasible to operate the two-motor cars singly during the middle of the day for a larger part of the year and thus secure the still greater energy economy as shown by the two-motor car tests. If, however, weather or rail conditions make it advisable, the four-motor cars could be run throughout the day, while the combination of a two-motor and a four-motor car in the rush hours would meet the conditions of heavy travel much more satisfactorily than a train of two two-motor cars, and practically as well as an eight-motor train. The report on the Public Service tests concludes that train operation in city service is desirable under certain conditions and during certain periods of the day. This is indicated (1) by the facilities which train operation provides for handling rush-hour loads and traffic of extraordinary proportions; (2) by the relief offered through congested sections by operation in trains instead of as single units; (3) by the reduction in platform labor expense which train operation makes possible in comparison with single cars.

SECTION IV

RAILWAY MOTORS

A.I.E.E. Standards. The following numbered sections (5101 to 5502, inc.) relative to the rating, capacity and selection of railway motors, are from the Standards of the American Institute of Electrical Engineers, 1922 edition.

5101. Temperature Limits of Railway Motors in Continuous Service. The following maximum observable temperatures are permissible in the windings of railway motors, when in continuous service.

Under extreme ambient temperatures it is permissible to operate, for short infrequent periods, at 15 deg C higher temperature than specified in this rule

Owing to space limitations and the cost of carrying dead weight on vehicles, it is considered good practice to operate propulsion machinery at higher temperatures than would be advisable in stationary machines

Class of material	Temperature By thermometer	Temperature By resistance
Cotton, silk, paper and similar materials when so treated or impregnated as to increase the thermal limit, also enamelled wire	85 deg C.	110 deg. C.
Mica, asbestos and other materials capable of resisting high temperatures, in which any Class A (above) material or binder is used for structural purposes only, and may be destroyed without impairing the insulation or mechanical qualities of the insulation	100 deg C.	130 deg. C.

5202. Nominal Rating of Railway Motors. The nominal rating of a railway motor shall be the mechanical output at the car or locomotive axle, measured in kilowatts, which causes a rise of temperature above the surrounding air, by thermometer, not exceeding 90 deg. C at the commutator, and 75 deg C at any other normally accessible part after one hour's continuous run at its rated voltage (and frequency in the case of an alternating current motor) on a stand with the motor covers arranged to secure maximum ventilation without external blower The rise in temperature, as measured by resistance, shall not exceed 100 deg C The statement of the nominal rating shall include the corresponding voltage and armature speed.

In the absence of any specification as to the kind of rating, the nominal rating shall be understood

5203. Continuous Ratings of Railway Motors. The continuous ratings of a railway motor shall be the inputs in amperes at which it may be operated continuously at ½, ¾ and full voltage respectively, without exceeding the observable temperature rises specified below, when operated on stand test with motor covers and cooling system, if any, arranged as in service. Inasmuch as the same motor may be operated under different conditions as regards ventilation, it will be necessary in each case to define the system of ventilation which is used. In case motors are cooled by external blowers, the flow of air on which the rating is based shall be given.

The temperature rise in service may be very different from that on stand-test. See Sec. 5502 for the relation between stand-test and service temperatures as affected by ventilation.

STAND-TEST TEMPERATURE RISES OF RAILWAY MOTORS

Class of material	Temperature rises of windings By thermometer	By resistance
Cotton, silk, paper and similar materials when so treated or impregnated as to increase the thermal limit; also enamelled wire....................	65 deg. C.	85 deg. C.
Mica, asbestos and other materials capable of resisting high temperatures, in which any Class A (above) material or binder is used for structural purposes only, and may be destroyed without impairing the insulation or mechanical qualities of the insulation....................	85 deg. C.	105 deg. C.

5204. **Ratings of Field-control Railway Motors.** The nominal and continuous ratings of field-control motors shall relate to their performance with the operating field which gives the maximum motor rating. Each section of the field windings shall be adequate to perform the service required of it, without exceeding the specified temperature rises.

5210. **Ratings of Electric Locomotives.** Locomotives shall be rated in terms of weight on drivers, nominal one-hour tractive effort, continuous tractive effort and corresponding speeds.

5211. The weight on drivers, expressed in pounds, shall be the sum of the weights carried by the drivers and of the drivers themselves.

5212. The nominal tractive effort, expressed in pounds, shall be that exerted at the rims of the drivers when the motors are operating at their nominal (one-hour) rating.

5213. The continuous tractive effort, expressed in pounds, shall be that exerted at the rims of the drivers when the motors are operating at their full-voltage continuous rating, as indicated in Sec. 5203. In the case of locomotives operating on intermittent service, the continuous tractive effort may be given for ½ or ¾ voltage, but in such cases, the voltage shall be clearly specified.

5214. The rated speed, expressed in miles per hour, shall be that at which the continuous tractive effort is exerted.

5337. **Losses in Gearing and Axle Bearings of Railway Motors.** The losses in gearing and axle bearings for single-reduction single-geared motors varies with the type, mechanical finish, age and lubrication. The following values, based upon accumulated tests, shall be used in the comparison of single-reduction single-geared motors, Sec. 5339.

Per cent of input at nominal rating	Losses in axle bearings and single-reduction gearing as per cent of input
200	3.5
150	3.0
125	2.7
100	2.5
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
[illegible]	[illegible]
30	6.7
25	8.5

Further investigation may indicate the desirability of giving separate values of the losses for full and tapped fields, or low and high speed motors.

5338. **Brush Friction, Armature-bearing Friction and Windage.** The brush friction, armature-bearing friction and windage, shall be determined as a total under the following conditions: In making the test, the motor shall be run without gears. The kind of brushes and the brush pressure shall be the same as in commercial service. Drive the machine idle as a series motor on low voltage. The product of armature counter-electromotive-force and amperes at any speed shall be the sum of the above losses at that speed. (See Sec. 5339.)

5339. **No-load Core Loss, Brush Friction, Armature-bearing Friction and Windage.** The no-load core loss, brush friction, armature-bearing friction and windage shall be determined as a total under the following conditions: In making the test, the motor shall be run without gears. The kind of brushes and the brush pressure shall be the same as in commercial service. With the field separately excited, such a voltage shall be applied to the armature terminals as will give the same speed for any given field current as is obtained with that field current when operating at normal voltage under load. The sum of the losses above mentioned is equal to the product of the counter-electromotive force and the armature current. The no-load core loss is obtained by deducting from the total losses thus obtained the power required to drive the motor at corresponding speeds as determined under Sec. 5338.

The core loss of direct current railway motors under load shall be assumed to have the values given below:

Per cent of input at nominal rating	Core loss as per cent of no-load core loss
200	165
150	145
100	130
75	125
50	123
25 and under	122

With motors designed for field control the core losses shall be assumed as the same for both full and permanent field. It shall be the mean between the no-load losses at full and permanent field, increased by the percentages given in the above table.

Approximate Losses in D.C. Railway Motors. In comparing projected railway motors, and in case it is not possible or desirable to make tests to determine mechanical losses, the following values of these losses, determined from the averages of many tests over a wide range of sizes of single-reduction single geared motors, will be found useful, as approximations. They include axle-bearing, gear, armature-bearing, brush-friction, windage, and stray-load losses

Input in per cent of that at nominal rating	Losses as per cent of input
100 or over	5 0
75	5.0
60	[illegible]
50	[illegible]
40	[illegible]
30	[illegible]
25	17 0

The core loss of railway motors may also be determined as specified for other machines.

Characteristic Curves of Railway Motors

5401. **General.** The characteristic curves of railway motors shall be plotted with the current as abscissas and the tractive effort, speed and efficiency as ordinates. In the case of alternating current, the power factor shall also be plotted as ordinates.

5402. **Voltages.** Characteristic curves of direct current motors shall be based upon full voltage, which shall be taken as 600 volts, or a multiple thereof

5403. **Field-Control Motors.** In the case of field-control motors, characteristic curves shall be given for all operating field connections.

Selection of Railway Motor for Specified Service

5501. **Data Required in Selecting Motor.** The following information relative to the service to be performed is required, in order that an appropriate motor may be selected.

(*a*) Weight of total number of cars in train (in tons of 2000 lb) exclusive of electrical equipment and load
(*b*) Average weight of load and durations of same, and maximum weight of load and duration of same.
(*c*) Number of motor cars or locomotives in train, and number of trailer cars in train.
(*d*) Diameter of driving wheels
(*e*) Weight on driving wheels, exclusive of electrical equipment
(*f*) Number of motors per motor car
(*g*) Voltage at train with power on the motors—average, maximum and minimum.
(*h*) Rate of acceleration in miles per hour per second.
(*i*) Rate of braking in miles per hour per second
(*j*) Speed limitations, if any (including slowdowns).
(*k*) Distances between stopping points
(*l*) Average duration of stops
(*m*) Schedule speed, including stops, in miles per hour
(*n*) Train resistance in pounds per ton of 2000 pounds at stated speeds
(*o*) Moment of inertia of revolving parts, exclusive of electrical equipment
(*p*) Profile and alinement of track
(*q*) Distance coasted as a percentage of the distance between stopping points.
(*r*) Duration of layover at end of run, if any.

5502. Method of Comparing Motor Capacity with Service Requirements. When it is not convenient to test motors under actual specific service conditions, recourse may be had to the following method of determining temperature rise from the stand-tests. The essential motor losses affecting temperatures in service are those in the motor windings, core and commutator. The mean service conditions may be expressed, as a close approximation, in terms of that continuous current and core loss which will produce the same losses and distribution of losses as the average in service.

A stand test with the current and voltage which will give losses equal to those in service, will determine whether the motor has sufficient capacity to meet the service requirements. In service, the temperature rise of an enclosed motor (Sec. 4044), well exposed to the draught of air incident to a moving car or locomotive, will be from 75 to 90 per cent (depending upon the character of the service) of the temperature rise obtained on a stand test with the motor completely enclosed and with the same losses. With a ventilated motor (Sec. 4045 and 4046), the temperature rise in service will be 90 to 100 per cent of the temperature rise obtained on a stand test with the same losses.

In making a stand test to determine the temperature rise in a specific service, it is essential in the case of a self-ventilated motor to run the armature at a speed which corresponds to the schedule speed in service. In order to obtain this speed, it may be necessary, while maintaining the same total armature losses, to change somewhat the ratio between the I^2R and core-loss components.

Calculation for comparing motor capacity with service requirements. The heating of a motor should be determined, wherever possible, by testing it in service, or with an equivalent duty-cycle. When the service or equivalent duty-cycle tests are not practicable, the ratings of the motor may be utilized as follows, to determine its temperature rise

The motor losses which affect the heating of the windings are as stated above, those in the windings and in the core. The former are proportional to the square of the current. The latter vary with the voltage and current, according to curves which can be supplied by the manufacturers. The procedure is therefore as follows:

(*a*) Plot a time-current curve, a time-voltage curve, and a time-core loss curve for the duty-cycle which the motor is to perform, and calculate from these the root-mean-square current and the average core loss.

(*b*) If the calculated r m s. service current exceeds the continuous rating, when run with average service core loss and speed, the motor is not sufficiently powerful for the duty-cycle contemplated.

(*c*) If the calculated r m s service current does not exceed the continuous rating, when run with average service core loss and speed, the motor is ordinarily suitable for the service.

In some cases, however, it may not have sufficient thermal capacity to avoid excessive temperature rises during the periods of heavy load. In such cases a further calculation is required, the first step of which is to compute the equivalent voltage which, with the r m s. current, will produce the average core loss Having obtained this, determine, as follows, the temperature rise due to the r m s service current and equivalent voltage

Let $\left.\begin{aligned} t &= \text{temperature rise} \\ p_o &= I^2R \text{ loss, kw} \\ p_c &= \text{core loss, kw} \end{aligned}\right\}$ with r m s service current and equivalent service voltage

$\left.\begin{aligned} T &= \text{temperature rise} \\ P_o &= I^2R \text{ loss, kw.} \\ P_c &= \text{core loss, kw.} \end{aligned}\right\}$ with continuous load current corresponding to the equivalent service voltage

Then $t = T\dfrac{p_o + p_c}{P_o + P_c}$, approximately.

(*d*) The thermal capacity of a motor is approximately measured by the ratio of the electrical loss in kw at its nominal (one-hour) capacity, to the corresponding maximum observable temperature rise during a one hour test starting at ambient temperature.

(*e*) Consider any period of peak load and determine the electrical losses in kilowatt hours during that period from the electrical efficiency curve. Find the excess of the above losses over the losses with r m s service current and equivalent voltage. The excess loss, divided by the coefficient of thermal capacity, will equal the extra temperature rise due to the peak load. This temperature rise added to that due to the r.m.s. service current, and equivalent voltage, gives the total temperature rise. If the total temperature rise in any such period exceeds the safe limit, the motor is not sufficiently powerful for the service.

(*f*) If the temperature reached, due to the peak loads, does not exceed the safe limit, the motor may yet be unsuitable for the service, as the peak loads may cause excessive sparking and dangerous mechanical stresses. It is, therefore, necessary to compare the peak loads with the short-period overload capacity. If the peaks are also within the capacity of the motor, it may be considered suitable for the given duty-cycle.

Relation between Ratings and Voltage. The change in rating of a railway motor due to change in the voltage at which the rating is made is difficult of exact determination except by repetition of the stand test at the various voltages under consideration. In general, however, it may be said that the nominal, or one hour rating, is almost proportional to the voltage for such ordinary changes in the latter as are encountered in practice. The one hour rating of a railway motor is a measure of the heat storage capacity of the motor. Most of the heat generated is used in raising the temperature of the motor, although part of it is dissipated by radiation from the frame and on ventilated motors by the air blown out by the fan. The amount of heat dissipated in this manner will vary from 15 to 40 per cent of the total heat generated on a one hour run depending upon how well the motor is ventilated. For a given current the core loss and friction and windage will increase as the voltage increases, since the speed varies with the counter-electromotive force. In a ventilated motor the ventilation, and hence the amount of heat dissipated by this means, will also increase with the speed. As a result, the one hour current ratings at say, 600 and 500 volts, will be practically the same, and it is generally true that the nominal one hour rating is proportional to the voltage.

The continuous ratings are given for ½, ¾ and full voltage, and the difference is seldom more than a few amperes for either a ventilated or non-ventilated motor. It is therefore usually safe to interpolate between these values for an intermediate voltage.

Motor Capacity for Specific Service. As suggested in the Standards of the A.I.E.E , the most satisfactory method of comparing motor capacity with service requirements is to make tests under the actual service conditions or equivalent duty-cycle and with the specific motor under consideration. Such tests are often costly and frequently impossible or not practicable; recourse must then be had to some such method as outlined in the notes under A I E E. Rule 5502 (page 221). This involves the plotting of run curves (the data for which are required by Rule 5501), the calculation of r.m s. current and average core loss, the consideration of peak loads, thermal capacity, commutating ability, armature speed and ventilation, all as outlined under Rule 5502.

Capacity of High Speed Self-ventilated Motors. In the older types of heavy, slow speed, totally enclosed motors, the temperature depends almost wholly upon the total loss in the motor, since all losses are dissipated through the motor frame. The self-ventilated motors of more recent design are arranged for circulation of air through both armature and field windings by means of fans mounted on the shafts of small high speed armatures, and as a result there has been a considerable reduction in weight of motor per nominal horsepower. In the application of these light weight, high-speed, self-ventilated motors, it must be remembered that they have much less thermal capacity than the older type enclosed motors, so that short time heavy loads are of relatively much greater importance, and the average running temperature must be considerably lower in order that the same maximum temperature be not exceeded. With the self-ventilated motor, the .temperature depends not only on the losses, but on the armature speed, and the core loss plays a much smaller part than in the enclosed motor. The temperature rise with a given current is usually less at a high than at a low voltage, as the increased ventilation due to the high speed more than offsets the higher core losses. It is therefore desirable to consider the average armature speed in service as well as the equivalent voltage. If the average armature speed is below that required to develop the average core loss, extra margin must be given. Conversely, if the average speed is higher than that given at the equivalent voltage, the equipment will have a good margin. It is also necessary to consider the possibility of operating at slow speed under such conditions as in bucking snow, when the losses will be high and the ventilation poor. To sum up, it must be recognized that the high speed, self-ventilated motor should be applied with greater care than the old enclosed types. The high speed fan will give good ventilation and normally gives a low temperature rise. But when heavy loads, low thermal capacity and low speeds are combined, disastrous heating is likely to occur.

The practical result in the use of the ventilated type motor is that in a given service a lighter motor may be used provided the lower thermal capacity of the smaller motor is not a limit. In general, in both city and suburban service, these lighter motors do meet the service requirements where formerly the heavier enclosed motor was used, and of course give a very considerable saving both in first cost and in reduced weight of car equipment. For instance, on a car where formerly a Westinghouse 101–B–2 motor would have been used, it is now found to be feasible to use the Westinghouse 532–B or even, possibly, the 514–C. By total weight of motors and gears, the comparison is shown in the following table:

Type	Hour rating	Weight, pounds
101–B–2	40 hp., 500 volts, 520 r p m.	2780
306–CV–4	65 hp , 600 volts, 700 r p m.	2660
307	50 hp , 600 volts, 615 r p m.	2700
532–B	50 hp , 600 volts, 665 r p.m.	2590
514–C	40 hp 600 volts, 760 r p m.	1770

If the user of 101–B–2 motors desires to duplicate that motor as to thermal characteristics in a modern commutating pole motor, it will be necessary to select a motor of approximately the same weight as the old motor. The total weight of the motor is a rough indication of thermal capacity, but a better check is the size of copper in motors of the same speed. For overloads lasting only a short time, say ten minutes or less, the temperature rise is determined almost wholly by the thermal capacity of the copper windings. For longer overloads the determining factor is the thermal drop through the insulation. By areas of copper size and armature weight the comparison is as follows:

Type motor	Approximate copper area, circular mils	Approximate armature weight, pounds
101–B–2	13,000	585
306–CV–4	16,600	615
307	13,000	615
532–B	10,400	550
514–C	7,420	390

It is evident from this table that from the standpoint of thermal capacity only it would be necessary to select a motor of about the size of the 306–CV–4 or the 307 to match up with the 101–B–2. Such a heavy motor is undoubtedly better able to stand heavy short overloads without injurious heating than are the lighter motors, but the user pays for this capacity both in first cost and in operating cost.

Commutating Ability. Allowable Accelerating Current. Few non-interpole motors will commutate successfully more than 100 per cent of their nominal rating, and the practical allowable current used in accelerating these motors should never be considered as greater than the one-hour rating. In the early days the rating of railway motors generally determined their commutating ability and for this reason these motors were rated at approximately their maximum commutating ability.

For interpole motors the practical allowable current for acceleration is determined by the heating effect of the current and the size of the commutator. The design of the motor also has a great effect on this value of current. Of two motors of nearly identical design, the one with the larger commutator or better ventilation would rate a greater accelerating current than one with a small commutator or poorer ventilation. There is no set percentage or comparison between the one-hour or continuous rating and the allowable current used for the acceleration of a modern interpole motor, but it is safe to allow 50 to 75 per cent more current with interpoles than without, where the question is simply that of commutation. Usually, however, commutation will not limit the accelerating current of interpole motors, since practically any of them will commutate double the one-hour rating current without serious sparking. The limitation is usually a matter of heating or wheel slippage.

Preliminary Determination of Motor Capacity. S. B. Cooper, General Engineer with the Westinghouse Electric and Manufacturing Co., has developed the general operating characteristics and approximate motor application curves shown by Figs. 1 to 6, inclusive. These apply to direct current railway motors, and are based on the great similarity between the characteristic curves of such motors, as pointed out on page 242, and also making use of

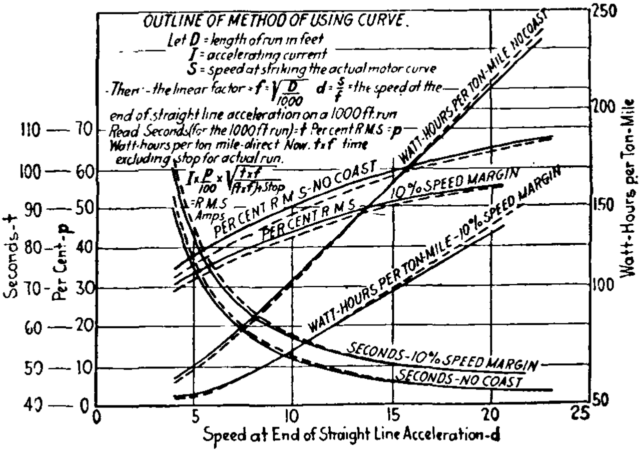


Fig. 1.—General operating characteristic. 1000-ft. run; 20 lb. per ton train resistance; acceleration and braking, 1.5 m.p.h.s. Solid lines—full load acceleration; dash lines—acceleration at 133 per cent full load. Speed margin based on time excluding stop, true speed margin therefore somewhat less than 10 per cent.

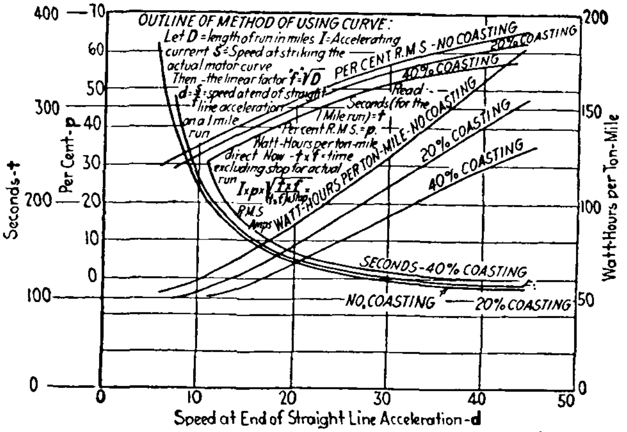


Fig. 2.—General operating characteristic. One mile run; 20 lb. per ton train resistance; acceleration and braking, 1.25 m.p.h.p.s.; coasting expressed as per cent of total moving time.

the fact that the ratio of the linear dimensions (speeds or times) of two similar speed time curves is equal to the square root of the ratio of the areas (distances), as shown on page 165.

In using Fig. 1 or Fig. 2 for the selection of an equipment, an approximation of the proper equipment must first be made, based on the weight of car, length of run and schedule speed desired. The accelerating tractive effort, calculated as in ordinary speed-time curve work (page 151), is 170 pounds per ton for Fig. 1 and 145 pounds per ton for Fig. 2. These values multiplied by tons per motor give the accelerating tractive effort per motor. From the characteristic curve of the motor to be used there can then be read the

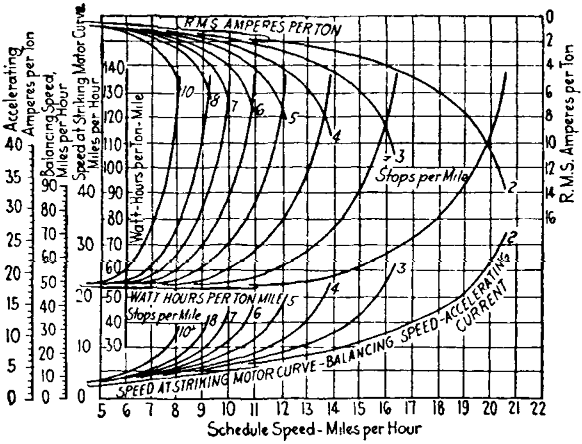


FIG. 3.—Approximate motor application curves for runs of 0.1 to 0.5 mile, showing balancing speed, speed at end of straight line acceleration, r.m.s. amperes per ton, accelerating amperes per ton, watthours per ton mile. Based on full load acceleration; acceleration and braking, 1.5 m.p.h.p.s.; 20 lb. per ton train resistance; coasting to give 10 per cent speed margin; 10 sec. stops; 500 volts. For other voltage, change r.m.s. and accelerating current in inverse proportion to voltage.

accelerating current per motor, I, and the speed, S, at the end of straight-line acceleration. Note also the approximate percentage of full load or nominal rating. Fig. 1 shows curves for 100 per cent and 133 per cent of full load acceleration, and other values may be interpolated; Fig. 2 is drawn for 133 per cent.

Inasmuch as Figs. 1 and 2 are drawn for 1000 ft. and one mile runs, respectively, a "linear correction factor" must be applied to the speed or time values shown thereon, to make them applicable to runs of other lengths. These correction factors are:

for Fig. 1, $f = \sqrt{\frac{\text{Length run in feet}}{1000}}$

for Fig. 2, $f = \sqrt{\text{Length run in miles}}$

Using $\frac{S}{f} = d$, the speed at end of straight-line acceleration for the equivalent 1000-ft. or one-mile run of Figs. 1 or 2, the curves show the following:

watthours per ton mile (applicable to run of any length if shape of speed-time curve is similar)

time, t, for run of 1000 ft. (Fig. 1) or one mile (Fig. 2)—multiply by correction factor, f, for run of other length

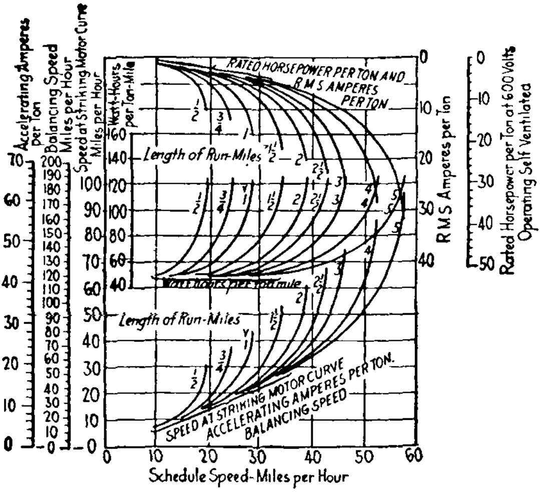


FIG. 4.—Approximate motor application curves for runs of 0.5 to 5 miles, showing balancing speed, speed at end of straight line acceleration, r.m.s. amperes per ton, accelerating amperes per ton, watthours per ton mile. Based on acceleration at 133 per cent full load; acceleration and braking, 1.25 m.p.h.p.s.; 20 lb. per ton train resistance; coasting 40 per cent of time excluding stop; stops of 10 sec. for one mile or less, and 10 sec. per mile for one to five miles; 525 volts. For other voltage, change r.m.s. and accelerating amperes in inverse proportion to voltage.

per cent r.m.s. current, p. Multiply by accelerating current, I, for r.m.s. current averaged over time of run excluding stop. For r.m.s. averaged over total time, take:

$$Ip\sqrt{\frac{\text{time exc. stop}}{\text{time inc. stop}}}$$

Based on the general operating characteristics of Figs. 1 and 2, the approximate application curves of Figs. 3 and 4 have been constructed, applying to runs of under and over one-half mile, respectively, and using data relative to accelerating current, acceleration and braking rates, train resistance, and stop time, as noted in the respective captions. From these curves may be read directly the maximum free running or balancing speed; the speed at end of straight line acceleration; the r.m.s. amperes per ton; the accelerating amperes per ton; and the watthours per ton mile.

If field control motors are used, Figs. 1, 2, 3 and 4, using the permanent (or short) field curves for the motor, will give approximately correct results except for r.m.s. amperes, rated horse power, and

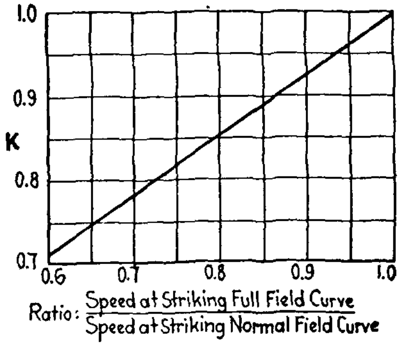


Fig. 5.—Constant, K, for corrections to field control motors.

energy consumption. The corrections for field control motors may be made as follows:

let H_N = (amperes)^2 × seconds, with normal field (found as above)

then H_S = reduction in (amperes)^2 × seconds, due to field control

$= K t_O (I_N^2 - I_F^2)$

where K = as shown by Fig. 5, speeds read from motor characteristic at accelerating currents

t_O = time of straight line acceleration (speed at end of straight line acceleration divided by rate of acceleration)

I_N = accelerating current on normal field

I_F = accelerating current on full field

then H_F = (amperes)^2 × seconds, with field control

$= H_N - H_S$

and $\sqrt{\dfrac{H_F}{T}}$ = r.m.s. current with field control

where T = total time, including stop, in seconds.

For energy saving due to field control:

Let P_N = kilowatts per car at end of straight line acceleration on normal field

Q = constant from Fig. 6

t_O = time of straight line acceleration (as above)

then E_R = energy saved by field control, watt-hours per run
$= P_N$ *to* Q

and E = energy saved, watt-hours per ton mile

$$= \frac{E_R N}{W}$$

where N = number of stops per mile
and W = weight of car in tons.

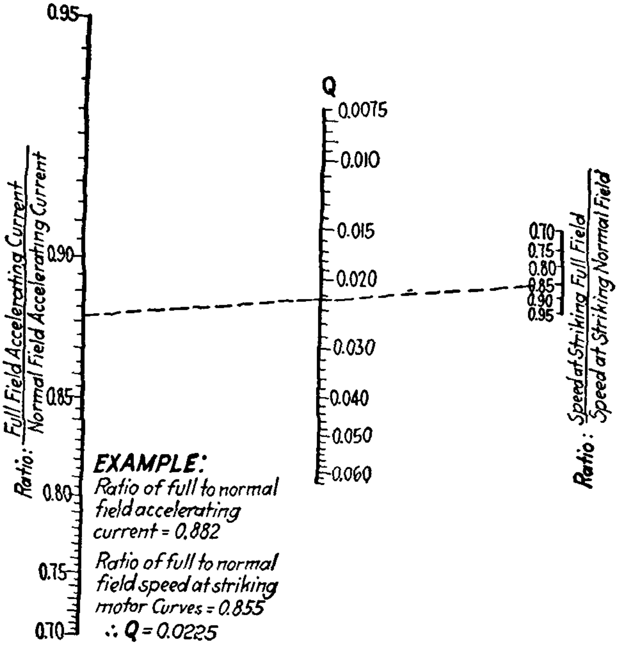


FIG. 6.—Constant, Q, for energy saving by field control.

Maximum Practicable Motor Capacity. The highest powered cars in service have equipments rating at average voltage 12 or 13 horsepower per ton of loaded car, while the (seldom approached) minimum is approximately 3 horsepower per ton. The maximum horsepower per ton is most frequently approached with high speed interurban equipments. The present average light weight double truck cars in city service run about 7.8 horsepower per ton; in interurban service, 8.5 horsepower per ton. The rated output of commercial railway motors at average voltage averages about 48 horsepower per ton of complete motor, up to 65 horsepower—above that rating, about 66 horsepower per ton. The minimum weight of complete car with average load which could be built to accommodate such motors and safely operate at high speeds would be approximately three times that of the motors alone Hence it is practically impossible to equip a car with more than about 20 horsepower per ton. Such cars and equipments would be very expensive to build and operate, and it is doubtful whether it will ever be practicable to exceed the present maximum of 12 or 13 horsepower per ton in equipping cars. The maximum possible schedule speeds, at instant of cutting off power, and energy consumption for various runs, with 12 horsepower per ton, are shown by Fig 7. These curves are based on maximum accelerating and

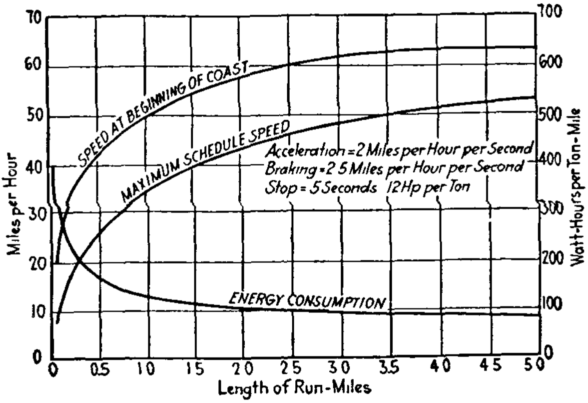


FIG 7 —Maximum practicable motor performances.

braking rates of 2 and 2 5 m p h p s, respectively, and a motor of the ventilated field control type. They are intended only as a guide to the possibilities of equipments, as such performance usually is too costly to be justified

Approximation of Required Motor Capacity. It has been noted that a definite railway motor equipment on a car of fixed weight will attain approximately the same temperature rise irrespective of the number of stops per mile demanded by service conditions (although the schedule speed will be affected thereby). Some engineers, therefore, for purposes of approximation, rate a railway motor at the tons train weight per motor which can be moved at a given maximum speed (or gear ratio) and with an allowable temperature rise The following table shows the approximate nominal one-hour horsepower rating of self-ventilated direct current motors required for various car or train weights and maximum speeds, with an allowable temperature rise of 65 deg. C.

HORSEPOWER MOTOR CAPACITY REQUIRED, DIRECT-CURRENT RAILWAY MOTORS, VENTILATED TYPE

Max speed, m p h	Weight of train including passengers, tons												
	20	30	40	50	60	80	100	120	150	180	200	250	300
25	103	142	181	215	248	312	375	435	526	610	667	824	975
30	116	160	202	240	278	346	419	483	580	673	734	900	1,075
35	131	179	226	268	309	383	461	531	638	738	807	993	1,175
40		200	250	298	341	419	505	579	695	802	877	1,080	1,270
45			275	326	372	450	549	632	754	872	955	1,165	1,371
50				360	408	502	600	685	818	940	1,030	1,257	1,477
55				392	442	548	651	744	885	1,013	1,116	1,358	1,583
60					483	596	708	802	960	1,097	1,200	1,458	1,695
65					528	640	759	863	1,025	1,169	1,280	1,559	1,788

Theory of Single Phase Commutator Motor. Figure 8 shows a four-pole stator with torque field coils assembled in place. This torque field is connected in series with the armature, as shown in Fig.

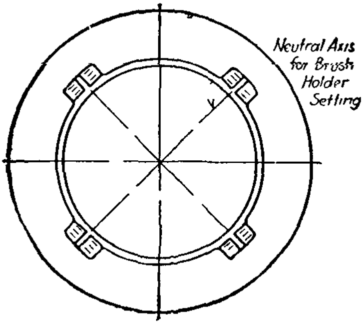


FIG 8—Four-pole single phase commutating alternating current motor with main or torque field winding

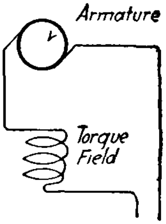


FIG 9—Connection of armature and torque field, a c motor

9. If an alternating current voltage is impressed across this circuit, current will flow and two flux fields will be set up, one by the armature and one by the torque field winding. These two fluxes will produce an inductance voltage at right angles to the current. Due to the current carrying armature conductors finding themselves in the flux of the torque field, they attempt to move out, and the armature rotates. On account of this rotation, a rotational or counter e m f, together with the resistance and iron loss drops and inductance volts, buck the impressed voltage and are equivalent to this voltage. An examination of Fig. 10 will indicate how high field and armature inductance will cause low power factor, as shown by the angle between the impressed voltage and motor current. It is desirable, therefore, to reduce the inductance to a minimum.

The useful flux is that due to the torque field and must not be disturbed. The flux set up by the armature must be eliminated, if possible. By refinement in design, the leakage flux due to the coil ends and through the slots is reduced to a minimum. The major portion of the armature flux is counteracted or compensated for by means of a winding placed in the faces of the pole—this

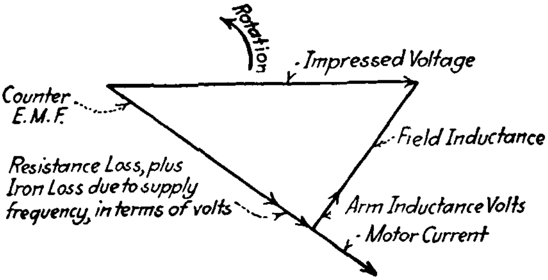


FIG. 10.—Relation between impressed and induced voltages, a. c. motor.

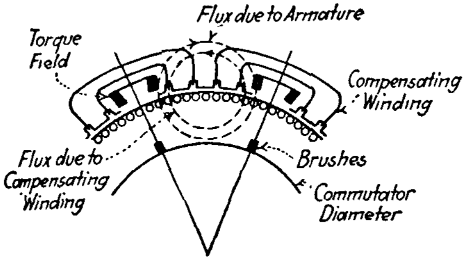


FIG. 11.—Single phase motor with torque field and compensating windings.

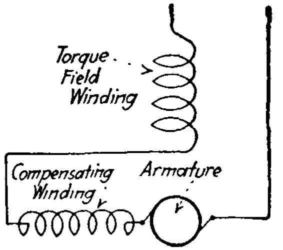


FIG. 12.—Connection of armature, torque field and conductive compensating winding, a. c. motor.

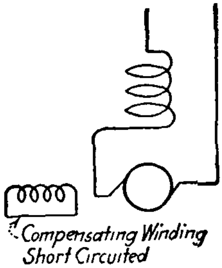


FIG. 13.—Connection of armature, torque field and inductive compensating winding, a. c. motor.

winding having just sufficient turns to neutralize the armature reaction. The arrangement of this winding in the pole faces is as shown in Fig. 11. If the motor is to run on both alternating and direct current, the winding should be connected as shown in Fig. 12, but if alternating current operation alone is required, it may be connected as shown in Fig. 13. The armature should then have zero impedance, for the same reasons that the

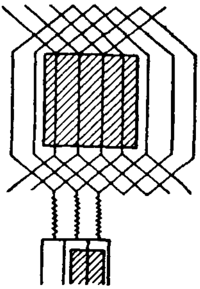


FIG. 14.—Resistance leads to commutator, a. c. motor.

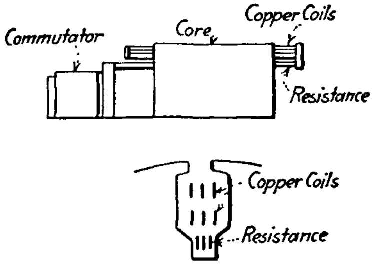


FIG. 15.—Arrangement of resistance leads to commutator in a. c. motor armature.

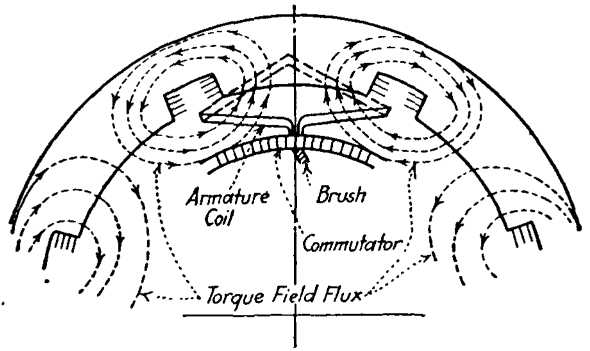


FIG. 16.—Transformer action of torque field on armature coil short circuited by brush.

primary of a transformer has no counter e.m.f. when the secondary is short-circuited.

Bad commutation on any motor is due to the brush short-circuiting two commutator bars of different potential, this resulting in a flow of current which, when interrupted as the bar leaves the brush, causes an arc or spark. In the case of the resistance lead type of motor, the commutation is improved by introducing a resistance in the path of the short-circuit current (see Fig. 14, which shows that the resistance is only in circuit when the coil is being commutated). The mechanical arrangement of the resistance leads in the armature is shown in Fig. 15. The sparking voltage is made of two components. One is a voltage due to rotation of the commutated coil in a leakage flux field set up partially by the main field and partially by the coil itself. This component exists in all commutator apparatus whether alternating or direct current,

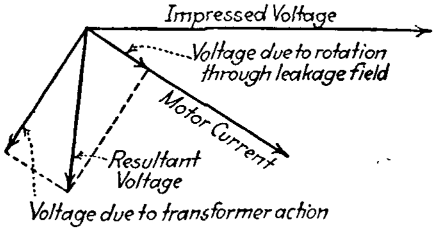


FIG. 17.—Components of sparking voltage, a. c. motor.

and is, of course, in phase with the armature current because the main field flux is in phase with the armature as well as the armature reaction field. The other component is a voltage due to the transformer action of the torque field flux on the coil being commutated (see Fig. 16). This last component, being due to transformer action, is at right angles to the motor or armature current. These two components with their resultant are shown in Fig. 17, in which it should be noted that the first vector depends entirely on rotation, while the second exists even at zero speed. In the doubly-fed motor, it was attempted to remove the sparking volts resultant vector and thereby allow the removal of the resistance leads. This was done by creating an opposing voltage equal to the resultant voltage shown in Fig. 17. This opposing voltage was produced by impressing a voltage across the compensating winding (Fig. 11). The diagram of connections was then as shown in Fig. 18. The flux in the compensating field, or cross field, must be at right angles to the impressed voltage. A voltage generated by rotation through this flux must be in phase with the flux, consequently the commutated coil, on rotating or moving through this flux, has a voltage generated in it which opposes the resultant voltage shown in Fig. 17. In Fig. 19 it can be seen that the two voltages practically neutralize each other, and consequently, resistance leads are not required.

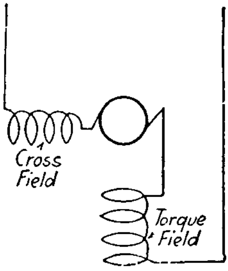


FIG. 18.—Doubly-fed a. c. motor with impressed voltage across compensating winding.

Due to the fact that the cross field winding is distributed across the faces of all the poles, considerable exciting current is necessary

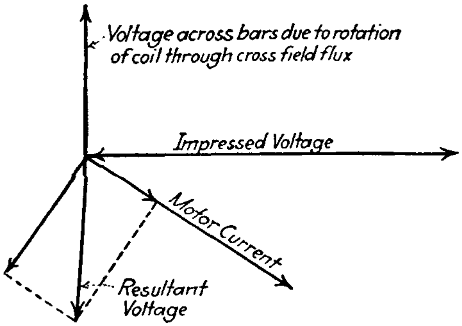


FIG. 19.—Sparking voltage neutralized by cross field

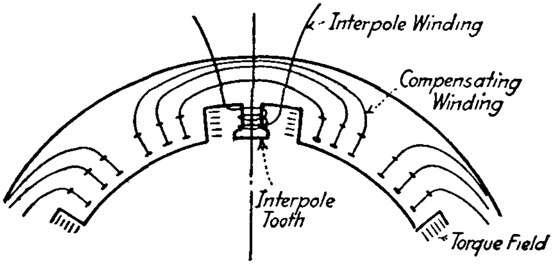


FIG. 20.—Interpole type a. c. motor.

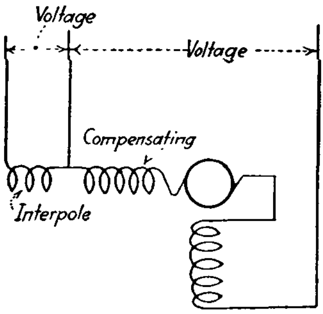


FIG. 21.—Connections of interpole type a. c. motor.

to maintain this flux field. The flux is only useful in the zone of commutation, therefore if all but the useful flux could be eliminated, the power factor of the motor could be raised and the rating of the motor increased, by elimination of heat due to exciting current and iron loss due to cross field This was accomplished in the interpole type of motor. In this motor the stator has a compensating winding similar to that shown in Fig. 11, and connected as shown in Fig. 12, but in addition, there is a small winding around a tooth in the commutating zone, which is called the interpole winding (see Fig. 20). The connection diagram is as shown in Fig. 21. Study of Figs. 18, 19, 20 and 21 will make clear how the commutation effect of the double-fed motor is obtained in the interpole motor, but with a gain in rating. It is evident from the foregoing that at zero speed no counteraction is obtained. In the doubly-fed motor the voltage impressed on the cross field is transformed over into the armature, making this voltage equivalent to having been applied on the armature plus torque field circuit. This complicates the control to some extent, which is not true in the case of the interpole motor.

General Structural Differences between Single Phase Commutator and Direct Current Motors. The single-phase series commutator motor is a low voltage machine. For traction purposes the armature composes a simple multiple wave winding with only one turn between adjacent commutator segments. This is to limit the voltage induced by transformer action in the coil short-circuited by the brush when undergoing commutation. The practical limit in the value of this transformer voltage likewise limits the value of the torque field flux which induces it, below that field strength ordinarily employed on direct current motors. For a given torque the armature current therefore is relatively greater. For these reason, (*a*) the armature diameter is larger as compared with that of a direct current armature, (*b*) the commutator is larger and wider and has a greater number of segments than that of a corresponding direct current motor, (*c*) the field poles are short and stubby as compared with the direct current motor, and (*d*) there are a comparatively large number of poles on the alternating current motor. Other structural differences are (*e*) the laminated core of the alternating current motor, (*f*) the use of a distributed auxiliary winding on the alternating current motor in addition to the main field winding, and (*g*) the smaller air gap on the alternating current motor. The field is laminated in order to reduce the losses and associated heating due to the alternating flux in the iron.

Types of Single Phase Commutating Motors. The purpose of the compensating winding, as shown above, is to neutralize cross-magnetization, thereby improving commutation and power factor. Compensation is classified as conductive or inductive, depending upon the process by which the current flowing in the compensating winding is obtained. If the compensating winding is connected to receive current from an external source, as by connecting it in series with the main motor circuit, the compensation is said to be conductive. If the compensating winding is closed on itself so that a current due to a directly induced electromotive force flows within

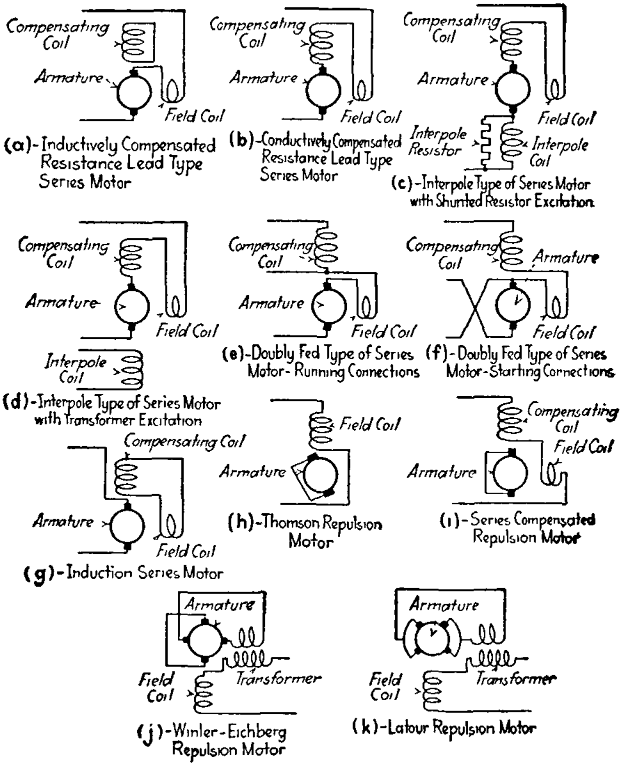


FIG. 22.—Various types of single phase commutating a. c. motors.

itself, the compensation is said to be inductive. On the New York, Westchester and Boston Railway, inductive compensation is used. On the New York, New Haven and Hartford Railroad, motors intended only for single-phase operation are inductively compensated, but those intended for both single-phase and direct current operation are conductively compensated. Insulation strain to ground is made negligible in the inductive compensation. When motors operate only on alternating current, the difference in results of either inductive or conductive compensation is inappreciable, but where the motor is to be operated with direct current, conductive compensation should be used. There are three general types of single-phase commutating motors, namely, the series, repulsion and series-repulsion. Of these the former two are in most general use in America (see Fig. 22).

Single Phase Motor Capacity. The single phase motor is essentially a high-speed motor, having a high copper loss and low core loss, and therefore is adapted more particularly to constant-speed running, and suffers in comparison with the direct current series wound motor when used for high rates of acceleration. There is no lack of starting torque with the single phase motor, but a large tractive effort is obtained only at the expense of a large copper loss, so that single phase motors are unsuitable for rapid transit service demanding high rates of acceleration, not because such motors cannot furnish the tractive effort required, but because the copper loss incident to such high tractive efforts would heat the motors unduly, or necessitate a motor much heavier than would be required with direct current. In interurban or express service with infrequent stops, the smaller core loss of the single phase motor brings it more nearly on a par with the direct current series wound motor as regards output per pound weight of motor. It is possible therefore to use the figures of the table on page 231, as applying to single phase motor capacity, where stops are not more frequent than one in 2 miles. For a higher frequency of stops, special study should be made, as before outlined.

Three Phase Motor Capacity. Owing to the few applications of three phase induction motors to motor cars, no general statement can be made regarding the capacity of such motors for service performed. The induction type of alternating current motor is not at all adapted for high rates of acceleration, owing to the poor efficiency of the equipment during fractional speed running. The field of the three phase induction motor lies in the direction of a service calling for constant effort at constant speed, and the problems wherein this type of motive power can be considered are so special and infrequent as to place it entirely beyond consideration for general motor car application, and to make its use in locomotives the subject of special study.

Locomotive Motor Capacity. Locomotive operation demands a special treatment of the motor capacity subject, in most cases requiring such special knowledge of motor construction as to prevent any general conclusions being drawn. With a locomotive working at 75 per cent or more of the slipping point of the drivers, there is a possibility of but small overload and hence the motor may be designed with lower density, in fact more like a stationary motor. In the absence of extreme overloads, the question of commutation becomes secondary, and the selection of a motor becomes a matter of its ability to radiate the heat generated in the specified service. The whole question of motor capacity for locomotives is so intimately associated with mechanical problems of locomotive design that it must be placed in the list of special problems requiring the careful cooperation of the manufacturers. These remarks do not apply to motors for switching locomotives, where the service is intermittent and the consideration more like that in motor car service, with weight on drivers and coefficient of adhesion of increased importance in determining maximum possible tractive effort or drawbar pull.

Three Phase Induction Motor. The three phase motor is called a constant speed motor (see p. 356 for methods of changing speed). It has the characteristics of a direct current constant speed motor. Neglecting control, the speed of the three phase induction motor depends upon the frequency of supply, and the speed falls off but little as the load on the motor is increased. The speed of a train driven by three phase induction motors ordinarily remains nearly constant up grade and down grade. The three phase induction motor regenerates energy, forcing it back into the line when the motor is driven above its synchronous speed. This process takes place as the train descends a grade, and since this regeneration brings a load on the motor acting as a generator, the train is not allowed to accelerate beyond the point at which the force due to gravity is balanced by the forces it is overcoming in driving the motor acting as a generator. Since the three phase motor is a constant speed motor, in ascending a grade it will make a demand on the power station proportional to the torque required and the losses to the motor. This demand may, however, be reduced by the energy of regeneration of trains descending. In order to operate on long up grades a three phase induction motor must have high continuous capacity.

The starting efficiency of three phase motors is low, so the motor is not well suited to service having frequent stops. The tractive effort may be held constant during the starting period, but this is done at the expense of energy lost in the motor starting resistance.

Since there is no electrical connection between the primary winding and the secondary winding of an induction motor, and there is no commutator, this type of motor may be built to operate on voltages which are high compared with those on which motors of other types are operated.

Speed of Induction Motor. The speed of an induction motor is equal to its synchronous speed minus the slip. The value of the slip increases at practically a constant rate from nearly zero at no load to about 2 to 5 or 6 cent of synchronous speed at full load.

$$S = \frac{f \times 60}{N}$$

in which S = synchronous speed, revolutions per minute
N = number of pairs of poles in the primary winding of the motor

f = frequency of applied electromotive force, cycles per second

$$S_1 = \frac{(100 - s)S}{100}$$

in which S_1 = actual speed of motor, revolutions per minute
s = slip, per cent.

TABLE OF SYNCHRONOUS SPEEDS

Number of pairs of poles in motor primary	1	2	3	4	5	6	7
Frequency, cycles per second	Synchronous speed, revolutions per minute						
15	900	450	300	225	180	150	128
25	1500	750	500	375	300	250	214
60	3600	1800	1200	900	720	600	514

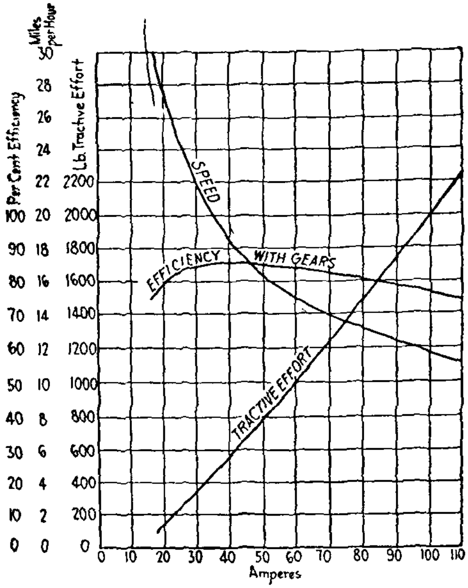


FIG 23.—Sample direct current railway motor characteristic curves General Electric No. 247 motor, 35 h.p., 600 volts, gear 63, pinion 15, ratio 4 20, wheels 30 in.

Railway Motor Characteristic Curves show:

tractive effort	(pounds)
speed	(miles per hour)
efficiency	(per cent)

and, in the case of alternating current motors:

power factor	(per cent)

each as a separate curve, plotted against

current	(amperes)

Such curves as furnished by the manufacturer apply to a definite motor only when that motor is operating at the definite voltage and with the definite gear ratio and driving wheel diameter for which the curves are drawn; for application to any other voltage, gear ratio or wheel diameter, the curves must be changed as described on page 246.

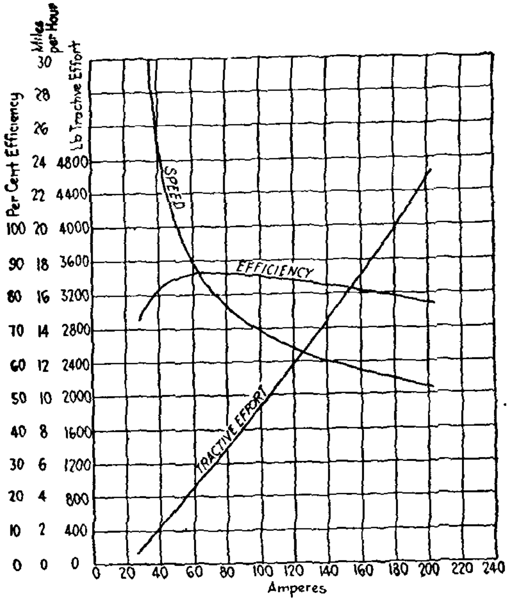


Fig. 24.—Sample direct current railway motor characteristic curves. Westinghouse No. 306CVD4 motor, 60 h.p., 600 volts, gear 72, pinion 15, ratio 4.80, wheels 33 in.

Typical characteristic curves are shown for direct current motors as Figs 23, 24 and 29; for a single phase series alternating current motor as Fig. 25; and for a three phase induction motor as Fig. 26.

There is a marked similarity in the characteristic curves of all railway motors of the same general type. As illustrative of this, Fig. 27, has been prepared, in which the characteristic curves of a number of self-ventilated, interpole, direct current motors, have been combined. The nominal ratings of the motors used in preparing Fig. 27 range from 25 to 140 horsepower, and the current, tractive effort and speed ordinates of all of them were reduced to terms of per cent of those values at the nominal, or one-hour rating. Fig. 27 shows the maximum, minimum and average efficiency, tractive effort and speed characteristics of all the motors considered, and although it was derived from a study of self-ventilated, interpole motors, it well illustrates the general shape of the characteristics of all direct current motors. The shape of the typical characteristics of the single phase series motor differ from that of the

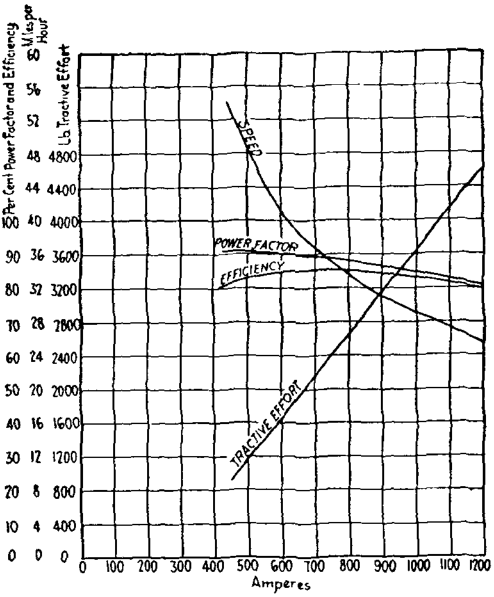


FIG. 25.—Sample single phase alternating current railway motor characteristic curves. Westinghouse No. 412B motor, 225 h.p., 300 volts, 25 cycles, gear 60, pinion 19, ratio 3.16, wheels 43½ in.

direct current in that its power varies much less with the speed, while the induction motor has a speed characteristic that is almost flat.

This marked similarity in railway motor characteristic curves has led to the use, by some engineers, of average curves, particularly for direct current motors, such as those shown in Figs. 27 and 28, for the purposes of preliminary calculations. In such use, any desired values may be given to the nominal or one-hour rating of two of the three elements, speed, tractive effort and input, these being connected together and with efficiency by the formula:

$$I = \frac{1.99\,ST}{EU}$$

where I = motor current, amperes
S = speed, miles per hour
T = tractive effort, pounds
U = efficiency, decimally expressed
E = voltage

Fig. 28, in which ordinates are shown as per cents of the nominal rating, thus may be used for an approximation of the characteristic curves of any direct current motor, by assigning any values of

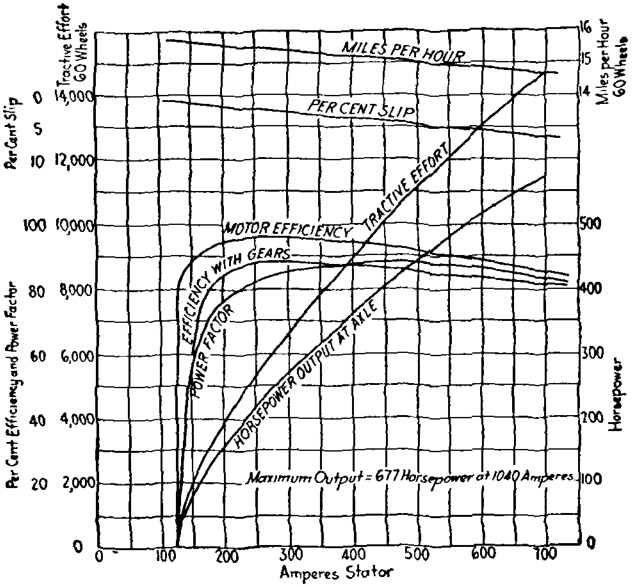


FIG. 26.—Sample three phase induction railway motor characteristic curve.

current, speed and input to the nominal rating, so long as the above formula is satisfied. Generally, speed and tractive effort are assumed and the current is derived from the above formula. If these assumptions are made for the nominal rating of the motor, the per cent ordinates of Fig. 28 may be expressed decimally and used as multipliers, thus enabling the curves to be read in amperes, miles per hour, and pounds tractive effort. Having determined the desired values of nominal rating speed, tractive effort and current, a commercial motor nearest to the proper rating may be selected, and knowing the driving wheel diameter, its gear ratio is determined. For further calculations, the characteristics of this commercial motor may be substituted for the general characteristics of Fig 28.

It is interesting to note that the characteristics of the direct current railway motor bear these inter-relations: *the speed is approximately inversely proportional to the cube root of the tractive effort and to the square root of the current.* This rule is based on an assumption of constant efficiency, so that it should be used only for rough preliminary calculations; the dotted curves of Fig. 28 correspond to the relations as above expressed, and show the extent of the error likely to result from its application.

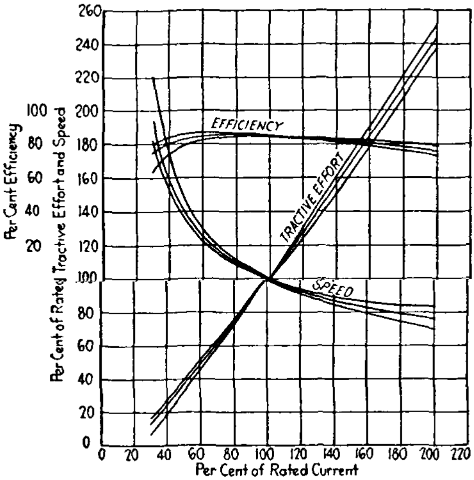


FIG. 27.—Railway motor characteristic curves. Current, tractive effort and speed values in per cent of nominal rated values Maximum, minimum and average for group of self-ventilated, interpole, direct current motors, from 25 to 140 h.p.

The approximate rule as given above lends itself to expression on the so-called polyphase slide rule, as has been pointed out by E E. Kimball. The polyphase slide rule shows four sets of logarithmic scales as follows, the latter named two being in addition to the first two which are shown on the ordinary slide rule:

D scale: the bottom scale, running from 1 to 10 in the length of the rule

A scale: the top scale, running from 1 to 10 to 100 in the length of the rule, and indicating squares of the values on the D scale

E scale: on the lower edge of the rule, running from 1 to 10 to 100 to 1000, and indicating cubes of the values on the D scale

CI scale: the middle scale—same as D scale but inverted or reversed in direction.

If the various scales are set with the ends coinciding, the runner may be set at the point on the A scale corresponding with any *per cent* of the nominal rated amperes current input; the corresponding point on the middle or CI scale will indicate the corresponding per cent of the nominal rated speed, and the corresponding point on the cube or E scale will indicate the corresponding per cent of the nominal rated tractive effort; all in accordance with the approximate relation beween these values as indicated by the above quoted rule. The polyphase slide rule in this way affords a quick approximation solution of problems involving direct current railway motor characteristics.

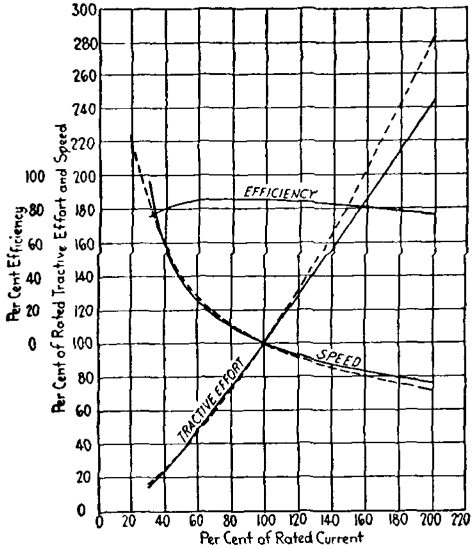


FIG. 28.—Railway motor characteristic curves. Current, tractive effort and speed values in per cent of nominal rated values. Solid line, average of direct current motors. Dotted lines from formula and as read from polyphase slide rule.

Gear Ratio is the ratio of the number of teeth on the driven gear to the number of teeth on the driving motor pinion. The ratio of the number of revolutions of the motor shaft per unit time to the number of revolutions of the car axle driven by that motor is equal to the gear ratio. The speed at which a car having a given driving wheel diameter will be moved by given motors at a given current and electromotive force depends (neglecting the variation of train resistance) upon the gear ratio. The approximate formulas connecting armature speed, car speed, gear ratio and wheel diameter are:

$$R = \frac{336SG}{d}$$

or $$S = \frac{Rd}{336G}$$

where R = armature speed, revolutions per minute
S = car speed, miles per hour
G = gear ratio as defined above
d = driving wheel diameter, inches.

The sum of the diameters of gear and pinion is twice the gear center distance and is therefore constant for a given motor. Then so long as the diametral pitch and gear center distance are constant, the total number of teeth in gear and pinion is constant, irrespective of gear ratio. This fact should be remembered in finding the number of teeth in gear and pinion for changing gear ratio. For instance, a gear ratio of $^{69}/_{15}$ (4 60) is to be changed to a ratio of 3.20. The total number of teeth is 84, and with the new ratio, this will be (3.20 + 1) or 4.20 times the number of pinion teeth. The pinion will therefore have (84 ÷ 4.20 =) 20 teeth, and the gear will have (3.20 × 20 =) 64 teeth.

Maximum Gear Ratio. The maximum gear ratio of any motor is limited mechanically by one or both of two things. The minimum pinion must be strong enough for the capacity of the motor, and the maximum gear must permit sufficient clearance below the gear case for safe operation. It is not practicable to use a pinion so small that with the best available material the teeth are weak or there is insufficient stock between the root of the tooth and the bore or keyway. The permissible clearance under the gear case is somewhat variable on account of the different characters of roadway in use. For example, a clearance which would be safe for low speed on an unpaved track with T-rails might be too small for use on streets with grooved rails and high crowned paving between rails. The approximate clearance under the gear case in any instance is readily determined when the number of teeth in the gear, the diametral pitch and the wheel diameter are known. The diameter of gear over the ends of the teeth is equal to the number of gear teeth plus two divided by the pitch. The outside diameter of the gear case is approximately one inch greater, on account of the clearance between teeth and case and thickness of case. Then the clearance from the bottom of gear case to top of rail is one-half the difference between wheel diameter and outside gear case diameter. A 69-tooth, 3-pitch gear has an outside diameter of (69 + 2) ÷ 3 = 23.67 in. For this gear the outside gear case diameter is 23.67 + 1 = 24.67 in. On a 33-inch wheel the clearance will be (33 − 24.67) ÷ 2 = 4.165 in.

Changes in Characteristic Curves, Due to Changes in Gear Ratio, Wheel Diameter or Voltage. When it is desired to apply a series railway motor to some other gear ratio, driving wheel diameter, or voltage than those to which the original characteristic curves apply, the ratios shown by the following formulas may be used. Note that voltage changes should be applied with care; when small changes are made, it may be safe to use the approximate formula in which it is assumed that the speed ordinates vary directly with the voltage; the exact formula, shown later, indicates that it is the counter electromotive force, and not the impressed voltage, that controls.

The ordinates for the *tractive effort* curve for a series motor with any gear ratio or driving wheel diameter may be derived from the ordinates of the original *tractive effort* curve for that motor as follows:

$$T^1 = T\left(\frac{G^1 \times D}{G \times D^1}\right)$$

in which T^1 = tractive effort ordinate for any current value on the derived curve

T = tractive effort ordinate for the same current value on the original curve

G^1 = gear ratio for derived tractive effort curve

G = gear ratio for original tractive effort curve

D^1 = driving wheel diameter for derived tractive effort curve

D = driving wheel diameter for original tractive effort curve.

The approximate values of the ordinates of the *speed* curve for a series motor with any gear ratio or driving wheel diameter operating at any other voltage differing slightly from that at which the original speed curve was obtained, may be derived from the original *speed* curve as follows:

$$S^1 = S\left(\frac{E^1 \times G \times D^1}{E \times G^1 \times D}\right)\text{(approximately)}$$

in which S^1 = speed ordinate for any current value on the derived curve

S = speed ordinate for the same current value on the original curve

E^1 = voltage for derived speed curve

E = voltage for original speed curve.

(NOTE: When the voltage is not changed, that is, when $E^1 = E$, the value given for S^1 is exact. See p. 248 for formula for exact change due to voltage change.)

Example of the Derivation of Tractive Effort and Speed Curves. Fig. 29 shows tractive effort and speed curves (dotted) derived from the original curves by the above process as follows: The original gear ratio, wheel diameter and voltage are 2.78, 33, and 500, respectively, and it is desired to derive the tractive effort and approximate speed curves for a gear ratio, wheel diameter and voltage of 3.83, 36, and 550, respectively. In this case, G^1 = 3.83, G = 2.78, D^1 = 36, D = 33, E^1 = 550 and E = 500. Therefore

$$T^1 = T\left(\frac{3.83 \times 33}{2.78 \times 36}\right) = 1.26T$$

and

$$S^1 = S\left(\frac{550 \times 2.78 \times 36}{500 \times 3.83 \times 33}\right) = 0.87S,$$

Thus in Fig. 29 the dotted tractive effort curve is the locus of all points the ordinate of each of which is equal to 1.26 times the corresponding ordinate on the original tractive effort curve, and the dotted speed curve is the locus of all points the ordinate of each of which is equal to 0.87 times the corresponding ordinate on the original speed curve.

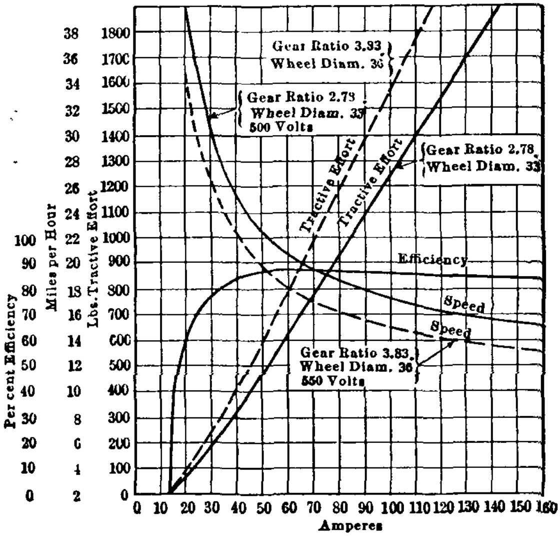


FIG. 29.—Illustrating change of tractive effort and speed curves for a change in gear ratio, driving wheel diameter and voltage. (GE-87 motor, 60 h.p.)

The actual change in the ordinates of the speed characteristic of a series railway motor due to a change in voltage is proportional to the counter electromotive force, as follows:

$$S^1 = S\left(\frac{E^1 - Ir}{E - Ir}\right)$$

where I = current
r = resistance of motor
S^1 = speed ordinate (at I current) on the derived curve
S = speed ordinate (at I current) on the original curve
E^1 = voltage for derived speed curve
E = voltage for original speed curve.

As the derivation of a new speed characteristic by the above formula involves a separate calculation for each value of current, it is convenient to make use of a graphical method for its application, such as the one proposed by A. M. Buck, and illustrated by

Fig. 30. Here the original speed-current characteristic is for 600 volts, and it is desired to plot a new curve for 500 volts. Lay off a second scale of amperes on the same base line and with the same ordinate values of amperes, but offset somewhat to the right, as shown. On the ordinate representing zero current on the second scale, and to any convenient scale of volts, the point E is at 600 (the original voltage) and E^1 is at 500 (the new voltage). The points C and D are located below 600 and 500 volts, respectively, by an amount representing the Ir drop for the current ordinate on which they are plotted (in this case 250 amp.). The lines EC and E^1D are drawn. To locate any point on the new speed curve corresponding to S on the original curve, proceed as follows: Take the point A on the line EC and at the current on the second scale of

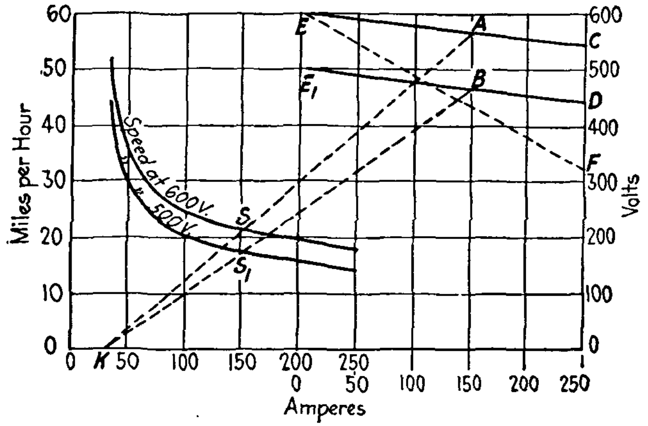


FIG. 30.—Graphical method of constructing speed-current motor characteristic for voltage differing from original curve.

amperes corresponding to the current at the speed S (in Fig. 30, 150 amp. is used for illustration). Then the line AS produced will cut the base line at some point K. The point B is on the line E^1D and at the same current ordinate as A. Then the point S^1, where the line KB cuts the current ordinate of S, is the point on the new speed curve corresponding to S on the original curve. Other locations of A and B corresponding to other currents and other points S on the original speed curve will result in other locations for K and give other points on the new speed curve, in the same manner.

A slight modification of Fig. 30 may be used to determine the performance of a motor when in series with an external resistance. In such case, locate F at a distance below the line volts corresponding to $I(R + r)$, I being the current on the ordinate of which F is located, R being the external resistance, and r the resistance of the motor. The procedure will then be as before, except that the points B will be taken on the line EF instead of on E^1D.

The use of the method shown in Fig. 30, and described above, generally involves either a replotting of the original speed-current curve or a piecing out of the sheet on which it is plotted, to accommodate the construction shown at the right of Fig. 30. Should either be inconvenient, the whole process may be performed within the limits of the motor characteristic curve sheet by a modification of the method as shown in Fig. 31. Here the base of the volt-ampere diagram is taken the same as that for the speed-current curve. The lines EC and E^1D are located as in Fig. 30. The point N is located so that $E^1N = OE$, and the line NM is drawn at an angle of 45 degrees. Then for any point S on the original speed curve, take B

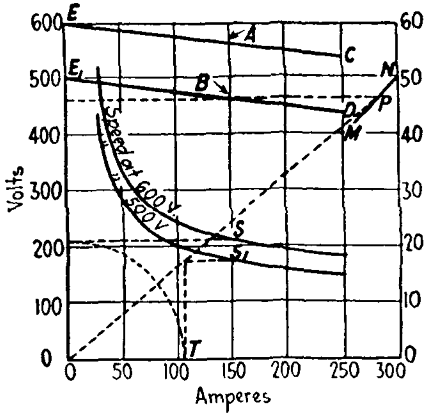


FIG. 31.—Graphical method of constructing speed-current motor characteristic for voltage differing from original curve.

(at the same current) on the line E^1D, project B to an intersection with NM at P, and draw PO. Then with a pair of dividers lay off the speed value of S on the base line as OT, project T up to the diagonal OP and thence across to the proper current ordinate at S^1, which is the point on the new speed curve corresponding to S on the original curve.

Effect of Variation in Driving Wheel Diameters. When all of the driving wheels on a motor car or train are not of the same diameter, the characteristic curves of the motors will vary accordingly. Since the speed of all wheels must be the same in miles per hour, the revolutions per minute of the armatures will not be the same, the motors driving the larger wheels will take more than their proper share of the load and consequently will show more heating than those driving the smaller wheels. The difference in current at various speeds may be determined from speed-current characteristics of the motors, corrected for the different wheel diameters, and the difference in heating may then be found by calculating the differences in core loss and I^2R loss due to the differences in currents and armature speeds. Although the shape of the speed-current characteristic and the distribution of losses between core and copper varies somewhat with different designs of motors, A. L. Broomall has worked out the curves shown as Figs. 32 and 33 based on an average speed-current curve such as in Fig. 28, and using the average distribution of losses as found in a number of direct current railway motors. Figure 32 shows the difference in current, and Fig. 33 the difference in heating which

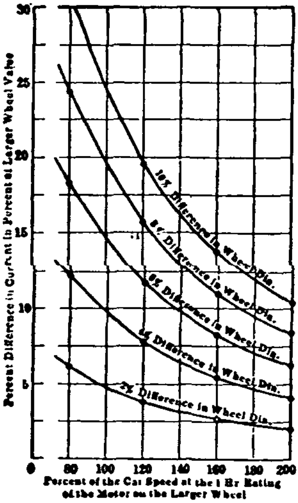


FIG. 32.—Differences in current due to differences in wheel diameter.

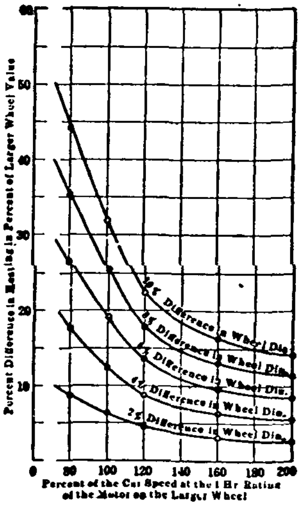


FIG. 33.—Differences in heating due to differences in wheel diameter.

results from various differences in wheel diameter and at different speeds. Mr. Broomall points out that the permissible difference in driving wheel diameter depends upon several considerations, among which he mentions the margin in temperature rise when the load is divided equally between the motors, the relative number of motors with the large and small wheels, and the ordinary operating speed of the car. If the car has three motors on large wheels and one on small wheels, the condition is not so serious as with three sets of small wheels and one large. If the service is such that the motor operates much of the time at heavy loads, a given per cent difference in wheel diameter is more serious than in service where most of the running is at high speed.

Resistance of Direct Current Motors. The electrical resistance of a large number of direct current railway motors, from data furnished by the manufacturers, is shown in the tables on pages 254 to 259, inclusive.

E. E Kimball (Trans. Am. Inst. Elec. Engrs., Vol., 33, p. 1715) proposed a method of determining the approximate resistance of a direct current railway motor from its efficiency curve and the following formula:

$$R = \frac{(8K_2 - 5K_1 - 15)E}{1100\, I} \text{ (approximately)}$$

in which R = approximate resistance of motor, ohms
K_1 = total losses at I current, per cent
K_2 = total losses at $2I$ current, per cent
I = any current not less than ¾ nominal rating, but not more than ½ largest current for which efficiency is known, amperes
E = voltage at which efficiencies are shown.

K_1 and K_2 may be taken from the efficiency characteristic of the motor, being 100 minus the corresponding efficiency. The method depends upon the assumptions that at ¾ nominal rating or more, the gear and friction losses amount to about five per cent (see A.I.E.E. rule No. 5339, p. 219) and also that the core losses may be taken as proportional to the cube root of the current. The accuracy of the method depends upon these assumptions and upon how accurately the efficiencies can be read. As errors in reading efficiencies may be cumulative, the resulting error in a single determination may be great, so that it is advisable to make check calculations using different values of current. This method was tested on nearly a hundred direct current railway motors of various types and capacities, the resistances of which were known; in 90 per cent of the trials the indications were within 10 per cent of the true values, in about half the trials within 5 per cent.

Resistance Losses due to Brush Contact. In calculations involving the Ir drop in direct current railway motors, it is usual to assume a drop of two or three volts due to brush contact resistance and to use this drop irrespective of the current.

Resistance of Car Wiring. In calculations involving the total resistance of the motor circuit, it is usual to assume the resistance of the car wiring as 20 per cent of the motor resistance, or the total resistance per motor including wiring as 120 per cent of the motor resistance.

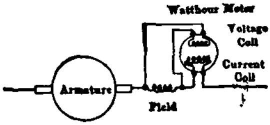


Fig. 34.—Watt-hour meter connected to measure I^2R loss.

Measurements of Equivalent Heating Current. The equivalent heating current or root mean square current may be obtained by the following method: A specially calibrated watt-hour meter is connected to record the energy lost in the motor field winding, as shown by Fig. 34, then the equivalent heating current will be equal to $\sqrt{\frac{e}{RT}}$

in which e = energy lost on motor field, watt hours
R = operating resistance of motor field, ohms
T = total time during which the equivalent current is desired (sum of accelerating, coasting, braking and stop periods), hours.

Another and very satisfactory method is by use of a "vacuum" or heat-insulated jar containing a measured quantity of water in which is immersed a known resistance, the leads to which are brought in through the cork, in company with a thermometer and a stirring rod. Since the heat generated is proportional to the square of the current, the temperature of the water will be raised in the same proportion (except for heat losses, which through the vacuum jar are very small). The instrument is calibrated by noting temperature rises resulting from the passage of various constant currents for given periods of time. In measuring large currents, a shunt or transformer is used, so that only a given proportion of the current is passed through the apparatus. By this method the r.m.s current values may be measured on either direct or alternating current, and, unlike many alternating current instruments, the accuracy is not affected by frequency or wave form.

Railway Motor Data. Tables on pages 254 to 259, inclusive, show trade numbers and ratings of Westinghouse and General Electric railway motors, and include not only the modern types but also a number of the older types which remain in use to some extent. Electrical resistances are shown, whether or not commutating pole, type of ventilation, type of frame, number and size of brushes, maximum safe armature speeds, maximum gear ratios, maximum axle diameters, minimum wheel diameters, rail clearances, and weights.

General Data on Westinghouse Direct Current Motors

Mfrs. No.	Volts	Nominal 1-hr. rating: H.P.	Nominal 1-hr. rating: R.P.M.	A.I.E.E. ratings continuous amperes at ½ voltage	¾ voltage	Full voltage	Electrical resistance, ohms	Weight: Without gears and gear case	Weight: With gears and gear case	Max. axle diam., in.	Min. wheel diam., in.	Rail clearance with min. wheel: Motor frame	Rail clearance: Gear case	Max. gear ratio	Max. safe arm. speed, R.P.M.	Brushes: No. per holder	Brushes: Size, in.	Commutating pole	Type of ventilation (See Note, p. 256)	Type of frame
12-A	500	36	835	23	20	...	....	1920	2270	4	33	4 3/4	4 9/16	68/14	2000	1	1/2 × 2 3/8	No	C	Split
38-B	500	45	550	34	31	...	.532	2050	2400	4	33	4 1/8	4 3/8	68/14	1500	2	1/2 × 1 1/2	No	C	Split
49	500	35	600	28	25	...	.693	1700	1900	4 1/4	33	4 3/4	4 1/8	68/14	1500	2	1/2 × 1 1/2	No	C	Split
56	500	55	500	46	41	...	.327	2700	3000	5	33	4 7/16	4 1/8	68/14	1500	2	1/2 × 2	No	C	Split
68-C	500	40	550	33	30	...	.618	1950	2280	4 3/4	33	4 13/16	4 1/8	68/14	1500	2	1/2 × 1 1/2	No	C	Split
76	500	75	500	56	46	...	.259	3400	3840	6	33	4	3 3/4	66/16	1500	2	1/2 × 2	No	C	Split
92-A	500	35	525	28	25	...	.790	1940	2265	5	33	4 5/8	4 3/16	69/15	2000	2	1/2 × 1 1/2	No	C	Split
93-A2	500	60	510	46	41	...	.380	3000	3440	5 1/2	33	3 3/8	3 5/8	71/16	1800	2	1/2 × 2	No	C	Split
101-B2	500	40	520	35	32	...	.585	2395	2780	5	33	4 1/8	4	69/15	1900	2	1/2 × 1 3/4	No	C	Split
112-B	500	75	650	56	46	...	.216	3035	3485	6	33	3 3/8	3 7/16	73/16	1800	2	1/2 × 2 3/8	No	C	Split
121-A	500	85	575	74	67	...	.192	3850	4300	6	33	5 1/4	3 3/4	59/16	1600	3	1/2 × 1 3/4	No	C	Split
300-B	600	220	610	150	120	...	.085	5815	6380	6 1/2	34	3 3/4	3 1/4	64/19	1400	4	5/8 × 2	Yes	C	Box
301	600	175	246	120	105	...	.103	4980	5510	6 1/2	34	4 1/8	4 1/8	59/18	1500	3	5/8 × 2 1/4	Yes	C	Box
302	600	140	665	95	80	...	.146	4200	4685	5 1/4	34 1/4	5 7/8	4 3/8	61/16	1600	3	5/8 × 1 3/4	Yes	C	Box
303-A	600	115	640	70	60	...	.240	3680	4150	6	33	5 7/8	3 3/8	61/16	1800	2	5/8 × 2 1/4	Yes	C	Box
304	600	90	740	60	50	...	.259	3140	3550	5 1/2	33	3 1/2	3 5/8	71/16	1850	2	5/8 × 1 3/4	Yes	C	Split
305	600	75	595	58	50	...	.426	3140	3550	5 1/2	33	3 1/2	3 5/8	71/16	1850	2	5/8 × 1 3/4	Yes	C	Split
306	600	60	695	50	44	...	.512	2390	2700	5	33	4 1/8	4	69/15	1900	2	5/8 × 1 1/2	Yes	SV	Split
306-CA	600	60	695	50	44	...	.512	2320	2630	5 1/2	33	4 5/8	4	69/15	1900	2	5/8 × 1 1/2	Yes	SV	Box
306-C	600	60	695	50	44	...	.512	2390	2700	5 1/2	33	4 5/8	4	69/15	1900	2	5/8 × 1 1/2	Yes	SV	Box

306-CV4	600	65	700	58	60	60	.475	2350	2660	5½	33	4⅝	4	6 9/15	1900	2	⅝ × 1½	Yes	SV	Box
306-CVD4	600	65	700	58	60	60	.475	2375	2750	5½	33	4 9/16	3¾	7 3/15	1900	2	⅝ × 1½	Yes	SV	Box
307	600	50	615	40	37	...	.671	2390	2700	5	33	4⅛	4	6 9/15	1900	2	⅝ × 1½	Yes	SV	Split
307-CV	600	55	595	50	52	52	.671	2390	2700	5	33	4⅝	4	6 9/15	1900	2	⅝ × 1½	Yes	SV	Box
307-V	600	55	595	50	52	52	.671	2390	2700	5	33	4⅛	4	6 9/15	1900	2	⅝ × 1½	Yes	SV	Split
307-CA	600	50	615	40	37	...	.671	2320	2630	5	33	4⅝	4	6 9/15	1900	2	⅝ × 1½	Yes	SV	Box
308	600	220	595	140	120	...	.088	6150	6740	7	36	4⅞	4	5 3/21	1400	4	⅝ × 1¾, 2¼	Yes	C	Box
308-B2	600	120	285	80	75	...	.374	6150	6740	7	36	4⅞	2¾	5 7/16	1400	3	⅝ × 1¾	Yes	C	Box
310-C	600	75	595	58	50	...	.423	3100	3510	6	33	4 13/16	3⅝	7 1/16	1850	2	⅝ × 1¾	Yes	C	Box
317	600	90	740	60	50	...	.262	3215	3660	6	33	3½	3 7/16	7 3/16	1850	2	⅝ × 1¾	Yes	C	Box
317-A	600	90	770	60	50	...	.262	3215	3660	6	33	3½	3 7/16	7 3/16	1850	2	⅝ × 1¾	Yes	C	Box
318	600	75	595	50	44	...	.462	3215	3660	6	33	3½	3 7/16	7 3/16	1850	2	⅝ × 1¾	Yes	C	Box
319-B	600	50	660	...	...	...	.671	2448	2865	5	33	4 11/16	3 15/16	7 1/15	2000	2	⅝ × 1½	Yes	C	Box
323-A	600	40	690	28	25	...	.965	1650	1890	4½	33	4¾	4	8 1/16	2200	2	½ × 1½	Yes	SV	Split
323-V	600	40	660	38	40	40	.968	1700	1990	4½	33	5⅛	4	8 1/16	2000	2	½ × 1¼	Yes	SV	Split
328	600	35	740	26	23	...	1.15	1490	1680	4	24	3¼	3	5 8/15	2400	2	½ × 1½	Yes	SV	Box
333-B2	600	115	800/665	80	75	...	.215/.261	3350	3870	6	33	5⅛	3⅜	6 1/16	1800	2	⅝ × 2¼	Yes	SF	Box
333-V	600	125	690	110	115	...	.206	3430	3850	6	33	4 13/16	3 5/16	6 1/16	1800	2	⅝ × 2¼	Yes	SV	Box
333-V8	600	125	735	110	115	112	.206	3430	3850	6	33	4 13/16	3½	6 1/16	1800	2	⅝ × 2¼	Yes	SV	Box
334-V8	750	120	730	85	88	89	.342	3430	3850	6	33	4 13/16	3½	6 1/16	1800	2	⅝ × 1½	Yes	SV	Box
340	600	48	660	39	39	...	.753	2125	2400	4½	26	2 13/16	2 13/16	5 7/15	2000	2	⅝ × 1¼	Yes	SV	Box
505-X	600	19	825	20	22	23	2.38	750	890	3	24	3¾	2⅞	6 8/14	2800	1	½ × 1¼	Yes	SV	Box
506-A2	600	25	1050	27	29	30	1.52	750	900	4	24	3 11/16	2¾	6 8/14	2800	1	½ × 1½	Yes	SV	Box
506-C2	600	25	1050	27	29	30	1.52	800	1000	4½	30	6 11/16	3¼	9 9/14	2800	1	½ × 1½	Yes	SV	Box
508-A	600	25	1235	32	35	35	1.10	870	1035	4	26	4 11/16	3¾	7 4/18	2800	1	½ × 1¾	Yes	SV	Box
508-C	600	25	1235	32	35	35	1.10	900	1100	4½	30	6 11/16	3¼	9 7/18	2800	1	½ × 1¾	Yes	SV	Box
510-A	600	35	1070	35	37	38	.886	1285	1475	4	26	3¾	3½	6 9/18	2400	2	⅝ × 1¼	Yes	SV	Box
512-A	600	35	780	33	35	35	.974	1360	1600	4½	24	3⅛	3⅛	5 8/15	2400	2	⅝ × 1¼	Yes	SV	Box
512-C	600	35	775	35	36	36	.974	1380	1600	5	30	5 9/16	3⅜	7 6/15	2400	2	⅝ × 1¼	Yes	SV	Box
514-A	600	40	760	35	36	37	1.00	1465	1650	4½	24	3⅛	3⅛	6 8/15	2400	2	⅝ × 1¼	Yes	SV	Box

GENERAL DATA ON WESTINGHOUSE DIRECT CURRENT MOTORS—(*Continued*)

Mfrs. No.	Volts	Nominal 1-hr. rating H.P.	Nominal 1-hr. rating R.P.M.	A.I.E.E. ratings continuous amperes at 1/2 voltage	3/4 voltage	Full voltage	Electrical resistance, ohms	Weight Without gears and gear case	Weight With gears and gear case	Max. axle diam., in.	Min. wheel diam., in.	Rail clearance with min. wheel Motor frame	Rail clearance with min. wheel Gear case	Max. gear ratio	Max. safe arm. speed, R.P.M.	Brushes No. per holder	Brushes Size, in.	Commutating pole	Type of ventilation (See Note, below)	Type of frame
514-C	600	40	760	35	36	37	1.00	1505	1770	5	30	5 11/16	3 3/8	79/15	2400	2	5/8 × 1 1/4	Yes	SV	Box
532-A	600	50	665	44	45	46	.745	2000	2300	4 1/2	26	2 13/16	2 13/16	57/15	2000	2	5/8 × 1 1/4	Yes	SV	Box
532-B	600	50	665	44	45	46	.745	2015	2325	5	33	6	4	69/15	2000	2	5/8 × 1 1/4	Yes	SV	Box
533-Y3	600	54	689/460	46	43	. . .	.673/.909	2270	2590	5	31	4 1/2	3	69/15	2000	2	5/8 × 1 1/4	Yes	SV	Box
534-Y1	600	58	785/590	48	48	. . .	.583/.724	2370	2690	4 1/2	31	4 1/2	3	69/15	2000	2	5/8 × 1 1/2	Yes	SV	Box
535-A	600	60	770	52	54	58	.462	2135	2400	5	26	3	2 3/16	57/15	2000	2	5/8 × 1 1/2	Yes	SV	Box
535-B	600	60	770	52	54	58	.462	2160	2475	5 1/2	33	6 5/8	4	69/15	2000	2	5/8 × 1 1/2	Yes	SV	Box
545-A6	600	67	585	61	62	62	.462	2725	3125	5 1/2	33	3 1/2	3 5/8	71/16	1850	2	5/8 × 1 1/2	Yes	F	Box
547-A	600	80	640	71	72	. . .	.358	2725	3100	5 1/2	33	3 1/2	3 5/8	71/16	1850	2	5/8 × 1 3/4	Yes	SV	Box
547-C	600	80	640	71	72	. . .	.358	2800	3175	6	33	3 1/2	3 1/2	78/16	1850	2	5/8 × 1 3/4	Yes	SV	Box
548-A	600	95	750	81	83	. . .	.248	2750	3125	5 1/2	33	3 1/2	3 5/8	71/16	1850	2	5/8 × 1 3/4	Yes	SV	Box
548-C8	600	100	810	90	93	93	.245	2775	3175	6	33	3 1/2	3 1/2	78/16	1850	2	5/8 × 1 3/4	Yes	SV	Box
557-A	600	140	800	130	135	. . .	.145	3630	4050	6	33	4 9/16	3 15/16	61/16	1650	2	5/8 × 2 1/4	Yes	F	Box
557-A8	600	140	900	130	135	135	.141	3630	4050	6	33	4 9/16	3 1/2	61/16	1650	2	5/8 × 2 1/4	Yes	SV	Box
562-A5	600	100	380	85	87	87	.363/.432	4480	4900	6	33	4 9/16	3 1/2	61/16	1550	2	1/2 × 2 1/4	Yes	F	Box
562-A6	600	100	325	120	125	120	.384	4480	4900	6	33	4 9/16	3 1/2	61/16	1550	2	1/2 × 2 1/4	Yes	F	Box
567-A6	600	165	640	130	135	130	.130	4480	4900	6	33	4 9/16	3 1/2	61/16	1550	3	5/8 × 1 3/4	Yes	SV	Box
567-R1	600	170	690/550	140	145	. . .	.114/.147	4480	4900	6	33	4 9/16	3 5/8	60/17	1500	3	5/8 × 1 3/4	Yes	SV	Box
577-A6	600	175	570	160	165	165	.1226	5200	5650	6 1/2	36	5	3 7/8	67/16	1500	3	5/8 × 2 1/4	Yes	SV	Box
577-R1	600	200	675/475	175	180	. . .	.110/.165	5200	5650	6 3/4	34	4	3 7/8	63/21	1500	3	5/8 × 2 1/4	Yes	SV	Box
577-D1	600	200	675/475	175	180	. . .	.110/.165	5200	5650	7	36	5	3 7/8	67/16	1500	3	5/8 × 2 1/4	Yes	SV	Box
552-A7	600	75	465	62	63	64	.535/.692	3630	4050	6	33	4 9/16	3 1/2	63/15	1650	2	5/8 × 2 1/4	Yes	F	Box
572-A5	600	150	440	130	135	130	.227/.305	5200	5650	6 1/2	36	5	3 7/8	67/16	1500	3	5/8 × 1 3/4	Yes	F	Box
572-A6	600	145	335	120	125	120	.253	5200	5650	6 1/2	36	5	3 7/8	67/16	1500	3	5/8 × 1 3/4	Yes	F	Box

NOTE: C—Closed. SV—Self-ventilated. F—Forced Ventilation.

General Data on Westinghouse Single Phase Alternating Current Motors

Mfrs. No.	Volts	Nominal 1-hr. rating		Weight with gears and gear case	Max. axle diam., in.	Min. wheel diam., in.	Rail clearance with min. wheel		Max. gear ratio	Max. safe arm. speed, R.P.M.	Brushes		Type of ventilation (see note)	Type of frame
		H.P.	R.P.M.				*Motor frame*	Gear case			No. *per* holder	*Size, in.*		
130	220	250	215	16,420	....	62	5½	Gearless		820	3	⅜ × 2⅜	F	Split
132-A	235	100	620	5,300	6½	36	5	4⅝	63/20	1,500	3	⅜ × 2⅜	SV	Box
132-F	235	100	600	5,300	6½	36	5	4¼	63/20	1,500	3	⅜ × 2⅜	F	Box
133	235	135	680	6,025	6½	36	4 9/16	3¼	73/17	1,400	4	½ × 2¼	F	Box
135-A	210	75	640	4,500	6	33	5	3½	69/17	1,600	3	⅜ × 2⅜	F	Box
137	235	240	360	14,485	8½	62	6½	6	85/16	1,000	3	⅜ × 2⅜	F	Box
148-A	235	125	700	5,926	6½	37½	5¼	4¼	65/28	1,350	4	⅜ × 2¼	F	Box
151-B	220	172	355	9,580	7½	50	5 11/16	4¼	79/17	1,000	3	⅜ × 2⅜	F	Box
156	225	150	630	6,208	9¾	42	6	4½	77/22	1,400	4	⅜ × 2¼	F	Box
403-A	300	310	440	14,375	8*	63	....	4 5/16	91/22	830	4	⅜ × 1¾, 2¼	F	Box
406	310	630	177	41,200	Rod drive	55	....	Gearless		407	5	⅜ × 2¼	F	Split
409-C	275	168	620	16,300	13	63	....	6 1/16	97/17	1,400	4	⅜ × 2½ (2)	F	Box
409-D	275	168	620	6,250	9	42	5¼	4¾	75/25	1,400	4	⅜ × 2½ (2)	F	Box
410	100	125	256	9,676	8*	63	....	5⅛	101/17	1,190	4	⅜ × 2¼ (2)	F	Box
412-A	300	225	760	6,638	7	38	4⅝	4¼	69/19	1,350	4	⅜ × 2¼ (2)	F	Box

Note: F—Forced. SV—Self-ventilated. * 13" Quill.

General Data on General Electric Direct Current Motors

Mfrs. No.	Volts	Nominal 1-hr. rating		A.I.E.E. ratings continuous amperes at			Electrical resistance, ohms	Weight		Max. axle diam., in.	Min. wheel diam., in.	Rail clearance with min. wheel		Max. gear ratio	Max. safe arm. speed, R.P.M.	Brushes		Commutating pole	Type of ventilation (See Note, p. 259)	Type of frame
		H.P.	R.P.M.	½ volt	¾ volt	Full volt		Without gears and gear case	With gears and gear case			Motor frame	Gear case			No. per holder	Size, in.			
800-B	500	19	325	.....	.....	.....	2.454	1615	1950	4 1/4	30	3 9/16	3 1/8	67/14	1680	1	1/2 × 2 1/4	No	C-NV	Split
1000-A	500	35	655	.....	.....	.....	.905	1905	2240	4 1/2	30	3 15/16	3 1/8	69/15	1800	1	1/2 × 2 3/4	No	C-NV	Split
52-A	500	27	640	.....	.....	.....	1.060	1500	1855	4 1/2	30	4 3/8	3 1/8	67/14	2100	2	1/2 × 1 1/4	No	C-NV	Split
54-A	500	25	638	.....	.....	.....	.872	1500	1855	4 1/2	30	4 3/8	3 1/8	67/14	1650	2	3/8 × 1 1/2	No	C-NV	Split
57-A	500	50	575	.....	.....	.....	.496	2896	3301	4 1/2	33	3 1/2	4 1/8	69/16	1830	2	5/8 × 1 3/4	No	C-NV	Split
58-A	500	37	505	.....	.....	.....	.677	1865	2190	4 1/4	30	2 3/4	2 3/4	69/15	1830	2	1/2 × 1 3/8	No	C-NV	Split
67-A	500	40	518	.....	.....	.....	.637	2075	2445	4 1/2	30	3	2 3/4	69/15	1840	1	1/2 × 3 3/8	No	C-NV	Split
70-A	500	40	502	.....	.....	.....	.638	2400	2850	5	33	4 1/16	3 15/16	71/15	1840	2	1/2 × 1 7/8	No	C-NV	Split
73-C	500	75	520	.....	.....	.....	.221	3630	4070	5 1/2	33	3 1/2	3 1/2	73/17	1500	2	1/2 × 2 1/2	No	C-NV	Box
80-A	500	40	510	.....	.....	.....	.637	2415	2850	5	33	4 3/8	3 15/16	71/15	1840	2	1/2 × 1 7/8	No	C-NV	Split
81-A	500	30	610	.....	.....	.....	.838	1640	2030	4 1/2	30	4 3/8	3 1/8	67/14	1800	2	3/8 × 3	No	C-NV	Split
87-A	500	60	500	.....	.....	.....	.347	2940	3380	5 1/2	33	3 1/2	3 7/8	71/16	1700	2	1/2 × 2	No	C-NV	Split
88-A	500	40	480	.....	.....	.....	.653	2550	2990	5	33	4 3/16	3 15/16	71/15	1840	2	1/2 × 1 7/8	No	C-NV	Box
90-A	500	50	600	.....	.....	.....	.378	2435	2875	5	33	4 3/8	3 15/16	71/15	1840	2	7/16 × 1 7/8	No	C-NV	Split
200-K	600	40	745	31.7	32.7	32.8	.9170	1800	2160	4 1/2	30	4 3/8	3	67/14	1950	2	9/16 × 1 1/8	Yes	V-SF	Split
201-G	600	65	710	45.6	45.3	44.1	.4869	2370	2805	5	33	4 1/4	3 3/4	71/15	1650	2	9/16 × 1 3/4	Yes	V-SF	Box
203-P	600	50	760	42.0	44.8	46.0	.5875	1865	2200	5	33	5 3/8	3 7/8	83/15	1800	2	9/16 × 1 23/64	Yes	V-SF	Box
205-B	600	110	625	.....	.....	.....	.2552	3345	3855	6	33	3 3/4	4	58/16	1590	2	5/8 × 2 1/4	Yes	C-RD	Box
207-A	600/1200	145	620	.....	.....	.....	.1637	4500	5110	6 1/2	33	3 7/16	2 13/16	64/17	1400	3	5/8 × 2	Yes	C-RD	Box
210-C	600	70	537	.....	.....	.....	.5020	2570	3100	5 1/2	33	4 9/16	3 15/16	71/16	1590	2	1/2 × 1 5/8	Yes	C-RD	Box

212-B	600	235	620	.....	.....	.....	.0940	5340	5970	6 1/2	33	3 3/8	3 1/16	6 4/19	1400	4	5/8 × 2	Yes	C-RD	Box
216-A	600	50	635	.....	.....	.....	.7300	2390	2830	5	33	5 1/4	3 13/16	7 1/15	1950	2	1/2 × 1 1/64	Yes	C-RD	Box
219-A	600	50	635	.....	.....	.....	.7300	2390	2820	5	33	5 1/4	3 13/16	7 1/15	1950	2	1/2 × 1 81/64	Yes	C-RD	Split
222-G	600	140	700	92.1	98.8	101.0	.1590	3570	4175	6	33	3 1/4	3 3/8	6 1/17	1475	3	5/8 × 1 13/16	Yes	V-SF	Box
225-B	600/1200	100	660	79.6	84.1	85.3	.1975	3250	3850	6	33	3 1/4	2 9/16	6 3/15	1482	2	9/16 × 2	Yes	C-SF	Box
240-A	600/1200	95	662	77.6	81.0	83.5	.3	3170	3755	6	33	3 9/16	3 1/2	7 3/17	1590	2	5/8 × 2	Yes	V-MF	Box
247-A	600	35	758	33.8	35.2	35.8	1.067	1510	1720	4	24	3 5/16	3 1/8	5 8/15	2160	2	1/2 × 1 1/4	Yes	V-MF	Box
248-A	600	160	659	132.0	138.5	143.0	.1425	5050	5630	6 1/2	33	3 1/2	3 3/8	6 9/17	1400	3	1 1/16 × 2 1/8	Yes	V-MF	Box
249-A	600	40	670	36.6	41.0	44.6	.8730	1910	2285	4 1/2	30	3 5/8	2 5/8	8 8/15	1800	2	5/8 × 1 1/8	Yes	SV-LSF	Split
251-A	600/1200	212	530	.....	.....	.....	.1194	5320	6005	7	36	4 3/4	3 3/16	6 8/18	1400	3	5/8 × 2 3/8	Yes	V-SF	Box
254-A	600	140	740	118.2	125.0	128.2	.1448	3855	4428	6 1/2	33	3 1/4	4	6 2/17	1485	3	5/8 × 1 13/16	Yes	V-MF	Box
257-A	600	165	605	.....	.....	.....	.1465	4440	5060	6 1/2	33	3 7/16	2 13/16	6 5/17	1400	3	5/8 × 2 1/4	Yes	C-RD	Box
258-C	600	25	1236	32.4	34.9	35.2	1.2168	746	896	4	24	3 7/16	2 11/16	7 5/18	2500	1	9/16 × 1 7/8	Yes	V-MF	Box
259-A	600	120	842	103.0	107.0	109.5	.2010	3400	3915	6 3/4	33	4 1/2	3 9/16	6 1/16	1540	3	5/8 × 1 3/4	Yes	V-MF	Box
260-A	600	195	632	160.0	171.6	177.5	.1299	5125	5665	6 3/4	33	3 1/2	3 3/8	6 9/17	1400	3	11/16 × 2 1/8	Yes	V-MF	Box
263-A	600	65	725	63.5	68.0	70.5	.49	2545	3010	5	33	4 1/4	2 13/16	7 1/15	1650	[illegible]	9/[illegible] × 1 3/4	Yes	V-MF	Box
264-A	600	25	1236	32.4	34.9	35.2	1.2168	844	994	4	24	3 7/16	2 11/16	7 5/18	2700	1	9/16 × 1 7/8	Yes	V-MF	Box
265-A	600	35	1125	36.5	39.0	40.5	.78	1134	1311	4	24	3	2 11/16	6 9/14	2200	2	9/16 × 1 1/4	Yes	V-MF	Box
269-A	600	55	905	54.0	58.0	59.0	.723	2044	2404	5	33	5 1/8	3 7/8	8 3/15	1830	2	5/8 × 1 1/2	Yes	V-LSF	Box
270-A	600	55	840	51.0	54.5	56.0	.7252	1975	2324	4 1/2	33	5 1/8	3 7/8	8 3/15	1830	2	5/8 × 1 1/2	Yes	V-LSF	Split
275-A	600	50	720	45.0	47.0	48.0	.546	2175	2405	5	26	2 15/16	2 13/16	5 7/15	1800	2	5/8 × 1 5/8	Yes	V-MF	Box

NOTE: C—Closed. NV—No ventilation. SV—Semi-ventilated. V—Ventilated. RD—Radial Ducts. SF—Series Fan. LSF—Large Series Fan. MF—Multiple Fan.

Two or Four Motors per Car. In the choice between two and four motor equipments for double truck cars, the determining factor usually is the coefficient of adhesion, *i.e.*, whether or not the available traction of two sets of driving wheels will give the required rate of acceleration, although another consideration is that of reliability or the proportion of total motive power lost by the disabling of one motor. Where there are grades of more than five per cent, or where rail conditions are poor, as in climates with bad winter conditions, the adhesion limitation usually will dictate the four motor equipment for single car operation. With two motor equipments the weight available for traction may be increased greatly by the use

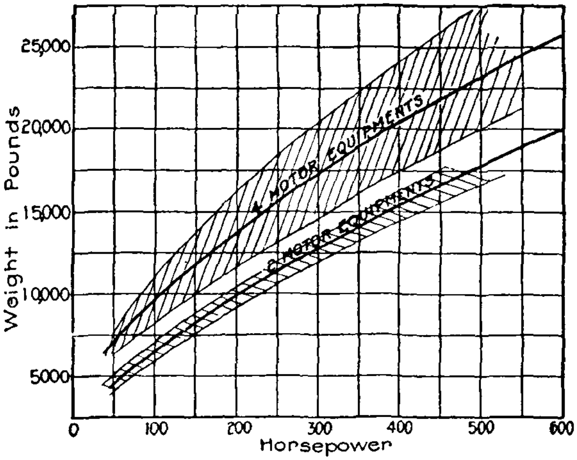


FIG 35—Comparative weights of two and four motor equipment.

of maximum traction trucks. Where conditions are favorable for the use of the two motor equipment, its less weight, energy consumption and maintenance cost frequently dictates its selection. The relative weights of two and four motor equipments of various capacities are shown by Fig. 35. The energy consumption of the two motor equipment will be less than the corresponding four motor equipment due to less weight and to the better efficiency of the larger motors. The number of men required for inspection and overhauling of motors is more nearly proportional to the number of motors than to their capacities, while the relative cost of repair material and replacement parts is not at all proportional to motor capacity; the resultant maintenance costs with two motor equipments is estimated as 25 to 35 per cent less than with corresponding four motor equipments.

Selection of Gear Ratio. If the gear ratio be *increased*, the speed of the car at a given current will be reduced, the tractive effort at the driving wheel treads will be increased, and the possible rate of acceleration will consequently be increased. If the gear ratio be *reduced*, the speed of the car at a given current will be increased, the tractive effort at the driving wheel treads will be reduced, and the possible rate of acceleration will consequently be decreased. Thus, with a low gear ratio, greater current will be required to produce a given tractive effort and acceleration than would be required with a higher gear ratio. (For effect of gear ratio and electromotive force on speed and tractive effort characteristics see p 246.) The proper gear ratio to be used with a given car depends upon the service the car is to render and the character of the track over which it is to be operated. Whether or not a gear ratio once proper will be proper at a future time will depend upon whether or not the service requirements and character of the track remain constant, for example, a route may at one time require a few stops widely separated and on it high maintained maximum speeds may be permitted or necessitated; later this same route may require many stops close together, thus permitting only low maximum speeds. A gear ratio satisfactory in the first case would be unsatisfactory in the second. Frequently a car is required to give both city and suburban or interurban service or a combination of the three. It is in such cases that unsuitable gear ratios are probably most often used—the common error being to use a gear ratio lower than is necessary. This is brought about by not giving the requirements of city service due consideration while seeking high speeds. Unless field control be resorted to, a compromise based upon the different classes of service required must be made. Where it is possible to use field control the low speed service will largely determine the proper gear ratio. The proper gear ratio is the one with which, other things remaining constant, the required service is afforded at a minimum cost for energy and maintenance. As above noted, this will be determined by the conditions to be met, but *in general the proper gear ratio is the highest with which the necessary schedule can be maintained* (with allowance for low voltage and making up lost time) without establishing a dangerously high armature speed on descending grades. Various authorities have estimated that by the installation of proper gearing in existing equipments from 5 to 15 per cent of the energy now used for traction may be saved. This saving is accompanied by a decrease in maintenance costs and the schedules are not impaired by the change. The following table from tests on the Brooklyn Rapid Transit System shows that by increasing the gear ratio a saving of 10 per cent in energy consumption was secured and the original schedule was nearly maintained even though the number of stops per mile was increased.

	Tests with 3.58 gears	Tests with 4.12 gears
Average number of stops per mile	7 2	10.3
Average number of slow-downs per mile	7.9	7.2
Schedule speed, miles per hour	8.2	8 1
Weight of car, tons	16.22	18.11
Average passenger load, pounds	3040	2860
Weight of car and average load, tons	17.77	19.54
Seating capacity	38	38
Watt hours per car mile	2672	2640
Watt hours per ton mile	150.4	135.1

Where stops are infrequent, as in suburban and interurban service, the possibility of maintaining comparatively high maximum speeds for a considerable length of time, together with the possibility of long coasting periods, make rapid rates of acceleration and braking of minor importance and a comparatively low gear ratio may be used to advantage. Where stops are frequent, as in city service, there is little or no operation at high maximum speeds, high rates of acceleration and braking are necessary and there may be little coasting. If a low gear ratio is to be used in such city service, a larger motor must be employed than would be necessary with a higher gear ratio and there will be a waste of energy. The use of a too low gear ratio with motors which would be large enough if a higher gear ratio were used causes the motors to operate in series or on resistance during a too great part of the time. This is equivalent to operating the motors on a voltage lower than normal. The motors are overloaded, and, in some cases, burned out, rheostats and controllers are overworked and, in some cases, generating stations and substations are overloaded and the load factor is abnormally low. All this is accompanied by a waste of energy.

Figs. 36 to 41 by N. W. Storer show some of the effects of different gear ratios when applied to a certain motor equipment. The example is a specific one in which it is desired to determine the best gear ratio to use with equipment already in hand. In the preparation of these curves the following conditions were assumed:

Weight of car loaded	40 tons
Number of motors	4
Size of motors	75 h.p.
Line voltage	500 volts
Size of wheels	33 in.
Length of run	0.6 mile
Schedule speed	20 miles per hour
Length of stop	10 seconds
Rate of braking	1.5 miles per hour per second.

Figs. 36, 37 and 38 show the results obtained with a constant accelerating current of 138 amperes for all gear ratios. Figs. 39, 40 and 41 show the results obtained when accelerating at a constant rate of 1.5 miles per hour per second for all gear ratios. Fig. 36 shows the time required to run with power on, time for coasting and the time for the maximum accelerating current; or, in other words, the time for running on resistance. The time for the maximum accelerating current varies from 37.5 seconds, with the 2 : 1 gear ratio, to 6.8 seconds with 4.5 : 1 gears. At the same time, the rate of acceleration varies from 0.8 mile per hour per second to 2.1 miles per hour per second. Fig. 37 shows that the average voltage of the motor while power is on is only 310 volts with 2 : 1 gears, while it rises to 480 volts with 4.5 : 1 gears. Motors are overloaded when

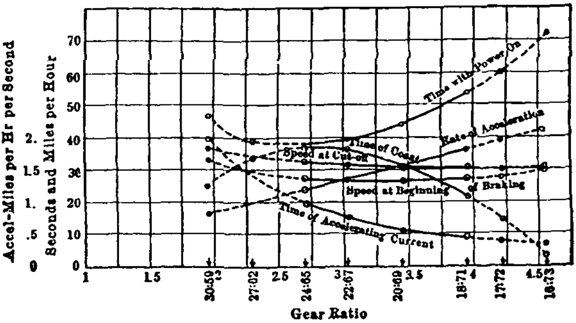


FIG. 36.—Typical performance curves of car with different gear ratios and constant accelerating current.

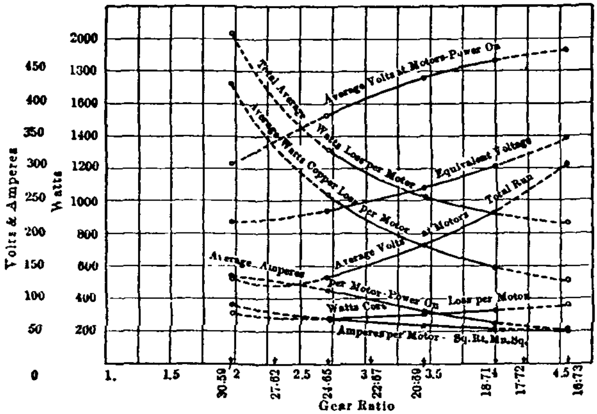


FIG. 37.—Typical performance curves of car with different gear ratios and constant accelerating current.

the voltage drops badly, and this is just as true when the average voltage applied to the motor is lowered by changing the gear ratio. This fact will be more readily appreciated when it is seen that the average rate of taking current from the line is 130 amperes per motor with the 2 : 1 gear and only 52 amperes with the 4.5 : 1. The root mean square current per motor is 87 amperes for the 2 : 1 gear and 49 amperes for the 4.5 : 1 gear. The curves show average watts loss in the motor in copper and iron and the total electrical loss. They show that the *motor losses* will be a *minimum* with the maximum gear reduction. (The maximum gear reduction possible for this schedule would not be commercial as there is absolutely no margin for making the schedule, but it is shown simply to point

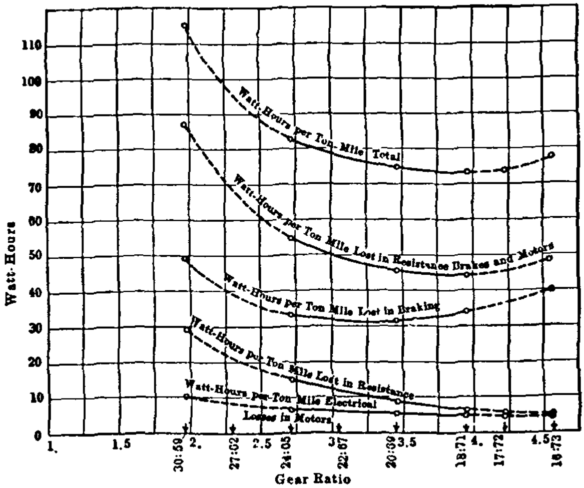


FIG. 38.—Typical performance curves of car with different gear ratios and constant accelerating current.

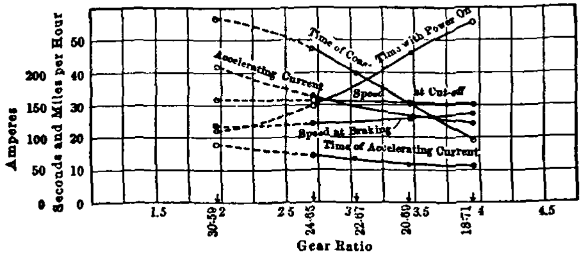


FIG. 39.—Typical performance curves of car with different gear ratios and constant rate of acceleration.

out the fact that the greater the gear reduction for a given schedule the lower will be the losses in the motor.) The curves in Fig. 38, however, show that the *energy consumption* is a minimum at about 4 : 1 gear reduction. The increase in watt-hours per ton mile between 4 : 1 and 4.5 : 1 is coincident with and due to the increase in speed at the time the brakes are applied, as shown in

Fig. 36. The rise in watt-hours per ton mile with the 2 : 1 gear ratio is due to the increased loss in resistance and motors as well as in the brakes. The motor loss, it will be seen, is controlled almost

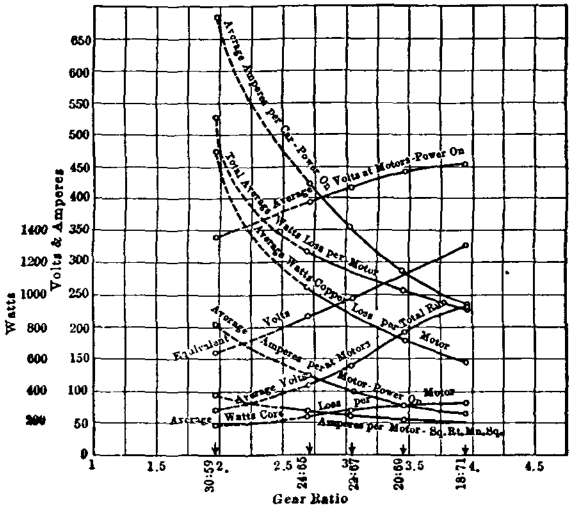


Fig. 40.—Typical performance curves of car with different gear ratios and constant rate of acceleration.

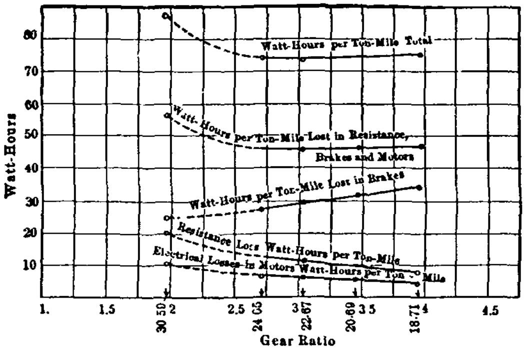


Fig. 41.—Typical performance curves of car with different gear ratios and constant rate of acceleration.

entirely by the root mean square current, as the iron loss is small at all times and remains close to 300 and 350 watts for all gear ratios. Fig. 39 corresponds to Fig. 36, Fig. 40 to Fig. 37 and Fig. 41 to

Fig. 38. A comparison of these curves indicates that while a constant rate of acceleration produces a nearly constant energy consumption per ton mile, the motors will be very much more overloaded with the lower gear ratios at the *constant rate of acceleration* than when accelerating at the *constant current*.

Commutating Poles. In the railway motor the commutating pole (also called the interpole) is a small pole piece placed between the main poles with its windings in series with the armature. The total weight of copper in a commutating pole motor is nearly the same as that in the non-commutating pole motor because the introduction of commutating poles makes possible a reduction in the number of field turns. The function of the interpole is to improve commutation at all loads and voltages on the motor. It accomplishes its purpose by providing a field which when swept through by the armature coil short-circuited by the brush will produce a complete current reversal in that armature coil so that there will be no spark when contact is broken between the commutator bar and the brush which it is leaving. Incidentally the commutating pole prevents or diminishes the possibility of flash overs by restricting rushes of current. The commutating pole is permanently connected directly in series with the armature, and whenever, as for the purpose of changing the direction of rotation, the direction of the current to the armature is changed the direction of the current in the commutating pole winding must be changed. The current in the commutating pole winding varies with that in the armature, consequently, if the motor operates successfully at one load it will operate successfully at any other load up to the point of saturation of the commutating pole. The commutating pole operates most satisfactorily at below saturation. At loads considerably above that whose current causes saturation of the commutating pole the influence of the commutating pole is annulled by the flux due to the current in the armature and with a still further increase of load commutation would be worse than with no commutating-pole.

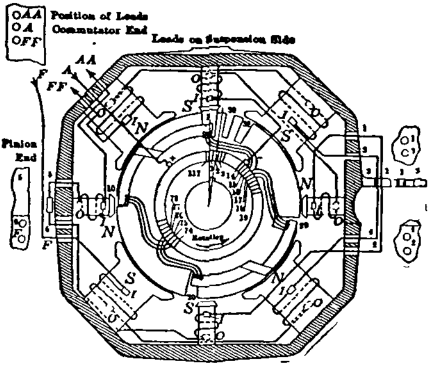


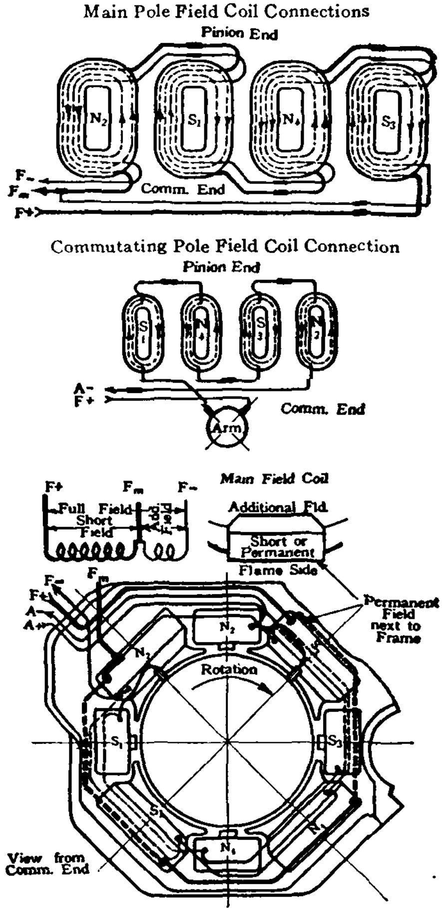
Fig. 42.—Circuits of typical commutating pole motor.

Commutating poles make possible a reduction in the air gap, with a consequent reduction in the weight of the motor, and raise the limiting voltage between segments to a considerably higher value, thus making possible the design of motors for operation on higher voltages. Further, the inductance of the commutating pole windings tends to reduce the number of flashovers by limiting the flow of surge currents. The practical elimination of sparking results in many operating advantages. The short-time ratings of motors have been materially raised, thus permitting the use of higher rates of acceleration. More rapid acceleration in turn results in decreased energy consumption and in many cases in decreased motor heating. Commutator maintenance costs are greatly decreased, as reflected by the fact that the sale of replacement commutators by manufacturers has practically ceased since the general introduction of interpole motors. Brush life has been increased about 50 per cent on the average as the result of the decreased flashing and sparking, and the accompanying decrease in commutator losses has reduced motor heating to some extent. Briefly summarized, the introduction of the interpole has made available a lighter, cheaper, more reliable type of motor. At the same time, the partial removal of important limitations in design has made possible the development of successful types of high-speed, high-voltage and field control motors.

The circuits in a typical commutating pole motor are shown by Fig. 42.

Field Control. The circuits of a typical field control motor are illustrated by Fig. 43. The idea of securing additional efficient *running speed by varying the effective field turns on a railway motor* is an old one, but on account of commutation difficulties it did not prove successful until the commutating pole was developed. The field control motor has the field arranged in two parts which are connected in series with a lead brought out from the point where they are joined together, *FM*, Fig. 43. In starting, the motor current passes through both parts of the field in series, called the full field connection, setting up a very strong field and developing a large torque with a relatively small current. When desired, one portion of the field is cut out, leaving only the permanent or short field in the circuit, so that a weaker field results and higher speeds are secured. This arrangement tends to economy and flexibility in that less starting current is required and the motor has two efficient running connections instead of one. Field control is very generally used on heavy multiple unit equipments, locomotives and for the largest car motors. It is not uncommon to cut out as much as one-half the field coils for the maximum speed.

Any well-designed commutating pole railway motor may be adapted for field control by a proper arrangement of its field windings. To get the full benefits, however, the gears must be properly selected. For interurban work the benefits of field control may be secured by the use of standard high-speed armatures with a larger gear reduction than usual. In most cases, also, sufficient space is available to permit the extra field winding to be used. Special armatures for use with field control are necessary only for cases where the slowest speeds and the maximum gear ratio are required. It usually will be found that where a motor of a given size is used in city service with the maximum gear reduction and the usual series-parallel control a slower speed armature may be used with the same motor frame and will make the same schedule with a lower energy consumption when field control is employed. The motor will have a lower horse-power rating, but the current used will be correspondingly less, and, consequently, the motor will have no more loss in it than with the motor of higher speed with a larger rating. In other words, the use of field control permits



Dotted lines show connections at pinion end.

FIG. 43.—Circuits of typical field control motor.

the use of a motor of a smaller rating for a given service. Where the maximum gear ratio is used in both cases, the same size of frame must be used. However, where the gear reduction can be increased for the field control motor it will frequently be found that a smaller size of motor can be used at a lower first cost and with less weight to be carried around. A double saving will thus be effected.

Ventilation of Railway Motors. The losses in a motor, which appear in the form of heat, result in a motor temperature that is a function of the magnitude of those losses, conditions of ventilation and (in some cases) the weight of the motor. On loads essentially constant over a time period of several hours, the weight of the motor has little or no effect upon the final motor temperature. Under these conditions, the temperature rise is almost entirely

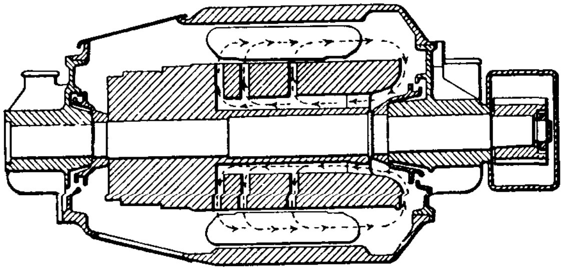


FIG. 44.—Internal ventilation—armature without fan—radial vent ducts. No air inlet in frame.

dependent upon the conditions of ventilation. On short time loads, the temperatures at any given time will depend upon both the ventilation and weight of motor per watt loss. The greater the overload the greater will be the influence of the weight of the motor in limiting the temperatures; since under these conditions a great majority of the losses must be stored up in the copper and iron of the motor. Also under these overload conditions, the maximum temperature reached will depend upon the initial temperature. A ventilated motor will naturally have a greater continuous capacity than an enclosed one of the same size; or conversely, of two equally rated motors, the better ventilated one will weigh the less. Hence, for railway work, where the minimum motor weight is desirable, the ventilated motor is a logical development. With regard to ventilation, there are three general classes of railway motors, namely: totally enclosed, self-ventilated, separately-ventilated.

A totally enclosed motor is one so enclosed as to prevent circulation of air between the inside and outside of the case or frame, but not sufficiently to be termed air-tight. Such a motor is sometimes called a non-ventilated motor, since all of the heat losses must finally be dissipated from the external frame surfaces by radiation and convection air currents. A totally enclosed motor may, however, have a system of internal ventilation such as shown in Figs. 44 or 45. This is highly desirable since such a system of ventilation will greatly aid in the transfer of the heat losses from the armature to the frame and housing surfaces.

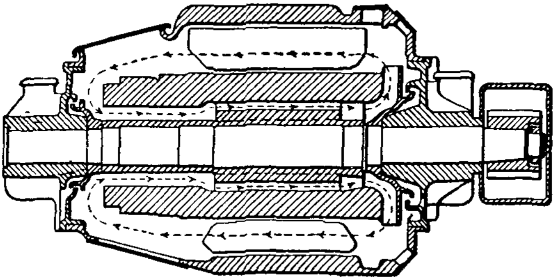


FIG. 45.—Internal ventilation—armature with fan—longitudinal vent ducts—air inlet and outlets closed.

A self-ventilated motor is one in which the ventilating air is circulated through the machine by a fan, blower or centrifugal device integral with the machine. Two common systems of self-ventilated motors are shown in Figs. 46 and 47. In both schemes, the fan mounted on the pinion end of the armature acts as an exhaust fan and pulls the air through the intake and ventilating passages

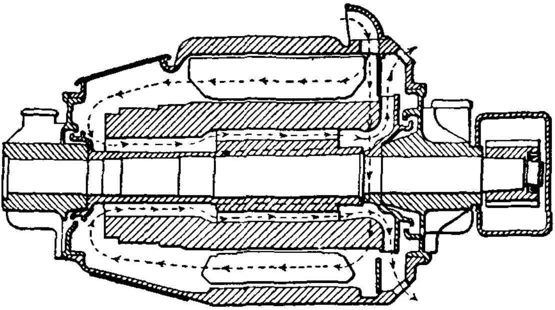


FIG. 46.—Series ventilation—armature with fan—longitudinal vent ducts—air inlet at pinion end, top side of frame.

and discharges it through the outlet openings in the frame. The parallel or multiple system of ventilation as shown in Fig. 47 is the better system since the air paths are shorter, and due to greater cross-sectional area for the air flow more air will be pulled through than is possible with the series system of ventilation as shown in Fig. 46. Since the losses in the armature range from 60 to 70 per cent of the total, it is desirable to have the armature well ventilated with the air entering it as cool as possible, and this condition is best met by the parallel ventilation as shown in Fig. 47. On motors of this type the entrance of foreign material, such as dirt, water or snow, should be prevented as far as feasible. This is usually done by some type of a baffle intake, one form of which is shown in Fig. 47. The entering air is forced to change its direction very abruptly and on so doing the heavier foreign particles are dropped. The parallel system of ventilation as shown in Fig. 47 will increase the continuous rating of a motor 50 per cent or more over its totally enclosed rating. A self-ventilated motor may be made totally enclosed by changing the intake and outlet covers to solid covers, as shown in Fig. 45. In this case the fan is beneficial in providing internal ventilation. This practice is adopted by some of the electric railway systems in the northern part of the country for

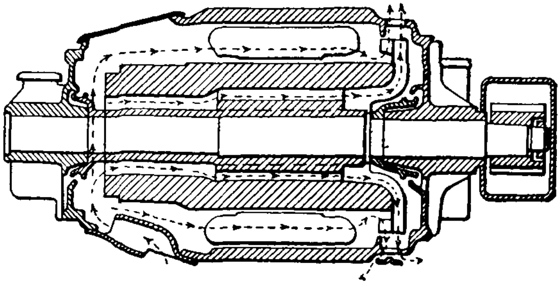


Fig. 47.—Parallel ventilation—armature with fan—longitudinal vent ducts—air inlet at commutator end under side of frame.

winter operation, where snow and ice troubles are especially severe, but it should be done only when the air temperatures are low, or the motor may be overheated.

A separately-ventilated motor is one in which its ventilating air is supplied by an independent fan or blower external to the machine. With a self-ventilated motor, the amount of air which will be pulled through the motor depends upon the size of the fan and the speed of the motor; on slow speed the air circulated will be very limited, and when the motor is stationary there will be no air circulation outside of small natural convection currents. On account of the above limitations, a separate blower is used where a maximum rating is desired. Thus, separately-ventilated motors are being used on the majority of locomotive motors. The air supply to the motors is independent of the motor speed, and since the air is usually taken from the locomotive cab, it is considerably cleaner than air picked up near the road bed as with self-ventilated motors. Where the motors are axle-hung the air connection between the motor and the air conduit must be a flexible one, a common type of connection being the canvas "accordion" connection. The air intake to the motor can be provided by one or more holes at either end, although the more usual position is at the commutator end, since more room is available there. The air paths through the motor ducts are usually similar to those shown in Fig. 47; there being a parallel air flow through both the armature and the fields.

A recent development in connection with the ventilation of non-ventilated type railway motors has been made in which an outside fan is mounted on an extension of the armature shaft at the commutator end of the motor. Air is drawn in at the pinion end of the motor, passed through and over the armature and between the field coils, over the commutator, then through openings in the specially designed commutator end housing, and finally is exhausted by the external fan mounted on the shaft extension. Some of the results in connection with this method of revamping a non-ventilated type of motor as worked out on two 60-h.p. motors were as follows: continuous rating of motors increased 30 to 40 per cent; one hour rating of motors increased 7 per cent; from 38 to 40 per cent reduction in temperatures observed during a service test. Cases are cited where by the addition of this ventilating device motor cars have been able to handle trailers during rush hours, which was not possible otherwise on account of excessive heating.

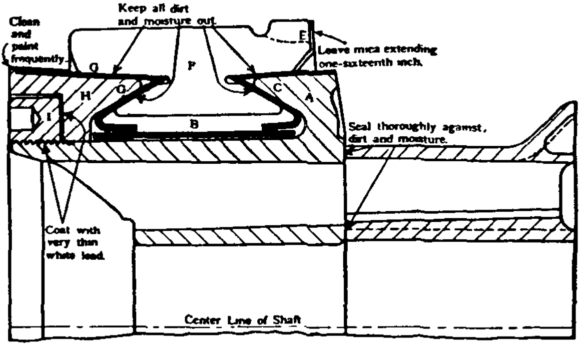


FIG. 48.—Section of an assembled commutator.

Commutator Material and Construction. Hard-drawn copper is recommended for commutators as possessing greater uniformity in size and hardness over the forged bars and corresponding superiority in life and service. Attention is also called to the importance of material and workmanship in the construction of commutators being such as to insure a solid structure which will not shrink, become loose, or get out of true. Built-up mica is preferable for commutator segments, but it is very important that the building-up should be even and compact. Assembled commutators should be baked at 230 deg. C. and compressed while hot to insure solidity, clamps being tightened before pressure is released.

Care of Commutators. The band over the front V-ring of a commutator should be wiped off every month. After cleaning, painting with an air drying varnish makes cleaning easier the next time. Painting every six months is advisable. Flat spots, high or low bars, ridges, burned spots, etc., should be smoothed up. Where these are not bad, the motor need not be removed from the car. A tool can be made by mounting a block of wood on a stick, one face of the block being cut to the radius of the commutator and lined with sand paper or stone. As the car is run by other motors, this tool, held against the commutator, will smooth the rough spots. The armature should be removed from the car and placed in a lathe for turning the commutator, if its face is very rough. Holes left by defective mica or pits in the side of bars can be filled with commutator cement supplied by dealers.

Undercutting. A clean undercut commutator prevents high mica, reduces the burning of the commutator bars, increases the life of the brushes, and practically eliminates flashing. This results in greatly reduced maintenance cost. The object of undercutting commutators is to clean out the mica between the copper segments from the face of the commutator to a depth of $\frac{3}{64}$ inch. The completed armature, after being rewound, soldered and banded, has the commutator trued up, and all excess solder removed from the neck and face, and is then undercut. If undercutting were done before soldering, the excess solder might fill up the grooves and short-circuit the commutator bars.

The most efficient machines for undercutting are special lathes with motor-driven or belt-driven circular saws, clamped on an arbor which is mounted on a head that moves on slide rails. By means of a hand-operated lever or a foot pedal controlled by the operator, the revolving saw is carried over the face of the commutator. The head carrying the arbor is fitted with an adjusting screw to adapt the height of the saw to commutators of different diameter. An air hose and hood is sometimes provided to carry off the mica dust. A less expensive equipment consists of an air or power-driven (through a flexible shaft) circular saw, which is guided over the commutator face by hand. The armature is mounted so that it can be rotated readily between the centers of the special lathe, or on two horses if the hand-guided saws are used. The circular saws are revolved at approximately 2000 r.p.m., and are drawn toward the operator so that he can guide the cut. The cutting edge of the saw should revolve in a direction toward the operator. After slotting is completed, the face of the commutator should be thoroughly polished and cleaned of all particles of copper by means of emery cloth. The special lathe can be so equipped that this polishing is done without removing the armature from the lathe centers. It is essential that all particles of mica be removed from between the segments; thus it is advisable to use a saw about 0.005 inch larger than the thickness of the mica to be undercut. A small diameter saw must be used in order to cut the slot to the proper depth and at the same time not cut into the neck of the commutator. After the slots are sawed, it is sometimes necessary to go over them with a small hand cutter to remove all remaining particles of mica.

An undercut commutator should operate from one heavy inspection period to another. However, if an armature is removed from the motor frame for any other repairs, the commutator should be carefully inspected, and if the mica is getting flush with the copper it should be re-undercut. The commutator should be re-undercut before it is worn flush, since the groove left will guide the saw and make the work much easier. Where commutation trouble is frequent, it is good practice to use a V-shaped hand tool to round the edges of the under-cut grooves between bars to about 1/32 inch radius. This can be done with the motors in the car. All particles of mica, copper, or dirt should be removed from the grooves after undercutting.

Commutator Repairs. A thin drift, driven in the top of the slot in the neck toward the shaft between the side of the commutator neck and the top filling piece, will loosen the filling piece so that it may be forced out by a gouge. Similarly, windings may be taken out of the first bar. With one set of leads removed, there is enough space left to bend one side of the next neck, thus permitting the removal of the leads by means of the gouge. This can be done without heating the soldered joints. Where a small number of bars are to be replaced, it is not necessary to remove all the leads from the armature neck or take the commutator from the shaft. Stand the armature on end, commutator up. Mark each separate piece so that it may be put back in its old position. After removing the ring nut or bolts, take out the metal V-ring and mica V-ring. If the bars are tight, tapping with a wooden or rawhide mallet will loosen them. The new bar must be filed to the exact shape or thickness of the one which it replaces to prevent the new bar or the old bars next to it from becoming loose. Clean out the space where the new bars fit and tap them in with a soft mallet.

Detached parts of the commutator must be kept clean and dry. When the parts are ready for rebuilding, sand paper the mica V-ring. Clean the metal V-ring and the V in the commutator bars, and paint the V in the bars with shellac. The shellac and brush must be absolutely free from all dirt or moisture. After putting the mica and the metal V-rings back in their original positions, the ring nut or bolts should be drawn up fairly tight. Painting the threads of the ring nut or bolts with very thin white lead will make it easier to remove these parts next time. The commutator should now be heated to 110 deg. C. in an oven where the air is dry, and the ring nut or bolts drawn up tight while the commutator is hot. The commutator should then be turned in a lathe. Test after these repairs should be at 110 volts alternating current to ground.

When it is necessary to replace the rear mica V-ring, a number of segments, or a complete set of segments, it is advisable to remove the commutator from the shaft. The method of taking down and rebuilding is the same as before described, care being taken to heat the commutator thoroughly to soften up the new mica parts, so that the ring nut or bolts can be drawn up tight. To this end, when the commutator is assembled, it should be put in an oven and heated to a temperature of 125 to 140 deg. C. While at this temperature the commutator should be placed in a press, using a pressure of 20 to 25 tons for a 50 hp. motor, and the ring nut should be drawn up tight while under pressure. Complete sets of segments are shipped, temporarily banded together, with the mica and copper segments in their proper position. The complete set should be assembled in the commutator as a unit, and the temporary band should be removed just before the commutator is finally tightened. Test after these repairs should be at 300 volts alternating current between commutator bars and 2000 volts alternating current to ground.

Brushholders. The brushholder castings should have at least $\frac{3}{4}$ inch air clearance from the motor frame. Railway motor frames have machined pads for the clamping blocks to hold the brushholders exactly in the neutral position. On replacing brushholders the clamping blocks must be seated properly. The carbon box should be parallel with the commutator bars, and the distance from the center line of one box to the center line of the other box on a four-pole machine should be equal to one-fourth of the number of commutator bars. The coil short-circuited by the brush should be midway between the main poles; ordinarily this means that the brush will be opposite the middle of the main poles. This spacing is adjusted at the factory and will be correct if the clamping blocks are seated properly and the brushholders are those belonging to the motor and are not bent. The underside of the carbon box should be kept within $\frac{1}{8}$ to $\frac{3}{16}$ inch (preferably $\frac{1}{8}$ inch) of the commutator surface, so as to prevent breakage of carbons. A piece of tapered fiber $\frac{1}{8}$ to $\frac{3}{16}$ inch thick makes a convenient gage. The nut which holds the clamping block should be carefully tightened so that the brushholder will not come loose.

Spring pressure should be about $4\frac{1}{2}$ lb. per sq. in. of contact surface for large motors of say 150 hp., up to $7\frac{1}{2}$ lb. per sq. in. for small motors of 25 to 30 hp. Factors that may modify this pressure are the amount of sparking, the play in the armature bearing and the track conditions. To prevent breakage and side wear of the carbon, it should fit in the box as snugly as possible without binding. This clearance should not exceed $\frac{1}{32}$ inch. The shunt formerly used on the carbon has been replaced by a shunt on the brushholder which connects the contact tip on the finger and the main casting. This shunt must be large enough to carry the current and make a good electrical contact where fastened. If a good electrical path is furnished for the current through the shunt, much of the side wear of the carbons and burning of the carbon box is prevented. It is therefore very important to keep the shunts in good condition. The object of the contact tip is not only to transmit pressure to the top of the carbon, but to conduct current between the shunt and the carbon. Its shape should be such that it will do the least damage to the top of the carbon and provide a large contact area to carry the current. Cable leads should be supplied with sleeves or terminals. These connections should be kept tight to secure a good electrical as well as a mechanical contact. Set screws with jam nuts and lock washers should be used. When carbon dust and dirt get into the working parts of a brushholder, it is likely to work stiffly. These parts should, therefore, be kept clean and occasionally lubricated. If neglected, they may become tight and not allow enough pressure, or become so badly worn that the parts get out of line and the tips press on the carbon box instead of on the carbon. The keeping of railway brushholders in good repair and proper alinement is a large factor in cutting down maintenance costs, as it lengthens the life of the carbons, commutators and motor windings by improving commutation, thus tending to prevent flashing between brushholders and ground. For this reason it is very important that brushholders be regularly inspected and overhauled.

Carbon Brushes. When making a selection, the following few fundamentals, in connection with operating conditions, should be considered: commutation; life of commutators; life of brushes; frequency of inspection; current density; cost per car mile. The above factors will vary more or less with the design of the motor, design of brushholder, correct spacing of brushes, condition of roadbed, condition of equipment, condition of armature bearing, condition of commutator surface, brush tension, weight of car, number of motors per car, schedule speed, service conditions, etc. Since all of these factors must be taken into consideration, the best and most reliable results are obtained by making tests of recommended grades in service under actual operating conditions. Carbon manufacturers generally are willing to assist the operator in making service tests. The following general information regarding carbons will help the operating man more intelligently to select carbons for service tests.

Carbon is a non-metallic element found in both crystalline and amorphous form. Natural graphite is carbon in a crystalline form, and is mined in many localities. Amorphous or non-crystalline carbon may be obtained in the form of coke or lamp-black. Artificial graphite is obtained by heating amorphous or non-crystalline carbon, such as coke, in an electric furnace to change its structure to a crystalline state. By the use of these materials, the following general classes of brushes are made: carbon brushes, made of crushed coke and binder; graphitized brushes, made of carbon and then electrographitized; graphite brushes, natural graphite and binder; metal graphite brushes, natural graphite with metal powder and binder. With modern commutating-pole motors having commutators undercut, and considering costs per car mile, brush end wear side wear, breakage, life, commutation, life and maintenance of commutator, etc., these classes of brushes are best suited for railway motor service as follows: graphitized brushes, best all-round results; carbon brushes, next best; graphite brushes, very special and limited uses; metal graphite brushes, not used. In the process of manufacture the most important operations, depending upon the class of brush being manufactured, are as follows· crushing, carbonizing (if it is done) and cooling; milling, mixing, cooling and re-milling; moulding and packing in the furnace; gas baking; electric baking or graphitizing. The two most common general methods of manufacture are: (1) extruded or squirted—where the material in the form of pulp is forced through a metal die under pressure and then cut off to the desired length and baked at a high temperature to carbonize the bond and permanently set the material; this method is used in making the cheaper grades of carbons which do not have the strength to resist breaking and chipping in service; (2) moulded and machined—where the material is moulded under heavy pressure into blocks, and then baked; the carbons are cut from these blocks and machined to exact size; this method is used in making high grade carbons and gives a brush of uniform texture and strength that is best suited for railway work. In the manufacture of various grades of carbon brushes best suited to meet the requirements of operating conditions, the following characteristics of the brush must be considered: contact drop; coefficient of friction; heat conduction; hardness; apparent density; conductivity; resistance; abrasiveness; toughness.

For use in modern brushholders, the carbons should be not over two inches long. That is, when new, they should not extend over the top of the carbon box, for the following reasons: if longer, they are subjected to a greater side pressure, due to the action of the contact tip, which increases the side wear, tends to bind the carbon in the box and reduce the direct pressure on the surface of the commutator; if longer, they are discarded due to excessive side wear before the added length can be used up in end wear; approximately the same mileage can be secured from the shorter carbon and since the carbons are bought on a cubic inch basis, the cost is less. The width is not so important. The brush can have as much as $\frac{1}{16}$ inch clearance in the box without causing any trouble in service. Thickness is very important, the initial clearance betwen carbon and carbon box should be approximately from 0.006 to 0.008 inch. If it is much less, the carbon will tend to stick in the box and bind, and if greater, it will soon rattle in the box, wearing away the side; it will also tend to chip and break, thus reducing its life.

When shunts or pig tails were used on brushes, it was considered necessary to copper-plate the carbon to provide a good electrical contact for the shunt connection. With the present design of brushholder having a heavy braided copper shunt from contact tip to carbon box, shunted carbons have been discontinued, so that copper plating is unnecessary; in fact, it is objectionable on the higher grades of carbons as it tends to peel off in service and bind the carbon in the box. For railway work, shunts or pig tails on carbons have been practically discontinued, for the following reasons: first cost, they were used with the cheaper grades of carbons that had a comparatively short life, which required renewal of carbons at frequent intervals; inspection, the pit-men could not be depended upon to maintain shunts during inspection; not reliable, the design was such that shunts became loose and disconnected. With these conditions greatly improved by the use of a higher grade of carbon with longer life and requiring less frequent inspection, and by improved methods of fastening pig tails to carbons, some advantages can be obtained by the use of pig tails; and in certain specific cases, especially for very heavy current densities, they have been adopted with a saving in maintenance. Foreign practice tends to the more extensive use of shunts than is customary in this country.

Flashing of Motors. Flashing of railway motors, also commonly known as bucking or blowing, is primarily caused by poor commutation which results in a sudden breakdown of the insulation over the face of the commutator from the brushholder to the motor frame or ground. As a result, there is a sudden rush of heavy current which either opens the circuit breaker or hangs as an arc, and badly burns the short-circuited parts. The results of flashing are varied. Sometimes the motor is so badly damaged that it is inoperative. In this case, the motor must be overhauled, armature windings repaired, commutator cleaned up, brushholders and wiring around frame put in good condition. When the short circuit is immediately cleared, the motor generally can be continued in service to finish its run, but should be reported as defective, and given a careful inspection for any damaged parts that need attention. A detailed layout of the various conditions that affect commutation of a railway motor and, in turn, may cause flashing, are as below:

Conditions That Tend to Produce Flashing

Design	Non-commuting pole type	
	Sensitive neutral	
	High voltage between commutator bars	
Commutators	Rough face	High bars
		Low bars
		Loose bars
		Flat spots
	Poor surface condition	Poor undercutting of mica
		Sharp edges on commutator bars
		Sharp corners on commutator bars
		Dirty commutators
Brushholders	Lack of spring tension	Broken spring
		Weak spring pressure
		Pressure finger sticking
		Worn mechanism
	Incorrect setting	Too far from commutator surface
		Incorrect spacing between brushholders
		Out of alinement with pole
	Worn carbon box	
	Loose in clamping block	
	Too small clearances to ground	
Carbons	Inferior grades	
	Length	Too long
		Too short
	Clearance	Loose in carbon box
		Tight in carbon box
	Broken	
Windings	Wiring around frame	Loose brushholder connection
	Armature	Wrong connection
	Main field	Short-circuited coil
		Reversed coil
		Field control—wrong connection
		Design of coil bobbin or case
	Commutating-pole field	Short-circuited coil
		Reversed coil
Operation	Sudden voltage changes	Faulty control
		Rapid acceleration
		Plugging motors
		Heavy operation
		Breaks in third rail
	High speed running	
	High trolley voltage	
	Overheating of motors	
	Loose or worn bearings	Too much end play
		Too much side play
	Rough track	
	Flat wheels	

Removal of Armature from Box Frame Motor. Several methods of removing armatures from box frame motors are in common use. A scheme used by a large operating company requires one short and one long piece of pipe, bent for convenience. Two rings that slip over the ends of the armature shafts are hinged at one end of each piece of pipe, about six inches apart. The motor is set on the floor and both housings removed. The rings on the long piece of pipe are slipped over the commutator end of the armature shaft, while the rings on the short piece of pipe are slipped over the pinion end of the shaft. The men at the end of each piece of pipe lift the armature bodily out of the frame through the pinion end opening, and let it down on a grooved block made to receive it. The number of men required to handle an armature by this method depends upon its size and weight.

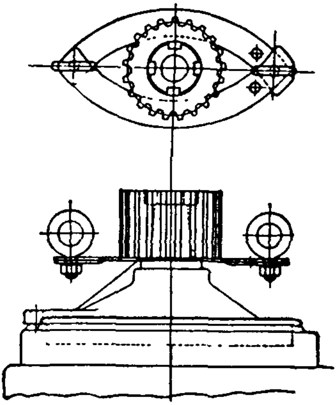


Fig 49—Yoke for armature removal.

A machine designed for removing armatures consists of two heads fitted with centers—one long, which is stationary, and the other short, which is adjustable—mounted on the end of a base plate. The long center must have a diameter slightly smaller than journal size at the commutator end of the armature, and should be long enough to extend through the length of the motor frame. A carriage to receive the motor travels on the base plate between the two heads. The motor is set on the carriage of the machine, with the commutator end toward the machine head with the long center pin. The pinion end housing bolts are removed and the center points are entered in the centers of the armature shaft and locked. The pinion end housing is then removed and the carriage supporting the frame is pushed toward the machine head carrying the long center pin. The armature being held rigid, the long center pin passes through the commutator end bearing opening. In this manner the frame is shifted far enough to clear the armature, which can be lifted to the floor after being released from the center pins. One man and a crane operator are required to handle an armature by this method.

A very quick and simple way of doing this work, but one which requires a careful and skilled crane operator, is as follows: The oil from the bearing housing oil wells must first be removed, after which the motor is up-ended, commutator end down. The pinion end housing bolts are removed, and a special threaded cap fitted with an eye bolt is screwed on the pinion nut fit of the armature shaft. The crane hook is attached to the eye bolt, and the armature is lifted out of the motor frame. When the armature is being let down on the floor, it is necessary to rest the commutator end of the shaft on a block to prevent damaging the commutator and end windings as it turns to a horizontal position. With this method one man and a crane operator can handle the largest motors used on city or interurban cars. In Fig. 49 is shown a yoke that is attached between the pinion and housing, replacing the special threaded cap. This scheme works well; the yoke is easily made and can be used on any size motor.

The facilities and number of men available will, to a very great extent, be the deciding factor as to which method will be best suited to individual cases.

Armature Winding. Great care should be used to guard against failure by ground, short circuit, open circuit, or loose bands. Sharp corners and roughness in the slots that might damage the coils should be filed down and all chips and filings removed before applying the insulating material to the core. In applying the insulating material on the coil supports, the material should be evenly placed, and no thin spots allowed. The coils should fit tightly in the slot, and so that the top of the coil is about $\frac{1}{32}$ inch above the band groove. Fillers should be used between the coils if necessary to meet these conditions, which will permit the bands to pull the coils down tightly and at the same time finally rest on the iron, which allows the minimum movement of the coils in the slots. If the coils are wound too high, the bands will not rest on the iron, and when the insulation dries out in service, loose bands will result. The coils should not be twisted, bent or abused any more than is absolutely necessary to get them in place. Care should be taken not to get the wires or leads crossed in such a manner that, when pressure is applied in banding, short circuits will occur. The coils on the end windings should be down, so as to make a solid foundation for the bands, but pounding should not be carried to the extent that the coils or leads are damaged. Insulating protecting pieces should be placed at all points where the coils cross, and where there is danger of short circuit. It is good practice to weave braid between the leads directly back of the commutator, both on the top and bottom layers. The leads should be tinned back to such a distance that there is no untinned copper in the commutator neck. The cotton sleeving on the leads should not be allowed to get into the commutator slot, as it may hinder soldering to such an extent that a poor connection will result. The tool used in driving the leads into place should be free from sharp corners that might nick the leads, as this may result in a broken lead.

Layout of Armatures and Commutators. Most direct current windings used in railway work are of the wave or two-circuit type. In laying out such a winding it is necessary to know the throw of the coils and the throw of the leads. The throw of a coil is the distance spanned on the core designated in terms of the slots, that is, if the coil throw is stated as 1 and 8, the bottom of the coil lies in slot No. 1, while the top of the coil is in slot No. 8. In a similar manner, the throw of the leads is the distance spanned on the commutator in terms of the commutator. When there is an odd number of leads per coil, or if, when a dead coil is taken care of, an odd number of leads remain, locate the center between the slots indicated by the coil throw. If the lead throw is an odd number of bars, this center will line up on a commutator bar, and if it is an

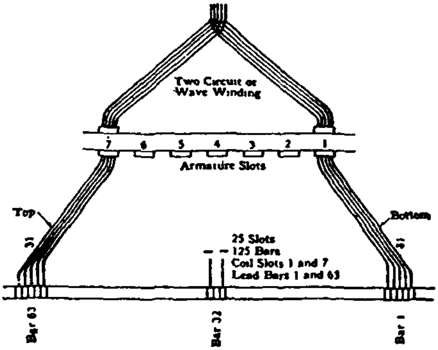


FIG. 50.—Layout of armature winding with an odd coil throw and an odd throw of leads.

even number of bars, it will line up on the mica between bars. This bar or mica is the starting point for laying off the commutator. If there is an odd number of bars in the throw, take one less than the number of bars and count off half of this number each direction

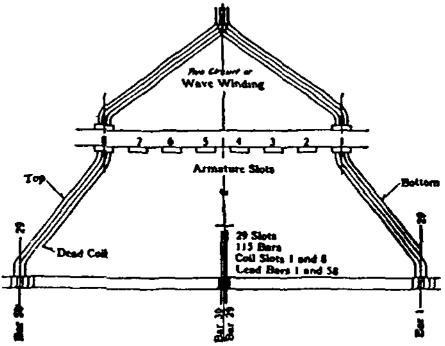


FIG. 51.—Layout of armature winding with an even coil throw and an even throw of leads.

from the starting bar, and this will give the first and last bar of the commutator throw. If there is an even number of bars in the throw, count off half the number in each direction from the starting mica. As the first coil put down will have an odd number of leads, the center one of the top and bottom leads should be placed in the first and last bar of the throw as determined. Sample layouts are shown in Figs. 50 and 51. A slightly different method is necessary when there is an odd number of coils per slot and an even number of slots, which case, however, very seldom occurs. If the lead throw is an odd number of bars, the center, as indicated by the coil throw, will line up on the mica and, if an even number of bars, it will line up on a bar. If there is an odd number of bars in the throw, take one less than the number of bars and count off half this number to the left and one more than half to the right, and this will give first and last bar of the commutator throw. If there is an even number of bars in the throw, count off half the throw to the right and one less than half to the left. If there are two leads in the first coil, No. 1 lead should lie in No. 1 bar, and if there are four leads in the first coil No. 2 lead should lie in No. 1 bar.

The alinement of commutator bar and mica with armature tooth and slot center lines is as follows: *coil throw even, lead throw even*—mica with tooth; coil throw odd, lead throw odd—bar with slot; coil throw even, lead throw odd—bar with tooth; coil throw odd, lead throw even—mica with slot.

These rules are of value: the wave or two-circuit winding always requires an odd number of commutator bars; when an even number of coils per slot are used, there will always be an idle or dead lead; when an odd number of coils per slot are used and there are an even number of slots on the armature, there will always be an idle or a dead lead; when an odd number of coils per slot are used and there are an odd number of slots on the armature, there will never be an idle or a dead lead.

Soldering Leads to Commutator. For armatures operating under normal service conditions, and not subjected to high temperature and unusual mechanical strains due to high speed, which tend to throw solder from the commutator necks and armature bands, half-and-half solder can be used with good results. When motor equipments are subjected to high temperatures and high speed, pure tin should be used to solder the armature leads to the commutator necks, and to solder the armature bands. When tin is used for soldering, it is necessary to have the clearance between parts to be soldered as small as possible. It is important that the flux does not contain any acid, as the acid may get to the insulation of the coils and cause short circuits and grounds. A good, cheap and safe flux is made by mixing one pound of rosin in a quart of either wood or denatured alcohol.

Banding Armatures. Since the coil insulation shrinks when heated, it is necessary to shrink it as much as possible before the final banding wire is applied. This is done by heating the whole armature to about 75 deg. C., when the insulation becomes pliable and can be pressed down into permanent shape. The hot armature is then mounted in a lathe, protecting strips of cloth are placed over the end windings, a temporary banding wire is wound over the coils with enough tension to draw them down into place, and the ends are fastened by soldering tin clips over the wire. The armature is then allowed to cool. After the temporary wires are removed, the armature is ready for the permanent banding. When a banding machine is used, the tension in the wire is regulated by passing it over a train of friction pulleys mounted on the carriage. The friction of the pulleys can be adjusted to any desired value by the regulating screws. In the absence of such a device, fair results can be obtained by passing the wire two or three times around a round wooden banding stick approximately two inches in diameter and adjusting the tension by hand. When core bands are used, the grooves are fitted with thin strips of tin, which protect the coils from the cutting action of the bands. In starting the permanent banding, a few turns are first made at one end to secure the necessary tension. All the banding groups are wound on continuously, to eliminate the necessity of fastening the ends of each group as they are wound. The bands are held together and the ends are fastened by means of narrow tin strips placed under the wires, bent back over the top and held by pure tin solder. These strips are inserted while the wire is being fed on and are located about every three inches around the armature, with closer spacing at the beginning and end of each band. For the core bands, these strips are placed in the slots and, being wider than the groove, they prevent any tendency for the bands to slide around the armature. The ends of the band wire groups are then cut and secured by being bent back outside one clip and inside the next one. Pure tin solder is applied to the whole surface of the bands, forming a solid web. The end windings are secured by groups of wire which are wound on insulating hoods to protect the coils. On the commutator end, strips of thin mica with overlapping ends are usually placed on the commutator neck and held in place with a few turns of twine. The drilling hood or head is wrapped around the neck, extending about an inch from the edge and placed *inside out*. After this end is secured with twine, the free end of the hood is pulled back over the windings, bringing the outside of the hood at the surface and making a neat folded-under edge. This hood is also held temporarily with twine until the wire is applied. The other end of the armature is similarly covered with a hood and banded.

The proper tension varies with the size of wire and the construction of the end windings. When the end coils have no rigid support and extend out considerable distance from the core, the tension is gradually reduced, as shown below:

Pounds Tension for Banding Wires

Diameter of wire, inches	Core bands	End bands	
		At core	At end of winding
0 0450	200	175	160
0 0641	300	250	225
0 0803	400	300	260

The best banding material is a high-grade steel piano wire, having a final breaking strength of 200,000 pounds per square inch. For temporary bands a cheaper grade can be used. Pure tin solder should be used, as this gives a band that will hold together for a longer time than half-and-half solder Tin clips and strips should be of commercial sheet tin about 0 012 inch thick.

Broken Armature Leads. Some companies have experienced a great deal of trouble from this source, in some cases continual, and in others sporadic The table below lists the probable causes of such trouble.

PROBABLE CAUSES OF BROKEN LEADS

Vibration	Internal	Loose armature windings	Poor workmanship
			Poor foundation
			Shrinkage of insulation
		Loose armature core	Worn key
			Poor fit on shaft or spider
			Poorly clamped
		Loose commutator.	Worn key
			Poor fit on shaft or spider
			Poorly clamped
			Not thoroughly baked
		Poor banding	Not well soldered
			Poor grade wire
			Too few bands
		Loose coil support	Worn key
			Poor fit on shaft or spider
		Excessive current	Overload
			Rapid acceleration
			Faulty handling of car
		Out of balance	Core unbalanced
			Commutator unbalanced
			Windings unbalanced
		High peripheral speed	
	External	Armature bearings	Loose
			Worn
		Armature bearing housings	Loose
			Worn
		Axle bearings	Loose
			Worn
		Gear case	Broken
			Loose bolts
		Axle caps	Broken
			Loose bolts
		Gears and pinions	Badly worn
			Out of mesh
			Eccentric
			Bottoming
		Track	Special work
			Poor conditions
		Non-resilient roadbed	
		Rigid mounting of motor	
		Faulty handling of cars	
		Flat wheel	
Overloads		Heavy currents.	Magnetic stresses
		Overheating	Shrinkage of insulation
			Loosening of band wire
			Expansion and contraction of winding
Leads damaged		Leads, nicked	Driving into slot
			Sharp corners on commutator slot
		Metal changed	Action of heat in soldering
			Action of the solder
		Metal crystallized by pounding	
Material (copper)		Mechanically weak—no elasticity	
		Hard copper	
		Impurities in metal	
		Defects in drawing	
Localized strains		Sharp bends in windings	
		Uneven placing of leads	
		Improper banding	
		Poor distribution of bands	

Armature Record Tags. The use of record tags similar to that shown by Fig. 52, of a standard size such as $5\frac{3}{8} \times 2\frac{5}{8}$ inches, preferably made of cloth, provides an accurate record of armature troubles, armature repairs, location of armatures in the equipment, and a means of analyzing armature and motor failures, it gives a record of life of motor parts, and helps to weed out defective frames and troubles due to incorrect winding and motor connections, requires little clerical work, and assists in figuring cost of repairs on armatures. The suggested wording may be modified to suit individual conditions.

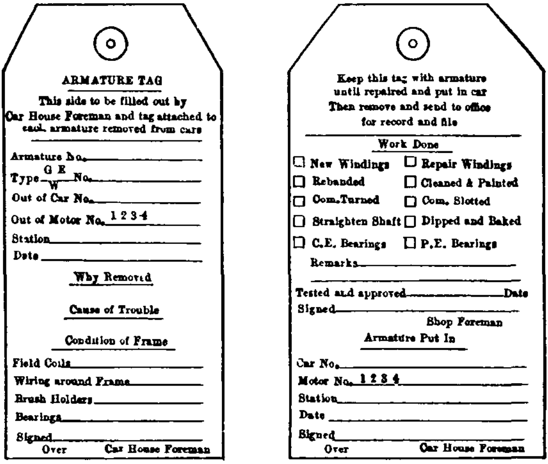


FIG. 52.—Suggested armature tag.

Field Coils. In general, the four coils have the same number of turns on each. On a few machines, however, the top and bottom coils have more turns than the two side coils, in order to shorten up on the gear centers of the motor. On commutating pole type railway motors, four commutating coils are most commonly used, in which each coil has the same number of turns. With small size motors, owing to the mechanical limitations in design, and to make use of some electrical advantages, the necessary commutating magnetic flux is obtained by the use of only two or three commutating coils. There is no standard arrangement of leads that can be applied to all field windings. In the more modern motors it has been necessary to place the leads at the end of the coils on account of lack of clearance for the commutating coils. By the use of open and crossed coils the wiring around the frame is simplified. This requires carrying in stock two types of coils, with the added chance of getting coils mixed in replacing damaged coils. To provide against this, these coils are marked with an "X" for crossed coils, and an "O" for open coils. It is considered best practice in connection with modern motors and is recommended for old motors, to use cables coming out of the body of the coil instead of heavy brass terminals for the wiring around the frame connections. The objection to the use of terminals is that they are more likely to break off due to vibration, or to develop a loose connection and burn off the lead. Another disadvantage is the difficulty of properly insulating the terminals after the connection is made.

Fig. 53 shows a common arrangement of coils for a field control motor (short or permanent coil B and added coil A), assembled together and having four leads. Generally, the permanent coil has about 60 per cent of the total number of turns, while the added coil has the remainder. In some cases the proportion is as low as 50–50. As the added coils are worked only a part of the time, the section of the copper conductor in this coil is made smaller, approximately 75 per cent of the section used on the permanent coil.

The following precautions are worth following while overhauling or repairing field windings: coils and frame should be cleaned and painted; coils should be placed on poles properly; coils should be

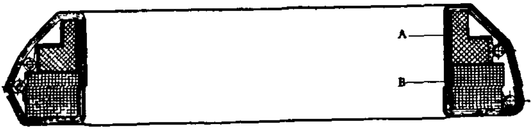


FIG. 53.—Section through main field coil.

spring supported, and where necessary, backed up by washers of either metal or fibrous material, well painted; in replacing coils, temporarily tape the spring and washers to the coil to keep them from working out of place and getting between the pole and pole seat when assembled; clean the surface of the pole and pole seats, to insure a good close fit when the bolts are drawn up tightly; do not pound the pole in place with a sledge—use a block of wood or a piece of soft metal, such as copper; be sure that the pole bolts are drawn up tightly and that lock washers are under each nut; add some white lead over the nuts after tightening, to prevent entrance of water; sound the assembled pole, using a small hammer, to insure that the poles are drawn up tightly; connections must be made as shown on the winding diagram; check the polarity of each coil; the connections between coils and with cable leads should be made by butting the ends together, covering the joint by a copper sleeve and soldering well; when the coils have terminals, the ends of the cable connected to the terminal should have a metal sleeve soldered to the wire, which is held in the terminal by screws securely tightened and locked; the ends of the cables connected to the brushholders should be provided with metal sleeves or terminals that are to be securely clamped and locked to the brushholder casting; all wiring around frame should be securely anchored to the coils and tied down, to prevent vibration and to keep the insulation from being rubbed or cut by parts of the rotating armature; dip and bake the individual coils;

to secure best results, dip and bake the entire frame after the coils are in place and all connections are made; the motor leads should be well protected by insulating bushings where they come out of the frame.

Aluminum Field Coils. The Lind method of winding and insulating aluminum field coils for direct current motors was introduced into this country about three years ago, after fifteen years' use in European countries. The aluminum coils are wound with bare square aluminum wire insulated with aluminum oxide. A layer of paper is placed between coil layers to facilitate even winding and to serve as a wick for distributing the oxidizing solution and a finish coating of insulating varnish. Lack of other insulating materials between turns and layers permits a considerably increased cross-sectional area of conductor. The characteristic relations between the cross-sectional areas and corresponding resistances of aluminum and copper, and the per cent saving in weight, are such that for the aluminum coil resistance to equal the copper resistance, the cross-sectional area of the aluminum conductor must be increased 59 per cent, and when increased by that amount the saving in weight is 52 per cent. For railway motor field coils wound with round copper wire, the aluminum cross-section can be made large enough to give coils of equal or lower resistance than the standard copper coils. For most strap-wound coils, the same is true, although for some the resistance is increased somewhat, but for coils wound with rectangular or square copper, the copper resistance cannot be met within the usual tolerances.

Testing Polarity of Field Coils. A reversed main field coil will cause the armature to run hot, due to the unbalanced magnetic circuit; a reversed commutating coil will cause poor commutation and flashing. The apparatus required for testing polarity is a polarity detector, such as a small compass, a switch resistance (such as several car heaters, grid resistors, or headlight resistor), a fuse or circuit breaker, and, until the resistance has been determined properly, an ammeter. The resistance should be such that the current is sufficient to give a readable deflection on the polarity indicator, but not above the rating of the motor. The motor may have the armature in or out of the frame and, if a split frame, may be open or closed. With the coils all connected in series, connect the two field leads to the test circuit. When the switch is closed, the polarity indicator, when held close to the ends of the coil, or to the pole stud bolts on the outside of the frame, will reverse at alternate main poles; *i. e.*, if No. 1 pole attracts the positive end of the polarity indicator, No. 2 should attract the negative end, No. 3 the positive end, and No. 4 the negative end. This test can be made by rubbing a screw driver on the pole bolts and then holding it to the compass needle or polarity indicator. If these conditions are not obtained, the field winding connections should be changed. In the case of a commutating pole machine, it is important to have the proper relation of polarity between the main and commutating field poles. To check this, connect the negative armature lead of the motor to the positive field lead, the positive armature lead to the trolley side of the test circuit, and the negative field lead to the ground side of the test circuit, and close the switch. If the armature is in the frame and the brushes are making contact on the commutator, current will flow through all the windings; if the armature is not in the frame, it will be necessary to short-circuit the brushholders. With these conditions, the polarity of a main pole should be the same as the polarity of the commutating pole next to it in a clockwise direction when facing the commutator end of the motor.

Tests of Rewound Armatures. The following list gives the tests *in the order of their application to completely rewound armatures* by the Chicago Rys. Co: (1) Breakdown test between commutator and sleeve as assembled off the machine; (2) breakdown test between the commutator and sleeve, with commutator on the shaft; (3) bar to bar test on commutator; (4) after coils are in slots, with lower leads connected to commutator; breakdown test, commutator to ground; (5) after coils are in slots, with lower leads connected to commutator; breakdown test between adjacent leads, (6) after coils are entirely connected in and before soldering; breakdown test, commutator to ground; (7) after coils are entirely connected in and before soldering; short circuit tests on armature coils; (8) after banding, short circuit and open circuit tests on armature coils; (9) after banding, breakdown test to ground.

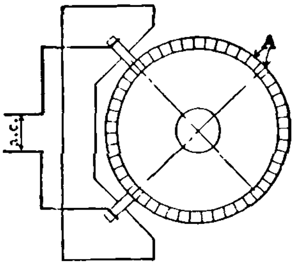


Fig. 54.—Test of armature for short circuit.

Locating Short Circuits in Armatures by Direct Application of Alternating Current. The following method described by Leonard Work, *Electric Journal*, 1910, is stated by him to be rapid and thorough (Fig. 54). Alternating current is applied to the commutator through contacts separated by an amount equal to the correct brush separation for ordinary operation of the motor. While this continues, connection is made and broken between any pair of adjacent bars as at *A* on the side of the commutator opposite the points where the alternating current is applied. If on such making and breaking there is a small spark, there is no short circuit between these adjacent bars. If there is no spark, a short circuit exists between these two bars. The points at which the alternating current is applied to the armature are advanced around the armature and the test is continued between each pair of adjacent commutator bars. In making this test, approximately full load current should be applied. The test is made with the armature removed from the motor frame.

Armature Test by Full Load Current. In the shops of the Brooklyn Rapid Transit Company armatures removed from the motor frames are tested for poor soldering, defective leads and short circuits in commutator or armature winding by passing full load current through the armature by way of a pair of brushes set in the same relative positions as for actual operation. The current flowing through the armature and the voltage drop between adjacent commutator bars are measured. Poorly soldered connections and abraded leads are burned off by the current The testing outfit consists of a rheostatic controller and a set of car resistances arranged to give any desired current from 8 to 300 amperes, an armature circuit breaker and switch.

Insulation Test of Armature Coils. Between windings and ground·

For roads using trolley:

New armatures	2500 volts, a c , 5 seconds
Old armatures	1000 volts, a c , 5 seconds

For roads using third rail where voltage fluctuations have to be taken into consideration:

New armatures .	3000 volts, a.c , 5 seconds
Old or partly repaired armatures .	1500 volts, a.c., 5 seconds

Between armature coils before windings are connected:

New armatures	220 volts, d c , 5 seconds
Old armatures .	110 volts, d.c., 5 seconds

Equipment for Armature Insulation Testing. The Chicago City Railway Company uses a motor generator set consisting of a single phase, 12-kw., 230-volt, 125-cycle generator excited by a 1.5-kw , 125-volt belted exciter and driven by a 550-volt, 1875-rev. per min. direct current motor. The testing electromotive force is obtained by the use of a 5-kilovolt-ampere 6000-volt transformer arranged to give a voltage of from 200 to 6000 in twelve steps.

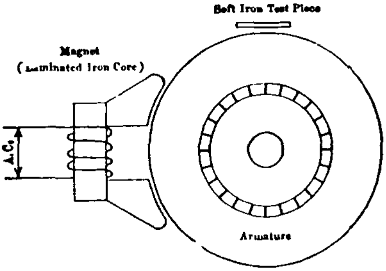


Fig 55 —Yoke transformer test for armatures.

Locating Short Circuits and Open Circuits in Armature Coil by Yoke Transformer Test (Fig 55). A "U" shaped two-pole laminated iron core having its pole pieces cut to embrace the armature as is done by the motor field pole pieces, and carrying an exciting coil about its yoke, is placed against the armature to be tested. When the coil on the yoke is excited by an alternating current, the resulting flux passes through the armature core and its windings and induces an electromotive force in the latter. The armature coils which are in a position to link with the greatest number of lines of magnetic force have induced in them the highest electromotive force and if one of these coils be short-circuited a heavy current will flow through it and this will be evidenced by heating of the coil, or by the noisy vibration of a soft iron test piece laid over the armature. Manifestations remain the same when the two adjacent commutator bars to which the terminals of a short-circuited coil are connected are short-circuited by having a piece of metal inserted between them and the testing current is momentarily applied The test is continued around the armature by revolving it so that one coil at a time will come under the influence of the strongest field The coils not found to be short-circuited are next tested for open circuit This is done by short circuiting adjacent commutator bars, pair at a time and applying the test for short circuit to the corresponding coil An indication of a short circuit when this is done proves the absence of an open circuit in the coil In testing for open circuit the vibration test should be used and the current should not be kept on long enough to damage the coil under test The alternating electromotive force applied to the testing coil may be obtained by connecting it to a convenient lamp socket. For ease in handling, the testing yoke may be bolted to the top cross piece of a two-wheeled warehouse truck arranged so the truck frame will stand vertically when tipped up This arrangement permits the testing yoke to be quickly moved to the armatures which are mounted on wooden horses of such height that the armature core center shall be even with the center of the testing yoke when the truck is tipped up

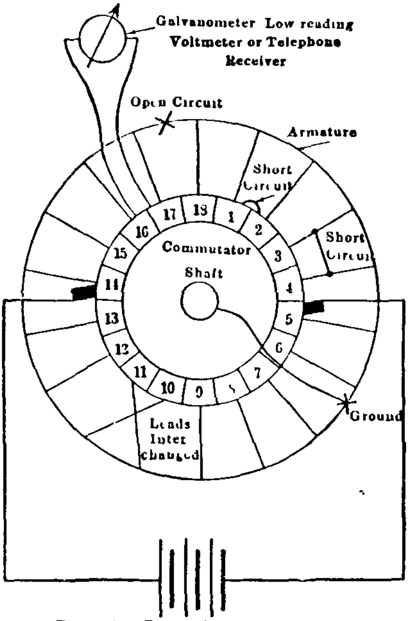


FIG 56 —Bar to bar armature test

Bar to Bar Armature Test. Method of locating open circuits, short circuits and grounds in a closed coil armature (Fig 56) In the following discussion, "meter" refers to the instrument with which the readings are taken, this may be a galvanometer, a low reading voltmeter, or a telephone receiver If a telephone receiver is used, "deflection" refers to the click of the telephone receiver when the circuit is made or broken through it Two or three cells of battery will be sufficient for ordinary work The battery is first connected across commutator bars corresponding to operating brush positions If the deflections are the same when the meter is connected to each pair of adjacent commutator bars, then there are no open circuits, short circuits nor leads interchanged.

Open Circuit The battery is connected to two points on the commutator and the meter is connected across each pair of adjacent commutator bars in turn Zero deflection indicates an open circuit When there is but one open circuit on one side between the battery contact points there will be a great deflection when the meter is connected across the adjacent commutator bars (as 17 and 18) which lie on either side of the open circuit When an open circuit is located it should be closed temporarily by a drop of solder between the two adjacent commutator bars between which the break occurs If other open circuits, exist they may be located by shifting the battery contacts to other points and repeating the above process

Short Circuit When the meter is connected between two adjacent bars such as 1–2 and 3–4, between which there is a short circuit, there will be a very slight deflection

Leads Interchanged If as between 10 and 11 the deflection is reversed, the armature leads connected to bars 10 and 11 are interchanged from their correct position

Ground The battery is connected to two points of the commutator, approximately opposite each other, one meter terminal is connected to the armature shaft, and the other is drawn completely around the commutator If there is only one ground there will be found two balancing points, or points giving the least deflection Next, the two battery contacts are to be shifted a few bars one way or the other When this is done one balancing point will be found at the same bar as before, thus showing the approximate location of the ground The other balancing point will move with each change of battery connection if but one ground exists.

Insulation Test for Field Coils. For roads using trolley

New fields	2500 volts a c 5 seconds
Old or partially repaired	1000 volts a c 5 seconds

For roads using third rail where voltage fluctuations have to be taken into consideration

New fields	3000 volts a c, 5 seconds
Old or partially repaired	1500 volts a c 5 seconds

Field Coil Test by Full Load Current. Full load current is applied to the field coil Temperature and voltage drop across the coil are observed and compared with those for a field coil known to be good

Field Coil Test by Transformer. During this test the field coil becomes the secondary winding of a transformer (Fig 57) The field coil is placed on the laminated iron core and the yoke placed to complete the magnetic circuit Alternating current is then applied to the primary The primary voltage applied and the ratio of the number of turns in the coil under test to the number of turns in the primary coil must not be such that a voltage dangerous to the operator or insulation will be induced If there is no short circuit in the field coil the current taken by the primary at any applied voltage will be the same as without the field coil in place but with the yoke in place A short circuit in the field coil is evidenced by a considerable current in the primary, and if the applied voltage be of sufficient value the field coil will become appreciably heated by the induced current flowing within itself. The terminals of a field coil found to contain no short circuit are then connected together and the test for short circuit is repeated on this field coil. If there is no evidence of short circuit when this is done then there is an open circuit within the coil. If, however, the test indicates a short circuit, this field coil contains no open circuit. Fig. 58 from the *Electric Rail-*

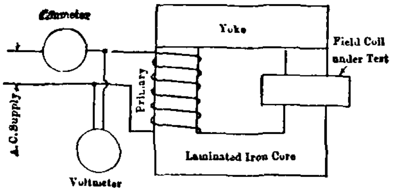


FIG. 57.—Transformer test for field coils.

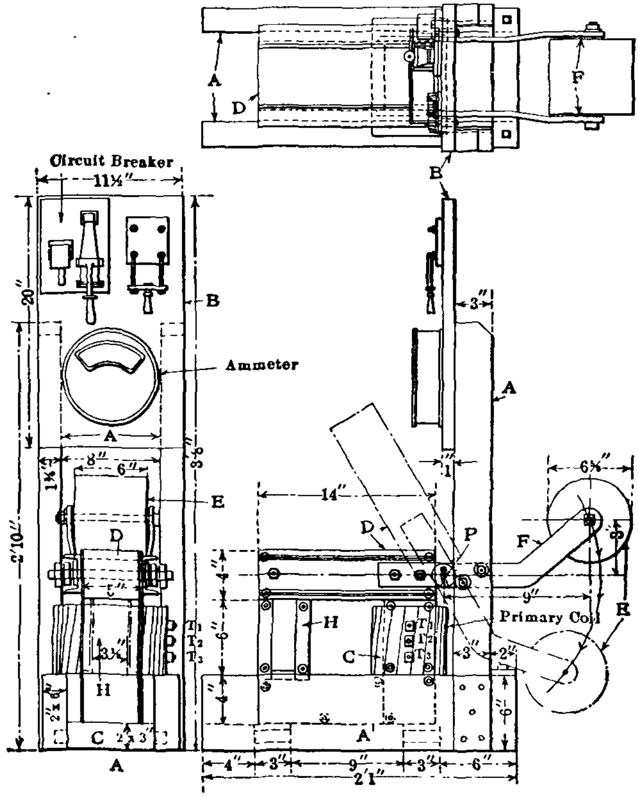


FIG. 58.—Arrangement for testing field coils—Brooklyn.

way Journal shows the construction of a field-testing transformer used in the shops of the Brooklyn Rapid Transit Company. The framework *A* is made of wood. *B* is a slate panel 20 × 11½ × 1 in., on which the circuit-breaker, ammeter and switch are mounted. *C*, *H* and *D*, which comprise the magnetic circuit, are made of several thicknesses of 1/32-in. wrought iron dipped in shellac. After the shellac was dried, the laminations were placed between the two ¼-in. side pieces and riveted together. The primary coil is made up of 81 turns of No. 5 B. & S. double cotton-covered wire with 27 turns between T-1 and T-2 and 81 turns between T-1 and T-3. *E* is a cast-iron counter-weight 6⅝ in. in diameter and 6 in. wide which assists in raising *D* and also in keeping it in the upper position while a field coil is being placed in position for testing or while one is being removed. The arms, *F*, made of ½ × 2 in. flat iron, bolted to *D*, have the counterweight attached at the other end and are free to turn on a pin. *D* and counterweight *E* thus revolve on the same pin, which is supported by two brackets, one on each of the upright members of the wooden frame.

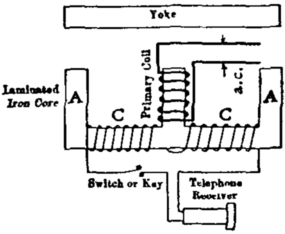


FIG. 59.—Transformer for testing field coils.

Fig. 59 shows another arrangement for making the transformer test of field coils. *C C* are two similar coils, so connected that the voltage induced in one shall oppose that induced in the other. The field coil to be tested is placed on one of the legs *A* (the yoke not being used in this case) and an alternating current is applied to the primary coil. If the field coil under test contains no short circuit, the sound in the telephone receiver will be very faint. If, however, there is a short circuit in the field coil, this will unbalance the fluxes through *C C* and the sound in the telephone receiver will be loud. The terminals of a field coil found to contain no short circuit are then connected together and the test for short circuit is

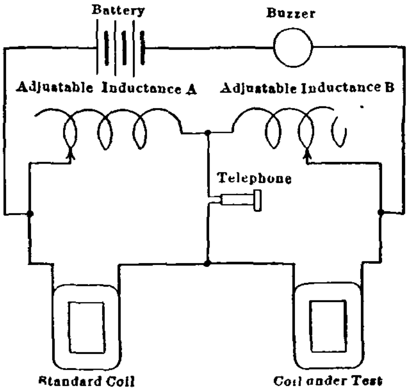


FIG. 60.—Inductance balance for testing field coils.

repeated. A loud sound in the telephone receiver now indicates that there is no open circuit in the field coil. A faint sound indicates that there is an open circuit in the field coil. If it is desired, this apparatus, together wih the yoke but without the telephone receiver, may be used to test field coils in the manner outlined in the preceding paragraph.

Field Coil Test by Inductance Balance (Fig. 60). Instead of the battery and buzzer indicated, a low-voltage alternating current may be used. A standard coil known to be faultless and similar to what the coil under test should be if perfect, also the coil to be tested are connected as shown. The adjustable inductances *A* and *B* are then adjusted to give a balance which is indicated by the faintest sound in the telephone receiver. If *A* and *B* are then practically alike, the coil under test is without short circuit or open circuit. If. however, *B* is less than *A* there is a short circuit in the coil under test.

Insulating Materials for Railway Shop Use

(A.E.R.E.A. Miscellaneous Methods and Practices)

Definitions. *Air Drying Varnishes.* Linseed oil or other drying oils, with such driers and treatment as to dry quickly without necessity for baking.

Baking Varnishes. Linseed oil or other drying oils, with gums and driers. Requires baking.

Spirit Varnishes. Animal or vegetable gums, shellac, copal, etc., dissolved in alcohol, amyl acetate, benzine, etc.

Insulating Paints. Solutions of asphalt, and mineral gums in carbon bisulphide, naphtha, etc.

Impregnating Compounds. Petroleum asphalt, melting point 105 to 115 deg. C., used for impregnating motor fields, etc. Paraffine and linseed oil are used as impregnating material for special purposes as described below.

Air drying varnish should be composed of linseed oil or other drying oils sufficiently treated and with proper amount of dryers so that varnish will dry in 2 or 3 hours, leaving a smooth, flexible surface. Baking not required. Should be both oil and moisture proof. Should not crack, soften or become tacky at 100 deg. C.

Baking Varnish. Composed of high-grade linseed oil, with some gums and dryers. Baking varnish should not be used except baked at 100 to 110 deg. C. for 10 or 12 hours. Characteristics: high melting point, flexibility, smooth, glossy surface, good filler, high insulating qualities, oil and moisture proof.

Spirit varnishes are composed of animal and vegetable gums, such as shellac, copal, etc., dissolved in spirits. Characteristics: oil proof but not moisture proof; good stickers and binders, but not necessarily good insulators. In general they soften at 100 deg. C.

Note: On string band or webbing covering mica at the outer end of the commutator it is recommended that no varnish be used that will soften at a temperature of 100 deg. C.

Insulating Paints. These are solutions of asphalt or mineral gum in naphtha or other solvents. Characteristics are moisture proof but not oil proof, lacking in toughness, good as a sticker and paint, but not high in insulating qualities.

Impregnating Compounds. *Asphalt Compounds.* Composed of asphalt. Characteristics: Different asphalts have widely different melting points. The asphalt to be used is one in which the melting point is above the operating temperature, preferably 110 deg. C. At ordinary temperature should be somewhat pliable, but should not flow. At melting point should flow freely. Is moisture proof, but not oil proof. Good as a filler, and has fair insulating qualities. Asphalt should be practically free from volatile matter. In order to get a good penetrating asphalt the committee recommends that the asphalt shall not contain more than 5 per cent of constituents insoluble in carbon bisulphide (CS_2).

Paraffine is a mineral gum of a low melting point, suitable for impregnating wood for outside or low temperature work. Is not suitable for use in railway apparatus where working temperature is over 50 to 60 deg. C.

Linseed oil is excellent for filling wood parts of railway apparatus. The parts should be boiled in raw linseed oil until filled and then baked to oxidize same, the wood to be finished with either an air drying or baking varnish. Wood treated in this manner is excellent for outdoor work, exposed to climatic conditions. Linseed oil hardens under temperature, does not melt, is moisture and oil proof and is flexible and strong.

Tests for Insulating Materials. The following simple tests are recommended for the various substances: Without an elaborate testing laboratory, it will be impossible to find the exact values of tests on the various insulating compounds, but these tests can be made with little or no apparatus and serve as very good general tests of the material.

Asphalt. The chief consideration in connection with asphalt is the melting point. Asphalt can be obtained from a low to a very high melting point. Some asphalts contain as high as 30 per cent of inorganic matter. In order to determine how much inorganic matter, a certain definite amount of the asphalt can be melted and burned away in an ordinary crucible or gas flame. The presence of sulphur and the degree may be determined by melting the asphalt and allowing a bright sheet of copper to remain in the melted asphalt for 2 or 3 days. The tarnishing of the bright copper will indicate the presence and quantity of sulphur.

Linseed Oil. Linseed oil may be heated up to the boiling point, and if there be any foaming and spluttering as it begins to boil, it indicates presence of water in the oil. Linseed oil may have a sediment usually known as "foots." The relative amount may be approximately determined by allowing a sample of the oil to stand in a test tube for about 24 hours and noting the amount of sediment deposited. Regular boiled oil has a specific gravity of 0.945 or higher, whereas the ordinary boiled oil which has been produced by pouring into the bunghole a certain amount of drier, has a specific gravity but little higher than raw linseed oil, this specific gravity being 0.931. Acids are used during the process of refinement. In order to test the amount of acids, it would be well to immerse a bright copper strip in the oil for several days.

Japans and Varnishes. These should be tested at a temperature which is ordinarily used in service. After the coating has thoroughly dried in the air or by baking, according to which treatment the material is designed, the flexibility of the coat may be determined. It would be well to test with a bright copper strip for acids.

Shellac. Shellac contains more or less rosin. The determination of this requires elaborate analysis. The quality of the shellac, however, may be determined by painting on bright tin, and by making comparisons with other shellacs, the flexibility may be determined. A very flexible shellac has little or no rosin. The presence of rosin makes it brittle. It would be well to test with a bright copper strip for acids.

Paraffine. The chief difficulty in paraffine is water and sulphuric acid from the refining processes. The amount of water can be determined by heating; foaming and sputtering on reaching the boiling point indicates the presence of water. Sulphuric acid can be detected by the immersion of a bright copper strip, or by mixing some of the paraffine with boiling water and then pouring off the water and testing with litmus paper.

Impregnation of Motor Insulation. Dipping in varnish or insulating compound and then baking thoroughly fills all cracks and pores in the insulation. This greatly reduces the possibility of breakdowns, which might be caused by moisture or other conducting materials filling these cracks. Further, such treatment acts as an effective bond to prevent vibration of motor parts. Dipping and baking, even of new motors, is an insurance against maintenance charges for rewinding. It improves the insulation, fills up the pores, keeps a smooth surface on the coils, and prevents vibration of motor parts. The equipment consists of a tank to contain the dipping solution, an oven in which to bake, and means of handling the apparatus. Heat the apparatus in an oven at 95 to 105 deg. C., for 12 to 24 hours, depending on the size of the armature, to insure that all moisture is driven off and to warm the armature for dipping. Dip in an oil proof and moisture proof baking and insulating varnish at approximately 0.840 specific gravity. This depends, however, upon the varnish and upon the thickness of coat desired. If the varnish is too heavy, thin it with benzine. Dip armatures in the varnish in a vertical position, pinion end down, and to such a depth that the varnish will come to the commutator neck. Allow to soak for approximately five minutes. If a tank is not available, results (though not so good) may be obtained by turning the armature in a shallow pan with the varnish deep enough so that the bottom of the slots will be completely immersed. If this is done, the insulated creepage surface at the end of the commutator should be treated by repeated paintings of the varnish. Turn until all the coils have been thoroughly soaked. Field coils for direct current machines may be removed and dipped and baked, but if desired, the frame may be dipped with coils in place. Drain at room temperature until all dripping ceases. The apparatus should be placed in such a position that pocketing will not occur. It may be necessary to turn during draining. Place apparatus in oven in a position to avoid pocketing of varnish, and bake at 95 to 105 deg. C., for the following time: armatures below 12 in. diameter, 48 hours; 12 to 30 in. diameter, 60 hours; over 30 in. diameter, 72 hours. If an armature is baked in a horizontal position, it should be turned at intervals during the first half of the baking period, otherwise the varnish will drain toward the lower side and throw the armature out of balance.

Handling of Copper. Copper cannot be given the same rough treatment that other metals will stand, but requires some important precautions in its handling and application. It fails quickly under localized stresses. Sharp bends, rough or nicked edges in copper straps or wires, limited movement to take care of expansion and contraction, all are points especially to be guarded against. This is particularly true when the copper is subject to quick sharp blows or vibration, such as are common with railway motors. All bends in copper should be made with as large a curvature as is possible in the space available. The effects of vibration, expansion, contraction, and centrifugal force show up first in sharp bends and fillets. Sharp bends should not be made at the ends of copper strap field coils or when making clamped or soldered joints, as this may later be the cause of a motor failure. It often happens that armature coil failure is at a point where wires have been carelessly bent or crossed. The nicking of copper should also be avoided. It is very easy to nick copper with the sharp edge of a metal drift or other tools as are used in connection with the winding of armatures. It is preferable to use a hard fiber drift and drive leads down into commutator neck slot by using a copper filling piece placed on the lead to receive the blow from the hammer. Another source of trouble due to nicking of copper is found in field coil cable leads breaking at the point where the insulation has been cut off with a knife, the break having been started by the knife nicking the strands of the cable. Extreme care should be used in removing insulation on all cables and wires of small cross-sections. For the same temperature change, copper will expand more than iron or steel. Further, it is usually found that, in a motor, the copper becomes hotter than the other materials of which the motor is made. Therefore, it is necessary to provide means for the copper to expand and contract to take care of the relative motion between the different materials for the changes in temperature. A common error in this respect is to anchor the wiring around frame connections to the frame proper, when it should have been securely bound to the windings so that it would be free to move with the windings. It is obvious that this is more important with solid strap conductors than with flexible cables. Properly supported copper stands up well against vibration. In fact, one finds it extensively used in certain applications where vibration occurs, because of its good behavior in this respect. But improperly supported, copper fails easily under vibration. This point is frequently overlooked in connecting car wiring cables to the motor leads when cleats are not applied. Care should be observed to properly locate the cleats in supporting the cables. It often happens that the weight of a solid connector, even though it may seem rather small, is sufficient under vibration to cause the copper strap or stranded cable to break at a point just behind the connector where stresses are localized. The bad effects of tinning stranded copper beyond the point where the joint is made must be emphasized. The solder should stop just inside the connector, so that the stresses will not localize where the strands are not supported against vibration.

Copper is subject to a form of "sickness" which so far as has been experienced is peculiar to copper alone. This subject has been thoroughly discussed by N. B. Pilling (*Electric Journal*, 1920). All commercial copper contains a small amount of oxygen in the form of copper oxide, without which it has poor mechanical characteristics. When it is heated in a flame which is rich in free hydrogen, the hydrogen unites with oxygen forming free copper and steam. It is a peculiar characteristic that hydrogen will readily enter hot copper, but the steam cannot get out. The copper is thus not only weakened by the elimination of the copper oxide, but the high-pressure steam expands, producing a spongy effect which still further weakens the copper. This effect is, of course, greatest near the surface. This peculiar form of sickness should be guarded against by maintenance men. An experience due to this change in structure will serve to bring out this lesson. An armature was being wound with coils having german silver resistance leads, with copper tips brazed onto their ends. A number of coils had been put in place when it was found that the first one had to be removed. In doing this, the copper tips were bent back to get them out of the way. With only a single bend, one of the tips broke off in the workman's hands. On examining all the tips of the coils, twelve more defective ones were found. Tests showed that the copper from which the tips were made was of good quality. A study of the process of handling the copper revealed that the defective tips had been heated in a flame containing unburned hydrogen. Since one cannot see, without breaking the strap or wire, whether the copper has been affected by this sickness, it follows that to be safe, copper should not be heated in a flame containing an excess of hydrogen. This means that with a blow torch the copper should be kept outside of the inner cone of blue flame. When heating copper in a gas and air furnace, an excess amount of air should always be used, as too little air will produce an excess of free hydrogen. Smoke from such a furnace always indicates too little air, and the mixer should be adjusted to give a little more air than is necessary to prevent any trace of smoke. Wherever possible, the copper should be heated without coming into direct contact with the flame. It has been common practice to burn the insulation from old coils. This should not be done where the coils are to be reinsulated and used again. The question then arises, how to remove the old insulation. One operator places the coils in an oven and passes steam through for 12 to 14 hours. He finds that the insulation peels off easily while hot. Another operator dips the coils in a weak solution of muriatic acid for a time (approximately 24 hours) so that the acid weakens the insulation, but not long enough to give the acid a chance to eat into the copper. The necessary time required can be established by checking carefully and removing the coil when the brightening of copper commences. After the acid treatment, the coils should be thoroughly washed in clear water.

Connections between Car Wiring and Motors. A scheme of connections and cable suspension which has been used with considerable success by a number of operating companies is shown in Fig. 61, which illustrates the method employed for inside hung motors; for outside hung motors, the general arrangement is the same. The knuckle joint connectors are held firmly in place between two cleats fastened to the car underframing. The cleats holding the connectors should be placed as near as possible to the bolster and the truck pin on the car underframing, in order to keep the cable loop, as it swings between the car underframing and the motor, as close to its original length as possible in any position of the truck. About three inches should be allowed on each side of the connector between the cleats to take care of creepage. The cleats are split so that they can be put in place after the connections have been made and insulating hose or tube has been slipped over the connectors. The cables are clamped to the housing with a pair of stationary cleats as they come out of the motor frame. To prevent chafing of the cables as they drag over the motor housing, a pair of loose cleats are clamped on to the cables at a point which will allow free movement but also such that the cables will not touch the motor housing at any point. To give further protection to the knuckle joint connectors against the weather, a box may house the cables at the point where the leads from the motors are connected to the leads of the car body.

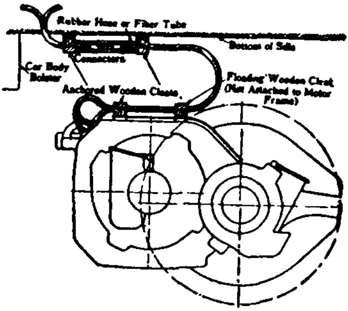


FIG. 61.—Method of cable suspension.

To simplify the changing of motors, the cables at the cleats on the car underframing should be arranged in the same sequence for all motors, and the cables as they leave the cleats on the motors should always be in the same sequence and should line up so that when the motor is placed in the truck, the proper lead at the cleat on a car underframing will be directly over the lead to which it is connected on the motor.

Broken Motor Leads. Assume that No. 3 motor has an open circuit, due to either field or armature leads being broken or burned off, and that the car is continued in service. With the control in any of the series positions, No. 1 motor receives double the current taken by the No. 2 or No. 4 motors, which means that No. 1 motor has an increased torque and tends to take too much of the load. With the control in any of the parallel positions, the load is equally divided between the three motors, 1, 2 and 4. Depending upon service conditions, such as headway, schedule, location of turnouts on single tracks, congestion of traffic, etc., the amount of series running will vary, and this will determine how much excess current No. 1 motor will receive. In many cases cars are continued in service with the motors operating in this unbalanced condition, which is discovered only when they are given their regular light inspections, which are usually from 7 to 15 days apart. Under these conditions, No. 1 motor has had a number of short time overloads, possibly resulting in overheating of the winding. Such overheating may not cause the motor to fail at the time the damage is done, but the insulation may have been charred and its life shortened. Some time later the windings on this motor may either ground or short-circuit, due to the poor condition of the insulation, and when examined the motor will be reported as having a roasted armature. It is quite difficult to locate the direct cause of the roasting, as the conditions responsible for this trouble may have happened weeks before, and may have been remedied without thought of the possible damage done to the windings of the running mate of the motor with a broken or burned off cable. In a service test duplicating the above conditions, the brushes were taken out of No. 3 motor. Run thus, the current in No. 1 motor was 80 per cent greater, and that in Nos. 2 and 4 motors 10 per cent less than when all four motors were operating normally in the full series position of the controller. In full parallel, each of the three working motors took 20 per cent more current than when four motors were operating normally. In this test, the (experienced) motorman was unable to tell by the operation of the car whether or not one motor was out. The results of this test indicate that one motor could be open-circuited, due to a broken or burned off cable lead, and the car could be operated with the motors in this unbalanced condition without the knowledge of the motorman; this would result in overheating the windings of the running mate of the motor with the open circuit. With the older and heavier type of motor on the car, there was a sufficient mass of metal available to absorb the heat resulting from short time overloading of the motor, due to unbalanced conditions resulting from a broken or burned off motor lead, and consequently armatures were not so apt to be roasted. On the other hand, the light weight, ventilated type of motor, weighing much less than the old, heavy type motor, and operating under the same service conditions, is unable to absorb so much additional heat due to short time overload, and its temperature rises rapidly, more often resulting in roasted armature windings. The always important precautions against broken motor leads, such as proper cleating, reinforced insulation, etc., therefore become of greatly increased importance with the light weight ventilated type of motor. It may even be advisable to provide for a daily inspection of motor leads, although a hand touch system of motor temperature observation as cars come in from runs should indicate a motor which has not been working.

Railway Motor Gears and Pinions

Pitch. The diametral pitch (number of teeth per inch of diameter of pitch circle) commonly used is 2.5; 3 is also used, but for motors of not greater than 75 rated h p Larger teeth are used on locomotives. Spur gearing has been of the standard 14½ deg. involute form.

Taper of Pinion Bore. A taper for bore of pinion in the proportion of 1.25 in in diameter to 1 ft in length was adopted as standard by the Am El Ry Eng Assn., 1911

Split Gears. The halves of eight-bolt gears are fastened together by four bolts on each side of the hub, two of each four being placed beside each other near the hub, and the other two as near the rim as possible. There are two methods of bolting the halves of four-bolt gears together. One method is to place one bolt on each side of the hub and the other two out near the rim Another method is to place two bolts on each side of the hub, side by side between hub and rim. A complaint in connection with split gears is that the bolts are often stretched beyond their elastic limit.

Solid Gears. Generally speaking, solid gears can be made stronger and lighter than split gears, moreover, they require no key seating in the axle

Diameter Allowance and Pressure Required to Fit Solid Gears. An allowance of 0 001 in. for every inch of axle diameter should require from 40 to 60 tons to force the gear in place, and such a fit is ample to prevent the gear from slipping

Tooth Pressure. Experience indicates a pressure of 1000 lb. per inch width of gear face as a perfectly practicable value for continuous rating of large gears, and that with special steel pinions and high grade gears it is probably safe to exceed this figure. In the St. Clair Tunnel locomotives the pressure is carried on a single gear having a 6-in. width of face on the gears. The life of the pinions is 40,000 to 50,000 miles. With the normal loads at which the locomotive operates on up grade the pressure is from 1500 to 2000 lb. per inch width of face on the gears.

Factors Essential to Durability of Gears and Pinions. The most important factors conducive to long life of gears are: (1) Proper steel properly treated; (2) proper alinement; (3) proper lubrication; (4) tight gear cases.

Specifications for Gears and Pinions. The Am. El. Ry Eng. Assn has adopted standard specifications for case-hardened and for quenched and tempered forged steel gears and pinions, which contain the following provisions Blanks to be made from open-hearth steel. Chemical composition of case-hardened gears and pinions—

Carbon	0 20 per cent—not less than 0 12 per cent nor more than 0.28 per cent
Manganese.. .	0.50 per cent—not less than 0 40 per cent nor more than 0 60 per cent
Phosphorus .	not over 0 05 per cent
Sulphur ..	not over 0 05 per cent

(Requirements for heat-treated gears and pinions include the last two, relative to phosphorus and sulphur, only) The minimum thickness of the rim of gears, measured ⅛ in from edge of rim, shall be ⅜ in for 3 pitch, 7/16 in for 2½ pitch, and ½ in for 2 pitch.

Approximate Life of Gears and Pinions. The following information is from a paper by T. W. Williams at a meeting of the

Street Railway Association of the State of New York. Data from gears and pinions (2.5 pitch) on 100-h.p. motors on two cars identical as to weight, motor capacity, running schedule and route. The life was estimated by subtracting the amount of wear on the pitch circle of the teeth after running 50,000 miles from the original thickness and then estimating how many more car miles the teeth could run before 0.1 in. had been worn from each face of the teeth, on the basis that the wear would be constant throughout their life. The pitch of all the gears was 2.5.

Description	Life of pinion, miles	Life of gear, miles
Untreated high carbon pinion running with cast steel gear.	70,000	220,000
Heat-treated high carbon pinion running with cast steel gear.	100,000	240,000
Heat-treated high carbon pinion running with heat-treated high carbon gear.	250,000	750,000
Case-hardened pinion running with case-hardened gear.	200,000	650,000

Some of the modern types of gears and pinions have been showing a life very considerably in excess of that indicated in the above table.

Material of Meshing Gear and Pinion Should Be the Same. If soft steel gears be run against hard steel pinions, or *vice versa*, the softer surface becomes charged with the gritty material which finds its way into the gear case, and when so charged this surface grinds away the harder surface, but when hardened gears and pinions are run together this trouble does not seem to be so great.

Gear Teeth of Special Design. *Long and Short Addendum Type.* The "addendum" of a gear tooth is its length above the pitch line; the "dedendum" is the depth below the pitch line to which the top of the meshing tooth enters. In the standard type of involute gearing, the addendum and dedendum are equal. In the "long and short addendum" type of gearing, the armature pinion has a long addendum and short dedendum, while the axle gear teeth are of the corresponding opposite design—short addendum and long dedendum. This accomplishes a shorter pushing action as the teeth go into mesh, and a longer pulling action out of mesh, with a consequent less tendency to vibration. It also results in a wider base and greater strength of the pinion teeth, as well as more metal between the roots of the teeth and the bore of the pinion. Where only a portion of the gear and pinion equipment is of this type, great care should be taken to avoid mixing with the standard type, as, obviously, the two types will not mesh together properly.

Helical Gears. The helical gear overcomes inherent gear vibration by eliminating the so-called "stepping over" action. The operation of this type of gear can best be realized by thinking of it as made up of a number of thin spur gears twisted on the shaft with respect to each other. Helical gears transmit practically average motor effort, with properly maintained bearings. The gears tend to wear evenly over the full tooth length, thus preserving the original tooth form. The contact from the tip of the tooth to the root and across the face is the property inherent in helical gearing which produces the smoothness of gear action. The action of engagement from one side of the tooth to the other is practically one of pure rolling. The action from the tip of the tooth to the root is one of sliding and rolling. The percentage of rolling action can be predetermined, within reasonable limits, and the teeth can be designed to give a maximum percentage of rolling. As a result of this rolling contact across the face, there will be at any time (after the tooth has come into full mesh) tip, pitch line and base contact. In the case of the spur gear, it is in contact across the face first at the tip and then progressively from the pitch line to the base. From the design of the involute form of tooth, there is pure rolling at the pitch line, while at the tip and the base there are sliding and rolling. The tendency is thus to greater wear at tip and base. As the gear wears it is continually destroying the original tooth form. The characteristics of helical gears for railway motors, as produced by the Westinghouse company, are: 4¼ to 2 diametral pitch; 3½ to 5 in. face; 7½ deg. helix angle; 20 to 22½ deg. involute, with long addendum on pinion and short addendum gear teeth. The advantages claimed are: reduction of gear vibration; increase in life of gears; reduction in gear noise; together with the advantages of the long and short addendum as before listed. The end thrust due to angularity of teeth has, according to some users, been productive of increased wear on bearing flanges and thrust collars; others, however, find the end thrust only sufficient to reduce the lateral movement of the armature, provide a cushioning effect to such lateral movement, and maintain sufficient end bearing to insure an oil film.

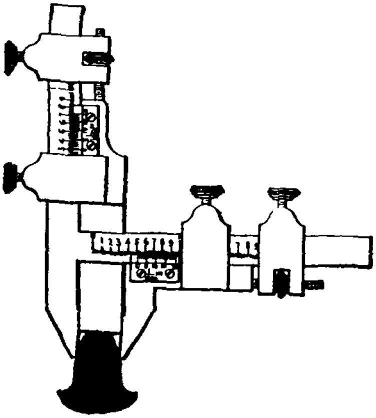


Fig. 62.—Gear tooth vernier.

Measurement of Gear Tooth Wear. A gear tooth vernier, which is most convenient for the measurement of gear tooth wear, is shown by Fig. 62, which illustrates its method of use. One vernier enables the measurement to be made at the proper distance from the top of the tooth (usually on the pitch circle), while the other is used for the measurement. Both verniers are scaled to read to 0.001 in.

Fitting Railway Motor Pinions. Experience has shown that in order to obtain satisfactory operation, pinions should drive their gears through the "press fit" or "shrink fit" on the shaft and not through the key. The key acts merely as a safety device should the pinion accidentally loosen. The desired fit for the pinion can be had by heating or by pressing. The following points should be observed when putting pinions on railway motor shafts with taper fit: The shaft should be clean and free from burrs or swellings; the pinion bore should be clean and free from burrs; the fit of the pinion bore should be in contact with at least three-quarters of the surface of the taper fit on the shaft. This can be checked by rubbing Prussian blue, thin red lead and oil, or thin lamp black and oil on the pinion bore and fitting it on the shaft. The pinion then should be put on the shaft cold to make sure that the keyway in the pinion is the proper size for the key mounted on the shaft, and that the pinion does not ride or bind on the top or sides of the key and will not ride the key when pressed further on. The keyway on the pinion can be 0.002 in. wider, but not less than the key. There should be at least 1/64 in. top key clearance in the pinion. The corners of the key should be rounded, so as not to cut into the fillet of the keyway. Pinions may be pressed cold onto the shaft with a wheel press. The pressure required will be 12 to 25 tons for pinions on motors up to 125 horsepower and 40 to 80 tons for pinions on motors of 125 horsepower or over. Pinions with bores up to three inches that are pressed on cold should advance on the shaft approximately 1/32 in.; those with three- to four-inch bore, 3/64 in.; and those with four- to five and one-half-inch bore, 1/16 in. This distance is measured from the point where the pinion is seated firmly on the shaft before pressing.

To fit by heating, pinions for motors up to 125 horsepower and up to three-inch bore should be heated in boiling water for thirty minutes, and those with three-inch or larger bore for sixty minutes. When the pinion has attained the temperature of the boiling water, it should be taken out of the water and the bore quickly wiped clean. Without allowing the pinion time to cool, it should be tapped on the shaft with a six or eight pound sledge hammer, using a heavy piece of wood or copper between the pinion and the hammer. This sledging is not to get a driving fit, but to make sure that the pinion is well seated. Three or four taps evenly distributed around the pinion end should be enough. The pinion nut with lock washer can then be screwed home tight with a wrench having a lever arm of three to four feet.

For motors of over 125 horsepower, the pinions should be heated with a gas flame, applied in the bore of the pinion in such a manner as not to touch the teeth of the pinion, as this might affect the temper. The flame should be so regulated as to take 45 to 75 minutes to bring the pinion to a temperature of 125 to 150 deg. C. The temperature can be measured by placing the bulb of a thermometer against the pinion between the teeth. The surface of the pinion where the bulb touches it must be made perfectly clean by rubbing with emery cloth. It is also important to protect the exposed part of the thermometer by covering it with asbestos cloth so the flame cannot touch the thermometer. When the pinion has reached the correct temperature, the bore should be wiped clean and the pinion put on the shaft in the same manner as suggested for motors up to 125 horsepower.

Any furnace in which the pinion is so located that the flame cannot touch the teeth can be used for heating pinions. The flame can be regulated and the pinions kept at a temperature of 100 deg. C. for pinions on motors up to 125 horsepower, or 125 deg. C. for 125 horsepower motors or larger, until the mechanic is ready to apply them.

Inspection of Gears. The inspection of gears and pinions on some predetermined basis is essential to their length of life, whether they are inspected on a time or mileage schedule, as the service or conditions they are operating under would have to be considered when deciding the rules to be followed. Cars operated with small motors in city service are generally inspected on a thousand mile schedule, or on a time basis of from seven to thirty days. This inspection, however, is only a casual one, as the gear case is seldom removed, and the small opening in the top half of the gear case is used to make this inspection as well as to apply the lubricating material. The alinement of gear and pinion is of first importance, as well as proper meshing, for if they are permitted to run out of true centers for any length of time the possibility of their being properly interchangeable is destroyed, on account of the uneven wear of the teeth. Two important points which affect the alinement of the gear and pinion seriously are the axle bearings and axle collar. If the axle bearing is worn on collar end, or the axle collar has not been properly adjusted, the motor will have a lateral motion which will carry the pinion out of alinement with the gear, and carry the gear case over against the gear, which in a short time will destroy the gear case. Again, if the axle bearings are worn in their inside diameter, they permit the gear to drop away from the pinion, thereby destroying the pitch line of the gear and pinion. The gear case, which has been planned for the protection of the gear and pinion, as well as to hold the lubricating grease, is designed to fit the allotted space in the best possible manner, and while it is not all that could be desired, it is essential that it be maintained in a manner that will protect the gear and pinion, and especially their lubricating material, from being contaminated with dirt and water.

Lubrication of Gears is regulated according to the service in which the equipment is being operated. Small quantities of grease applied frequently will give better results than large quantities at longer periods, both as to the cost of grease and life of gear and pinion. The opinion of the master mechanics on different electric railways as to the kind of grease best adapted for the purpose is diversified, some preferring a sticky grease because of its adhering to the gear and pinion teeth, while others, realizing the amount of energy lost in overcoming the resistance of a sticky grease, prefer a grease of better lubricating qualities. The idea of a sticky grease being a preserver of gears and pinions probably is a false one. The oily grease not only shows a longer wearing quality, but the temperature increase of the gear and pinion is nil as compared with temperatures obtained with sticky grease under the same conditions. The excessive wear on gears and pinions occurs during the winter months, and an investigation as to why this should be shows that sticky greases become so hard in zero weather that

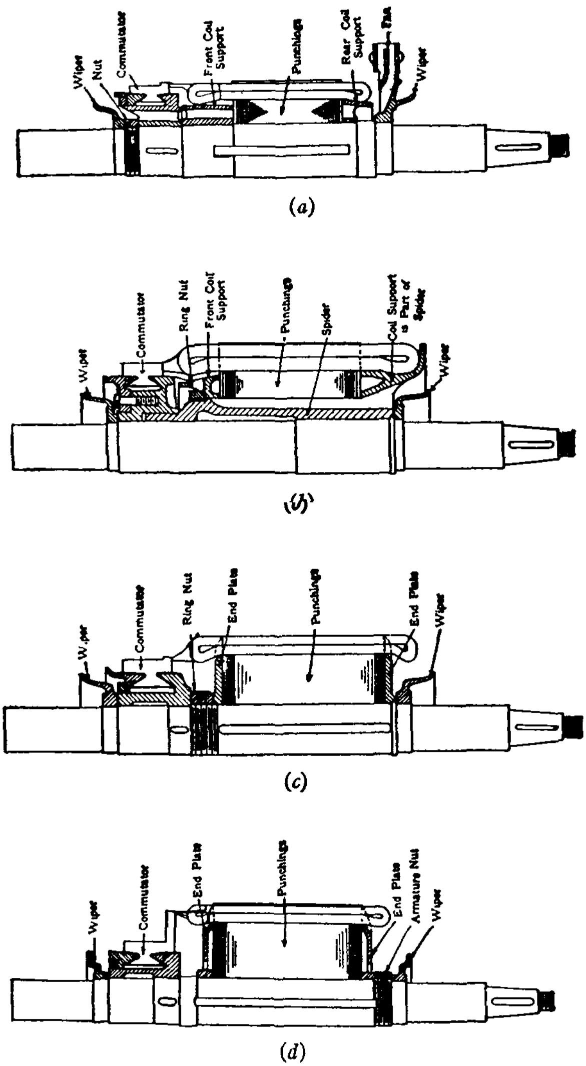


FIG. 63.—Mechanical assembly of armature.

the lubricant flakes off and leaves the gear teeth bare until such time as the temperature of the gear and pinion increases to a point where the grease becomes more pliable. If there is not enough lubricant left on the gear to cover the teeth, they continue in service without lubrication until inspection time, when another application of lubricant is made, and if the weather continues cold, the same condition prevails as before and the gears and pinions suffer on account of it. The conclusions are that an oil grease should be used, that it should be applied in small quantities at short intervals, that maintenance of the gear case is of first importance, and that proper attention must be given to motor alinement.

Renewal of Armature Shafts. There are four general methods of mounting the commutator and core on a railway motor armature shaft. On the more modern motors, especially of the smaller sizes, the core is built directly on the shaft and held in place by a nut at the commutator end; this type is used mostly on the ventilated type motors having longitudinal ventilating ducts through the core and a fan at the rear end. Where space is available, the spider

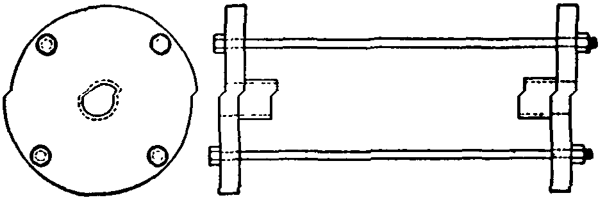


FIG. 64.—Clamp for armature core for shaft renewal.

type of construction is also used. The other two types shown in Fig. 63 have been superseded, as it is necessary to disconnect the armature leads from the commutator when a shaft is to be replaced. Some of the older type shafts have wiper rings screwed on the shaft, but this practice has been changed to a shrunk-on type of wiper on the more modern motors.

At (*a*) in Fig. 63 is shown the construction in which the armature core and commutator bushing are mounted directly on the shaft and depend upon the press fit, the key and the lock nut at the commutator end to keep them in place. The fan and commutator end wiper ring which have been shrunk on, are removed by heating with blow torches (keeping the shaft cool by wrapping it with wet asbestos), and are driven off with a hammer and chisel. After the fan and wiper are removed, clamp the core and commutator together by placing over the shaft two pieces of iron pipe, one on each end, large enough to clear the armature nut and long enough so that plates placed over the shaft and against the pipes will clear the end of the commutator and the end of the coils at the pinion end, as shown in Fig. 64. Bolt the plates together, using four or more bolts just clearing the outside of the armature. Take special care to cut the pipes off square and to pull the bolts up evenly to prevent warping the core when the shaft is removed. Allow clearance enough at the pinion end between the plate and the shaft so that the key will clear. Another form of clamp has rings machined to fit the commutator and rear coil support, with bolts through the longitudinal vent ducts. When the clamp is in place, back off the lock nut at the commutator end, press out the shaft by applying pressure at the commutator end, and replace with a new one. With the new shaft in the core, apply and tighten the armature nut, remove the clamp and shrink on the fan and wiper ring.

At (*b*) in Fig. 63 is shown the construction where the armature core and commutator bushing are mounted on a spider, which is in turn mounted on the shaft. The commutator end wiper and the fan are shrunk on. To remove the shaft, take off the pinion end wiper ring (or fan, if a fan is used) by heating as described above. With this type it is not necessary to remove the commutator end wiper, as it comes off with the core. Pressure is applied at the commutator end of the shaft in removing. The new shaft is pressed in and after it is in place, the commutator end and pinion end wiper or fan are shrunk on. This method applies both to spiders with separate rear end bells and to spiders with rear end bells cast integral.

At (*c*) in Fig. 63 the commutator spider and core are shown mounted directly on the shaft with a ring nut between the commutator and the core. With this type of construction, it is necessary to lift the armature leads out of the commutator neck, remove the wiper rings and pull the commutator first. The commutator is provided with tapped holes for bolts to aid in this operation. After the commutator is removed, clamp the core together, using a modification of the clamping device shown in Fig. 64, back off the ring nut, and press out the shaft, applying pressure at the commutator end. After the new shaft is in place, apply the ring nut to secure the core, press on the commutator, shrink on the wiper rings and reconnect the windings.

At (*d*) in Fig. 63 is shown a type in which the nut is at the pinion end and there is a shoulder on the shaft between the commutator and the core. In this case also it is necessary to disconnect the windings. Remove the wiper rings and pull the commutator as described above. Clamp the core together, back off the ring nut at the pinion end and press out the shaft, applying pressure at the pinion end. Press in the new shaft and apply the ring nut, which should be drawn up tight before the clamp is removed. Replace the commutator and reconnect, then shrink on the wiper rings.

In connection with the replacing of shafts of railway motor armatures, the following points should be kept in mind: new shafts should have fillets at all changes in diameter; new shafts should be made about 0 004 in. larger than the original shaft at the press fit, to insure the proper tonnage in pressing; check clearance between top of key and key seat in core to prevent binding at this point; chamfer the start of the commutator bore to allow the shaft to enter straight; shafts should be pressed in at approximately 20 to 25 tons on motors from 25 to 50 h p. and about 40 to 50 tons for sizes above 50 h.p.; whenever possible, the armature nut should be removed and replaced while the clamps are in place; shafts will press out at approximately 1½ to 2 times the tonnage used when they are pressed in, because of slight rusting and flowing of metal; the press fit will vary, depending upon the material of the core—steel or malleable iron can safely stand a higher tonnage than cast iron; a little white lead on the shaft at the fit before pressure on acts as a lubricant and prevents rust.

Motor Bearings

Ball Bearings. Among the advantages claimed for the use of ball bearings, as stated by O. R. Wikander, probably the one which has received the greatest amount of attention, is that of the decrease in starting current, due to the smaller starting friction of ball bearings over plain types. With properly constructed ball bearings, the wear at the bearing should be considerably reduced, and the danger of the armature wearing down in its bearings so as to cause rubbing on the field pole faces is reduced. A reduction in bearing wear might also make it practical to reduce the air gap of the motors to a considerable extent, which would produce a lighter motor for the same power or a more powerful motor of the same weight. Another advantage claimed for the use of ball bearings is that, due to the narrow construction at the bearings, it may be possible to build the whole motor narrower, so that a motor of greater capacity could be designed for a certain width between the wheels. The use of anti-friction bearings also should produce a considerable saving in the cost of lubrication, since this type of bearing should require lubrication only about every six months instead of weekly, as is the requirement for plain bearings. While some installations have met with considerable success, others have failed and have caused great inconvenience. Bearings with insufficient load-carrying capacity have been applied, together with improper design of the bearing housing. As a result of this last consideration, bearings have been damaged while being mounted or when removed for the purpose of inspection. Other failures have resulted from stray current passing through the bearings, as ball bearings are very sensitive to electric current, and even a very slight amount passing continuously through a ball bearing will cause pitting and premature destruction. Another difficulty which has been encountered in the use of anti-friction bearings is that due to heavy shocks, particularly in the case of old and worn tracks. Ball bearings are quite sensitive to shock loads, due to the fact that their capacity to carry even temporary overloads is not much larger than their continuous load-carrying capacity. While proper selection and mounting may to a considerable extent overcome such failures, and while several arrangements have been proposed for overcoming them, still in spite of the great efforts made to design reliable ball bearings for railway motors, the margin of safety obtainable is not quite satisfactory.

Roller Bearings. Properly designed roller bearings possess the same advantages as ball bearings and the additional ones that they do not appear to be subject to pitting by electric current to anywhere near the same degree as ball bearings, and that they are capable of sustaining comparatively heavy shock loads. It can be said conservatively that a roller bearing will carry about 60 per cent more load than a ball bearing of the same outside dimensions and stand more than double the shock load. For the above reasons the results obtained with precision roller bearings in electric railway service have been on the whole far more satisfactory than those obtained with ball bearings. Roller bearings have seldom been used for electric railway motors in America. The reason for this is that "precision" roller bearings, as they are termed in Germany, have not as yet been marketed to any large extent in the United States. The term precision roller bearings is applied to those which are manufactured of as high grade material as the best ball bearings, are of equal precision in the workmanship and have about the same coefficient of friction. The commercial roller bearings most widely marketed in America do not measure up to these standards. They have a coefficient of friction which is from two to five times larger than that of high class ball bearings and do not meet the requirements for railway motor service. A description of various types of roller and ball bearings, and the results obtained from their use, is given by Mr. Wikander in *Elec. Ry. Jour.*, Vol. 60 (1922), p. 935.

Sleeve Bearings. This is the type of bearing most commonly used. A sleeve or plain bearing can be either a split or solid cylinder of a hard metal lined with babbitt, or a split or solid cylinder of babbitt or bronze material not lined. It should have enough clearance over the journal to allow a thin film of oil to form between the journal and the bearing to float the journal. With this type of bearing, the running friction is low if the film of oil is constantly maintained and is free from dirt, although the starting friction is comparatively high. The bearing will last for a number of years if it is not damaged mechanically.

To allow bearings to be removed without taking off the pinion, most of the older motors are provided with split armature bearings at the pinion end. Due to improved lubrication, with a resulting increased life of bearings, all modern motors use solid bearings at both ends. This gives a much better mechanical design, which does not require frequent renewals. The pinion end bearings are always larger than those on the commutator end. Axle bearings are always made in two halves, so they can be removed conveniently. The two bearings are of the same size and in modern type motors are interchangeable.

The most common types of armature bearings are made of either a bronze or a malleable iron shell lined with babbitt. The babbitt lining of a bronze shell is less than the single air-gap in thickness, so as to save the armature core from rubbing on the pole pieces should there be excessive wear or the babbitt melt out of the shell. Depending primarily upon size of axle and size of axle bearing seat in the motor frame, axle bearings are made of either bronze tinned or malleable iron lined with a soft alloy. Other factors, such as first cost, cost of repair, and the experience of the operating man, enter into this selection.

Babbitt Metal used to line railway motor bearings consists either of a tin base or of a lead base alloy; in other words, the bearing metal should be an alloy composed of at least 80 to 90 per cent tin or lead. Both classes give good service if properly handled during the melting and pouring process. Where large numbers of bearings of the same size are used, sufficient to support the expense of special tools, the surface obtained by broaching is found most satisfactory. With many operating companies it is the common practice to machine armature bearings after rebabbitting, although some consider this unnecessary, and babbitt bearings to exact size. Whichever method is used, it is customary to scrape the babbitt to get a uniform bearing surface. Axle bearings, when not lined with babbitt, are given a machined finish and this surface often is tinned to fill up the small irregularities, thus helping the bearing to seat itself while new. When axle bearings are lined with babbitt, they should be machined to get best results in service, although it is the practice of some operators to babbitt to exact size and fit by scraping.

Method of Holding. In addition to a press fit of from three to five tons, keys are commonly used to prevent armature bearings from turning in the housing. Most of the older motors without housings have their bearings held from turning by means of a dowel in addition to the clamping action of the frame. Depending largely upon the size and location of the lubricating openings, axle bearings are held from turning by a dowel or a key, as well as by the clamping fit of the axle caps. Dowels in the flange or shoulder of the bearing shell are being used successfully in modern motors. Special forms of lugs cast on the bearings, or plates inserted between the two halves of the bearings, have given satisfactory results in service, but are difficult to manufacture.

Clearances which are considered good practice for railway motors using grease or oil waste lubrication, are:

	Minimum	Maximum
From 2 in. up to and not including 3 in	0 006 in.	0 008 in.
From 3 in. up to and not including 4 in	0 008 in.	0 010 in.
From 4 in. up to and not including 5 in	0 010 in.	0 014 in.
From 5 in. up to and not including 6 in	0 014 in	0 016 in.
From 6 in. up to and not including 7 in	0 016 in	0 018 in.

Oil grooves are machined, cut or moulded in armature bearings so as to help the oil to enter and more evenly distribute itself throughout the bearings. In general they are not required in the case of axle bearings, on account of the slow speed and low unit pressure. The sharp edges at the split of axle bearings should be beveled to prevent wiping away of the oil film by these edges, but this bevel should not extend to the ends of bearings, as it would drain off and waste too much oil. It is good practice to round off or bevel the edges of the windows in both armature and axle bearings to encourage the flow of oil into the bearing surface.

Bearing wear, especially that of armature bearings, should be carefully checked by periodic inspection to avoid armature rubbing on pole faces. Such inspection, sometimes made visually, is more satisfactorily and safely done by means of sweep gages inserted between armature and pole face. Where the direction of rotation, especially with single-end cars, is such that the bearing pressure is up, means must be provided to lift the armature so as to measure the minimum air gap at the top, as well as under the armature. This may be done mechanically or by the application of power on the first controller notch with the brakes set.

Lubrication of Motor Bearings.

Grease. Heavy grease, stored in a box or cup over the bearing, melts and runs on the journal as heat is developed by the friction of the bearings.

Grease and Oil. Same as above, with the addition of an oil well located under the bearing, from which oil is fed up to the bearing by means of a felt wick.

Oil and Rings. Oil well located directly under the bearing, in which a small brass ring runs suspended on the journal, and carries the oil to the bearing.

Oil and Waste. Oil well below and to one side of bearing, packed with saturated waste which presses against the journal through a window.

Vaseline Packed. The entire bearing packed in vaseline. Used mostly in the case of anti-friction (ball and roller) bearings.

Circulating Oil. Where the oil is forced through the bearing by means of a small pump.

Special Adaptions. Small oil cups placed in the grease box, from which wicks are suspended down to the top of the journal.

Joggle Type. Where a grease cup is used as an oil well and a metal ball or pin is placed at the opening of the journal.

The older types of railway motors used the grease method, later types grease and oil, and modern types, oil and waste.

Oil and Waste Lubrication. Use a good grade of mineral oil, light oil in winter and heavy oil for summer use. For best results, use a long fiber wool waste. Before using, it should be saturated in oil for at least 24 hours, and left on a screen or grating to drain for several hours. The oil wells should be of ample capacity to hold sufficient oil to last between inspection periods, should have an accessible opening for inspection and refilling, and should be provided with a tight fitting cover, held in place by a strong spring or bolts to keep out water, dirt or grit. Provision should be made for proper drainage of the spent oil, and means provided to gage the depth of oil at regular inspection periods. Before the bearings are packed, all water, dirt and small particles of metal should be removed from the oil well. Saturated waste should be loosely packed in the oil chamber and forced into place by a pronged rod of brass or some other soft metal, so that it will not injure the journal. In this manner, pack the waste up over the bearing window, forcing it in place, so that its springy action tends to hold the waste against the journal. Well-designed bearings of this type, if in good condition, properly packed with a long fiber wool waste and using a good grade of oil, should run from one to three weeks between oilings. The time is determined largely by the system of inspection of the other equipment on the cars, which makes it advisable for each operator to work out in actual service the most suitable oiling schedule to fit the operating conditions of his equipment. One to two gills of oil per bearing are required at each oiling period. This varies with the size of motor and the location of the bearings and upon the length of time between oiling periods and the service conditions. It is considered good practice to repack the bearings every three months, at this time removing all the waste, discarding that which is glazed and charred, refilling the bearings with good, clean, old waste, to which has been added sufficient new waste. About once a month it is advisable to "tease up" the waste in the bearings to make it more effective.

Grease is used in connection with the lubrication of some of the old type railway motor bearings. Grease consists of a fatty soap impregnated with a mineral oil. The solid part acts as a carrier for holding the oil in position, and has little value as a lubricant. Greases are usually graded according to their stiffness, from the softest to the hardest, with numbers indicating their relative melting point. A limesoap grease with a neutral reaction (which does not show any traces of an acid or an alkali) gives best results in service. To insure its being retained in the bearing, the grease should have a melting point of 10 to 15 deg. C. above the normal operating temperature.

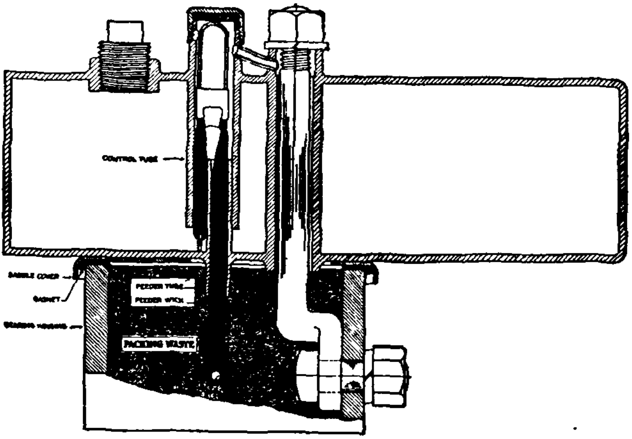


FIG. 65.—Rico oiler for railway motors.

Adaptation of Oil Lubrication to Old Type Motors Originally Designed for Grease. A number of devices have been designed for this purpose, among which is the "Rico" oiler, as shown by Fig. 65. The reservoir, 3 in. diameter and 10 in. long, with a capacity of approximately 8 gills, is provided with a removable filler plug at the top. A bushing extends entirely through the reservoir for the anchor bolt. In the interior of the reservoir are two tubes, one inside of the other. The larger, or control tube, extends from a point above the bottom of the reservoir up through the top, and is closed at the upper end by a cap. The smaller, or feeder tube, extends from a point near the center of the reservoir down through the bottom. A vent tube connects between the control tube and the anchor-bolt bushing. The feeder wick is inserted in the control tube, the short leg extending to the bottom of the reservoir, with the long leg inside of the feeder tube and projecting out the lower end. After filling the reservoir, inserting the filler plug seals the interior against the atmosphere. The oil feed is regulated by the number of strands in the feeder wick, different styles being used according to the rate of feed desired.

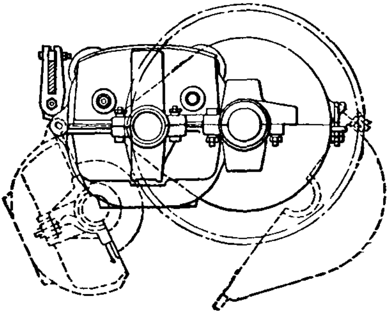


FIG. 66.—Nose suspension of motor

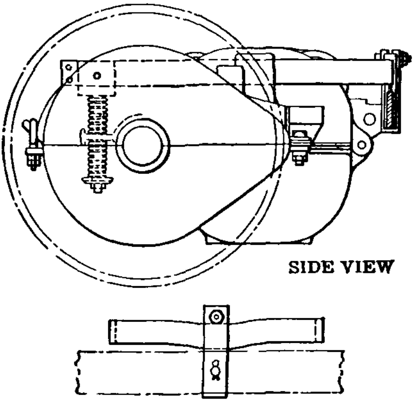


FIG. 67.—Cradle suspension of motor.

Motor Suspension on Cars. *Nose Suspension* (Fig. 66) is the most common method of motor suspension on motor cars. Springs in the motor nose suspension lessen shocks during starting or sudden changes of torque. Approximately 60 per cent of the weight of the motor is carried directly on the axles without spring support.

Cradle Suspension (Fig. 67). The total weight of the motor is hung by lugs on either side from a longitudinal horizontal bar which at the back end is spring supported from lugs on the arm which carries the axle bearing and at the front end by a cross beam and the truck frame. This type of suspension is seldom used.

Spring Supports. The motor manufacturers do not ordinarily furnish suspension parts other than the bracket or nose casting on the motor frame, and the suspension problem usually has been dependent on solution by the truck manufacturers. A resilient suspension for motors is very desirable, to relieve shocks and hammer blows; too short a spring gives too rigid a suspension. The truck suspension should be so designed that the motors are reasonably free to move transversely with the axle; otherwise axle thrust collars will wear rapidly and there will not be freedom of movement between the motor and the truck frame. The suspension also should be designed with as few wearing parts and of as simple design as possible, so that the motor can be removed from the truck with minimum labor. It is generally conceded that three-point suspension presents certain advantages over the other forms, but unfortunately truck space frequently prevents its use.

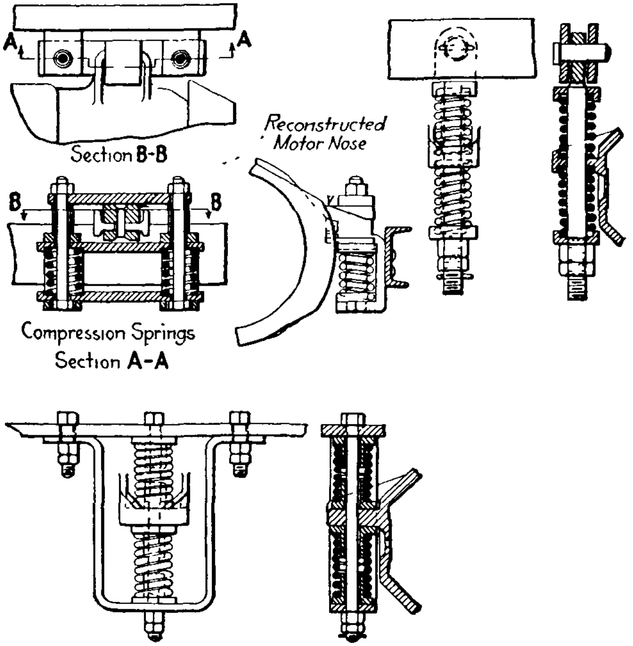


FIG. 68.—Improved types of motor spring suspension proposed by A.E.R.E.A. Equipment Committee, 1923.

The 1923 Committee on Equipment, Am. El. Ry. Eng. Assn., states that the springs should be so designed that they will compress 1 in with the following loads and should compress ½ in. more before going solid, thus allowing for a 50 per cent overload; this allows a normal movement of 1 in and a total movement of 1½ in.

	Load in pounds	
Motor, h.p.	Downward	Upward
25	2000	1200
35	3000	1500
40	3500	1800
50	4000	2000
65	5000	2500

It has been found in practice that many suspensions, originally designed to have from 1 in. to 1½ in. motion, have been screwed down by the repair men until they have practically no motion. This condition could be avoided if shoulder bolts were used. Some operators have removed the spring suspensions entirely, due to excessive spring breakage, but very probably at the expense of commutation and excessive maintenance.

The Committee presented three types of motor suspension spring arrangement, as shown in Fig 68, in each of which proper allowance has been made for lateral motion without spring distortion. It is believed that while these designs or modifications of them may cost slightly more than those then in common use, the increased cost would be offset by reduced maintenance cost within a short time.

SECTION V

CONTROLLING APPARATUS

Controllers for Direct Current Series Motors

Rheostatic Control. Rheostatic control is carried out entirely by making changes in the resistance of the motor circuit to maintain the desired current through the motor during acceleration. In this method there is no arrangement for series-paralleling. Rheostatic control may be used with equipments of one or more motors, but because of the greater energy economy secured by series-paralleling in two or four motor equipments, the use of rheostatic control is practically limited to single-motor equipments or two-motor equipments on double voltage. The hand controller commonly used to make the necessary connections for rheostatic control is known by its trade name, the type "*R*" controller.

Series-parallel Control. The series-parallel method of control is the method most commonly used with two- and four-motor equipments. In starting a two-motor equipment by this method the two motors are first connected in series and in series with the control resistance, then the control resistance is reduced by steps until the motors are running in series on the working conductor potential; control resistance is then added as the motors are connected in parallel, after which the control resistance is reduced by steps until the motors are running in parallel on the working conductor potential. Four-motor equipments are usually arranged in two groups of two motors permanently connected in parallel with each other and these two groups are controlled as the two motors of a two-motor equipment. There are two general methods of making the transition from series to parallel connections in series-parallel control, one opening the power circuit, while the other leaves it closed.

Type "K" Controller. The type "*K*" controller does not open the power circuit during transition from series to parallel. This type of controller is in most common use. The type "*K*" controller is arranged to cut the current off half the motors during transition. This is done by first shunting this half, then disconnecting it before placing the halves in parallel, and is known as the "shunt" type of transition. Some forms of the type "*K*" controller were designed for the use of the "bridging" type of transition from series to parallel, but these have been superseded by those using the "shunt" transition

Type "L" Controller. The type "*L*" controller opens the power circuit during transition from series to parallel connections of the motors. This type of controller is little used.

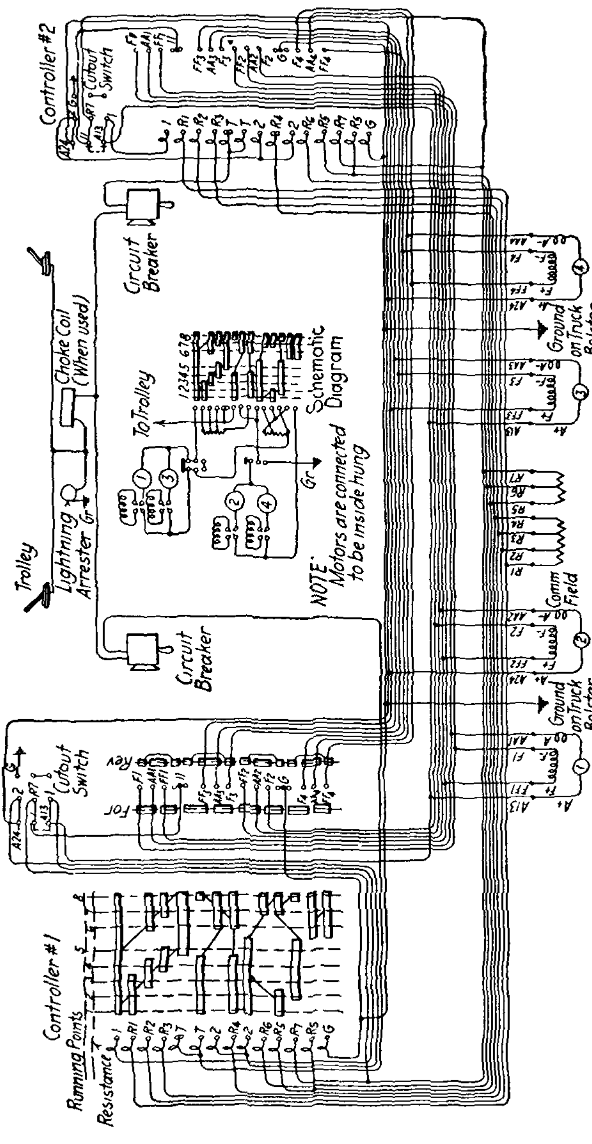


FIG. 1.—Diagram of connections, two K-35-HH controllers with four 600-volt motors.

Type "B" Controller. The type "*B*" controller has the usual power connections and in addition it is arranged to make connections for electrical braking by operating the motors as series generators to excite the electrically operated brakes. This method of electric braking imposes an additional load upon the motors, thus necessitating larger motors.

Shunt Transition. There are three transition steps between series and parallel, as shown in Fig. 2. On the first transition step, part of the resistance is inserted in the circuit with the motors still in series. On the second transition step, one of the motors is short-circuited or shunted, which gives the name to this method. On the third transition step, the shunted motor is disconnected from the other motor and the shunt. On the second and third transition points, one motor is developing no torque, but the other is still working. Therefore, this method has an operating advantage as compared to the open circuit method in that only one half of the torque is dropped during transition instead of all torque being lost. Series-paralleling by this method is used with most platform controllers of the *K* type and with unit switch control except where automatic operation is required or the current handled is comparatively large.

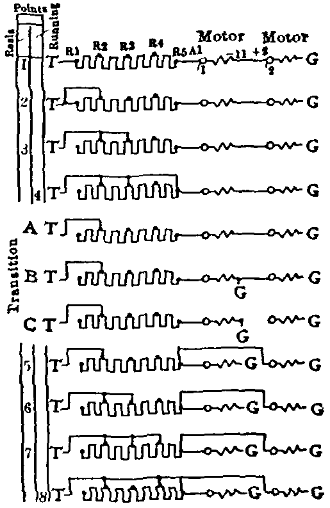


FIG. 2.—Sequence of connections type *K* controller, shunted motor transition.

Bridging Transition. Figs. 3 and 4 illustrate diagrammatically the sequence of connections in the bridging method of control. This method has the advantage that both motors (or pairs of motors) are developing torque at all times through the transition period. It has been used in some type *K* platform controllers, but on account of heavy arcing at controller fingers under certain conditions in transition, its use has been discontinued in platform controllers, and is now confined to unit switch control where automatic operation is desired or heavy current is handled.

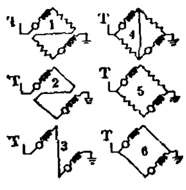


FIG. 3 —Diagram of bridging method of control.

Three-speed Control. Fig. 5 illustrates the various combinations of the three-speed motor control which was designed by P. N. Jones of the Pittsburgh Railways Co. primarily for use with low floor cars and small motors. The amount of resistance required is

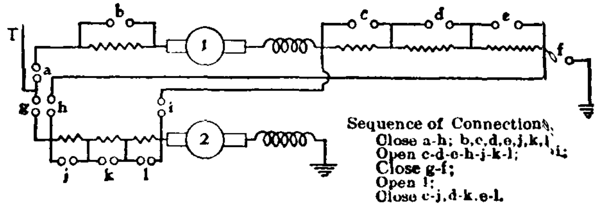


FIG. 4.—Circuits of bridging type of control.

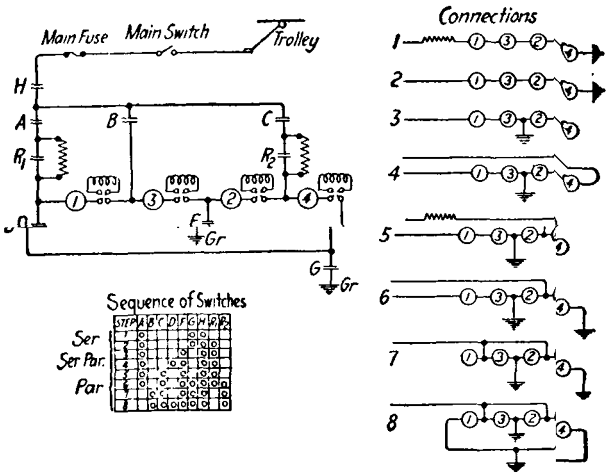
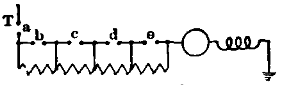


FIG. 5.—Diagrams of circuits, three-speed four-motor control.

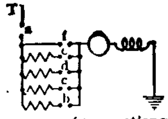
reduced to a minimum by the use of four motors in series on the first two controller points, and the special connections shown in Fig. 5 for the fifth, sixth and seventh points.

Resistance Connections. There are three general methods of connecting control resistance in the motor circuit, the series, the parallel, and the combination method. In the series method (Fig. 6) all the resistance is connected in series at the beginning and it is cut out of the circuit by short-circuiting sections of it, progressively. By this method much radiating surface is made available during starting. *In the parallel method (Fig. 7)* the resistance in circuit is reduced by adding resistor sections in parallel, progressively, to the sections already in circuit, thus reducing the total resistance of the circuit and finally short-circuiting all the resistance. This requires smaller switches than the series method. Fig. 8 is an illustration of a combination of these two methods.



Sequence of connections:
Close a,b,c,d,e, consecutively

Fig. 6.—Series method of connecting control resistance.



Sequence of connections:
Close a;
Close b,c,d,e,f, consecutively

Fig. 7.—Parallel method of connecting control resistance.

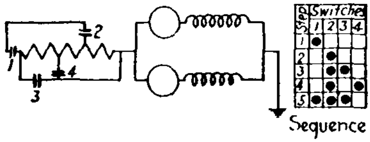


Fig. 8.—Series-parallel method of connecting control resistance.

Controller Rating. The rating of a controller is based upon the combined nominal 1-hour rating of the motors it is to control. This rating is generally given for a certain voltage and for a voltage variation from this value, but within the insulation and arc-breaking capacity of the controller, the horse-power capacity of the controller will vary approximately as the voltage.

Reversing Series Motors. The direction of rotation of a series motor may be reversed by inverting the relation of the direction of the current in the armature to the direction of the current in the field. This may be done by reversing the connections to the armature or by reversing the connections to the field. Older types of controllers were arranged to reverse the connections to the brushes, but controllers for use with commutating pole motors are arranged to reverse the connections to the field and leave the interpole winding permanently connected on the ground side of the armature, thus securing minimum insulation strains on the interpole winding. Controllers built to reverse the connections to the brushes can be arranged to reverse the connections to the field.

In order to simplify the control with some types of field-control equipments, the main field is permanently connected to ground and the armature and commutating field are reversed as a unit, thus throwing the commutating-pole winding next to the trolley. With modern insulation and construction this is permissible, although less desirable than to retain the armature always next to the trolley, in its relation to the commutating field.

High Voltage Direct Current Control. The control for 1200-volt, 1500-volt, 2400-volt and 3000-volt car equipments is essentially the

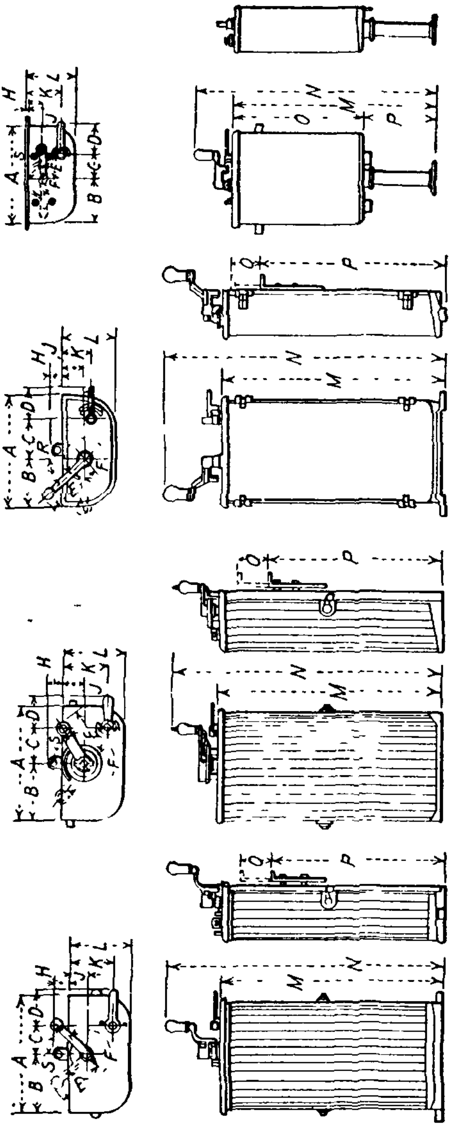


Fig. 9. Fig. 10. Fig. 11 Fig. 12.

Figs. 9, 10, 11, 12.—Drum type controllers—see dimension table, page 325.

same as that described above for 600-volt service except that two motors are usually connected permanently in series. Series-parallel connection of the two groups of motors provides for half speed and full speed, but when full speed is desired on 600-volt supply, a commutating switch is provided with the 1200-volt equipments which operates to change the permanent connection of the motors from two in series to two in parallel. Usually, however, two motors remain permanently in series and provide half speed when 1200-volt cars run over terminal tracks supplied with energy at 600 volts. A dynamometer or motor-generator set run from the trolley supplies current at a low voltage for actuating the switches. The air compressor and heaters operate direct from the trolley circuit.

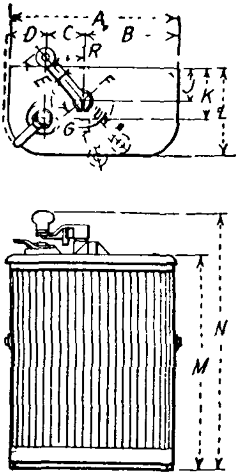


Fig. 13.

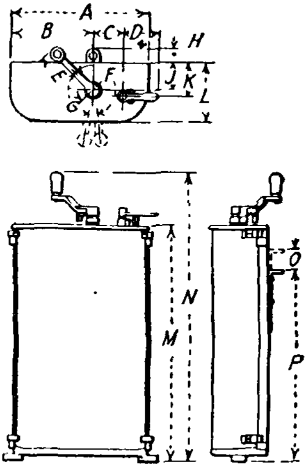


Fig. 14.

Figs. 13 and 14.—Drum type controllers—see dimension table, page 325.

Drum Type Controllers. There are in use a very large number of forms of drum type controllers, varying in minor details of circuits. The general types, with capacities, number of control points, and weights, are as shown in the following tables. Minor variations in type are designated by a sub-letter following the type letter and number as "*K*-39-*C*." Unless otherwise noted, the ratings given are for 600 volts, and the maximum peaks should be limited to 750 volts. Type *B* controllers are for electric braking service; type *R* are rheostatic control.

Type	No of motors	Max allowable capacities, each motor (neither to be exceeded)		Number of points		Approx. weight in lb	Remarks
		Hourly rating, H P	Continuous rating amp	Series	Parallel		
K-6	4	40*	43*	6	5	265	
K-10	2	40*	41*	5	4	210	
K-12	4	30*	36*	5	4	225	
K-27	2	60*	65*	4	4	225	Metallic return circuit.
K-28	4	40*	41*	5	5	240	
K-29	4	40*	43*	6	5	265	Metallic return circuit.
K-31	4	30*	36*	4	4	230	Metallic return circuit.
K-35	4	65	60	5	3	270	
K-39	2	70	66	4	4	230	Metallic return circuit
K-40	4	65	60	5	3	280	Metallic return circuit
K-47	4	75	65	6	4	437	Two motors permanently in series for 1200 volts
K-51	2	70	66	5	4	250	For tapped field motors
K-63	2	40	38	4	3	135	
K-64	4	110	105	6	4	450	
K-67	8	40	37	5	3	270	For train operation, furnished with line breaker and separate motor reverser
K-68	2	70	66	4	4	225	
K-45	2	32†	70†	4	3	122	For storage battery service.
K-52	4	16†	35†	4	3	127	For storage battery service.
B-50	4	60	53	5	4	492	9 braking points
B-51	2	120	105	5	4	492	9 braking points
B-54	2	75	75	4	3	288	7 braking points.
R-17	1	50*	55*	6		180	
R-200	2	90‡	65‡	6		245	Two motors permanently in series.

* Rating at 500 volts; maximum peaks of 600 volts.
† Rating at 250 volts; maximum peaks of 275 volts.
‡ Rating at 750 volts, maximum peaks of 1650 volts
All others rated at 600 volts, maximum peaks of 750 volts.

Dimensions of Drum Type Controllers

See Figs 9 to 14, inclusive F, G, R and S in degrees All other dimensions in inches.

Type	Fig	A	B	C	D	E	F Deg	G Deg.	H	J	K	L	M	N	O	P	R Deg	S Deg
K-6	11	17 1/16	6 21/32	5 11/16	5 3/4	8	327		2 1/2	3 13/16	4 11/16	9 1/16	38 11/16	47 3/16	4 7/16	33 3/8	45	
K-10	11	17 1/2	7 7/16	6 11/16	5 3/4	8	325		2 1/2	3 3/16	4 5/16	8 3/16	33 3/16	42 1/4	4 1/2	29	45	
K-12	11	17 1/2	7 7/16	6 11/16	5 3/4	8	325		2 1/2	3 3/16	4 5/16	8 3/16	33 3/16	42 1/4	4 1/2	29	45	
K-27	11	17 1/2	7 7/16	6 11/16	5 3/4	8	325		2 1/2	2 7/8	4	8 3/16	35 1/16	44 1/8	4 1/2	29 7/8	45	
K-28	11	17	7 1/16	5 11/16	5 3/4	8	324		2 1/2	3 5/8	4 1/2	8 1/8	36	44 1/2	4 5/8	30 15/32	45	
K-29	11	17 1/16	6 21/32	5 11/16	5 3/4	8	327		2 1/2	3 13/16	4 11/16	9 1/16	38 11/16	47 3/16	4 7/16	33 3/8	45	
K-31	11	17 1/2	7 7/16	6 11/16	5 3/4	8	325		2 1/2	2 7/8	4	8 3/16	35 1/16	44 1/8	4 1/2	29 7/8	45	
K-35	9	18 5/8	9 11/16	4 5/8	5 3/4	8	264		2 11/16	3 1/4	6 9/16	9 3/4	36 7/8	44 7/8	4 1/2	26 15/16		48
K-68	9	16 3/4	10	4	5 3/4	8	264		2 11/16	3 1/4	6 1/4	9 13/16	36	44 1/2	4 5/8	27 11/16		48
K-39	9	16 3/4	10	4	5 3/4	8	264		2 11/16	3 1/4	6 1/4	9 13/16	39 1/2	48	4 1/2	31 7/16		48
K-40	9	18 5/8	9 11/16	4 5/8	5 3/4	8	264		2 11/16	3 1/4	6 9/16	9 3/4	42	50 1/2	4 1/2	26 15/16		48
K-45	12	17 3/4	8 3/16	4	5 3/4	6	190		3/8	3	6	8 5/8	36	42 15/16	23	13		85
K-47	10	22 1/4	10 5/8	6 1/2	5 3/4	8	300		2 11/16	4 1/8	8 3/8	11 7/8	43 11/16	52 1/8	4 1/2	33 1/2		30
K-51	9	18 1/16	9 5/16	4 5/8	5 3/4	8	290		2 19/32	3 9/16	6 7/8	10 1/16	36 3/8	44 7/8	4 1/2	30 7/16		35
K-52	9	17 3/4	8 3/16	4	5 3/4	6	190		1/4	3	6	8 5/8	36	42 7/8				89
K-63	9	12 5/8	6 7/16	4	4 1/2	8	300			3 3/16	6 3/16	8 1/8	35 1/2	44 9/16				30
K-64	10	22 1/4	10 5/8	6 1/2	5 3/4	9 1/2	244		2 11/16	4 1/8	8 3/8	11 7/8	43 5/8	51 1/2	4 5/8	33 1/2		58
K-67	9	18 5/8	9 11/16	4 5/8	5 3/4	8	264		2 11/16	3 1/4	6 9/16	9 3/4	36 7/8	44 7/8	4 1/2	26 15/16		48
B-50	13	25	14	5 5/8	6	9 7/8	185	148		4 3/4	7 1/4	12 3/4	43 1/4	49 1/8			45	
B-51	13	25	14	5 5/8	6	9 7/8	185	148		4 3/4	7 1/4	12 3/4	43 1/4	49 1/8			45	
B-54	14	18 5/8	8 3/8	5 7/8	5 3/4	8 1/4				4 5/16	6 5/8	10 1/2	35 7/8	41 7/16		30 1/8		
R-17	11	17 1/2	7 7/16	6 11/16	5 3/4	8	315		2 1/2	3 3/16	4 5/16	8 3/16	33 3/16	41 11/16	4 5/8	28 15/16	45	
R-200	9	18 5/8	9 11/16	4 5/8	5 3/4	8	205		2 11/16	3 1/4	6 9/16	9 3/4	36 3/8	44 15/16	4 1/2	26 15/16		77 1/2

Auxiliary Line Switch Equipment. Fig. 15 shows the circuits of an auxiliary line switch (or contactor) equipment. It consists of a ratchet attachment in the controller, combined switch and fuse for protecting the line switch operating circuit, and a box containing a line switch (pneumatically or magnetically operated) with the overload relay and necessary resistance tubes. The energy for operating the magnet valve of a pneumatic line switch, or the electromagnet line switch, is obtained from an auxiliary circuit carried through the contacts of the ratchet attachment in the drum controller and the control switch. This auxiliary circuit is called the control circuit. The pneumatic line switch is operated by compressed air taken from the main reservoir of the air brake system on the

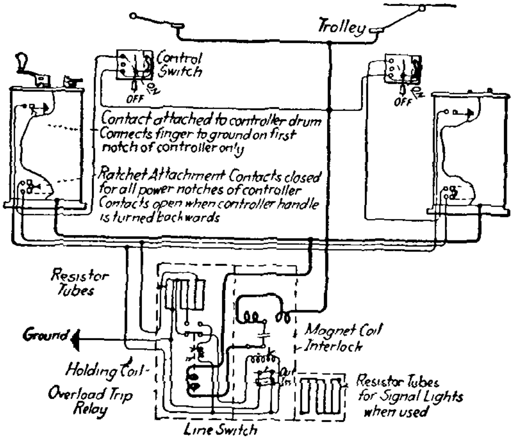


FIG. 15.—Circuits of auxiliary line switch equipment.

car through a cut-out cock and strainer. The admission and release of air from the line switch air cylinder is accomplished by a magnetically operated valve. The ratchet attachment which is usually mounted near the lower bearing of the main drum operates to close the control circuit contacts when the controller handle is placed on the first notch. The auxiliary contacts then remain closed throughout all succeeding positions of the controller handle. If, however, the controller handle is moved backward an appreciable amount, the ratchet opens the control circuit to the line switch operating coil which causes the line switch to interrupt the main motor circuit. In order to close the line switch again, it is necessary to move the controller handle to the "off" position and then advance to the first notch. In case of an overload, or a surge due to a short circuit, the overload trip will operate to open the control circuit, thus opening the line switch and cutting off power. In order to re-establish the power circuit, it is necessary to return the controller handle to the off position and then move to the first notch. The control circuit then is automatically restored and the switch comes into normal operation. The regular operation of the line switch during each service application of power insures that it is always in good working order, and ready to operate in case of overload or short circuit. This auxiliary equipment accomplishes the purpose of making and breaking the main circuit in a properly designed switch outside of the controller and underneath the car.

Circuit-Breakers. To furnish protection against extreme overloads or grounds and short circuits, an adjustable circuit-breaker is connected between the trolley and controller. Circuit-breakers are rated according to the current which they will carry continuously, and for any specific equipment are selected so that the current rating of the breaker is approximately two-thirds of the nominal one-hour current rating of one motor multiplied by the number of motors in parallel in the equipment. The breaker usually is set to trip at a current 50 per cent greater than the nominal current rating of one motor multiplied by the number of motors in parallel per equipment. For example: a 600-volt car equipped with four motors each rating 100 amperes for one hour, should have a circuit-breaker with a continuous rating of $\frac{2}{3} \times 100 \times 4 = 267$ amp. Ordinarily this breaker should be set to trip at $1.5 \times 100 \times 4 = 600$ amp. On locomotives it is customary to set the breaker to trip at double full load on the motors. In the above case this would be $2 \times 100 \times 4 = 800$ amp.

Lightning Arresters. For protection against lightning and surges which occur in the contact line (trolley or third rail), a lightning arrester is connected next to the collector and a choke coil is connected in the power circuit between the lightning arrester tap and the circuit-breaker. If lightning or a line surge tends to produce an abnormal voltage at the car, which in turn tends to send a heavy rush of current through the equipment, the choke coil limits the current rush and piles up the voltage next to the arrester until the current finally jumps to ground through the arrester, the tension is relieved, and normal conditions are re-established.

Design of Starting Resistance for Direct Current Motors

The design of direct current railway accelerating resistance involves three distinct problems: (1) determination of the total amount of resistance required in the circuit when the controller is on the first notch, (2) determination of the number of steps (and their ohmic values) into which the resistance must be divided to give uniform acceleration, and (3) determination of the current carrying capacity of the various steps, necessary to prevent the overheating of the resistance elements.

The total *resistance required in the circuit at start*, and the amount of resistance that is cut out at any given instant, depend upon the permissible rate of acceleration. Assuming this it is possible to determine the total tractive effort which must be exerted by the motors. Dividing this total tractive effort by the number of motors gives the tractive effort which must be exerted by each motor. Knowing the gear ratio and wheel diameter, the value of the current which will produce the above tractive effort can be obtained from the characteristic curve of the motor. Then, by means of the line voltage, the total resistance can be determined. The value of the resistance obtained by this method must be reduced by a factor which varies from 0.7 to 0.8, *i.e.*, the resistance must be multiplied by approximately 0.75, to give the value of resistance which is actually placed in the circuit. The necessity for using this factor is due in general to two causes: first, the inductance of the motors, and second, the regulation of the transmission system. By following the method outlined and by using a constant of 0.75, the amount of resistance which is required in the circuit at start can be determined, and then by subtracting the motor resistance (see page 252) from the total the external resistance which is required is obtained.

Current Fluctuations During Straight Line Acceleration Period. When resistance is cut out of the motor circuit in passing from one controller point to the next, there is a sudden increase in current, resulting in an increase of tractive effort and a corresponding sudden increase in rate of acceleration. (See Fig. 16.) This is evidenced by a jerk the violence of which depends upon the magnitude of current fluctuation, train resistance and inertia and the tractive effort characteristic of the motor. Excessive sudden increase in rate of acceleration may cause discomfort to passengers, it may produce too great mechanical strains in trucks, draw bars or gears, or it may cause driving wheels to slip. The increase in rate of acceleration between controller points should be kept below about 0.6 mile per hour per second in passenger service, and still lower in freight locomotive operation. For ordinary conditions of city service, the corresponding maximum and minimum accelerating currents will be 10 to 15 per cent above and below the average, with total current fluctuations of 22 to 33 per cent. The degree of perfection to which the acceleration of a train is accomplished depends upon the number of steps into which the starting resistance is divided, the proportioning of these steps and the speed and degree of uniformity with which the steps are cut out. The proportioning of the resistance steps demands a study of equipment and conditions, as discussed below. The speed and degree of uniformity with which the steps are cut out are cared for automatically in the case of automatic control, but in the case of hand-operated control, except where an auxiliary regulating device is attached to the controller, they depend upon the skill and efforts of the motorman.

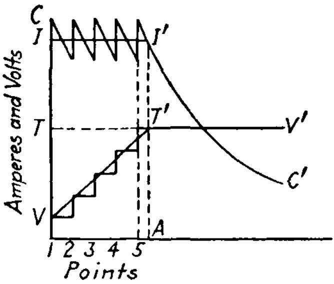


FIG. 16.—Fluctuation of motor current on successive controller points.

Analytical Method of Calculation of Motor Starting Resistance Steps. The following simple analytical method of determining motor starting resistances requires as preliminary data only the tractive effort-ampere curve and the resistance of the motor. No assumed constants are used, and the formulas may be applied to any type of series motor.

Let I_1 = current per motor at instant controller makes contact at any point, amperes (maximum current)
I_2 = current per motor at instant controller breaks contact at any point, amperes (minimum current)
T_1 = tractive effort per motor corresponding to I_1, pounds
T_2 = tractive effort per motor corresponding to I_2, pounds
$b = \dfrac{T_1 I_2}{T_2 I_1}$
E = line potential, volts
r = resistance of each motor, ohms (see page 252)
R_1 = total resistance of circuit at first series point of controller, ohms
R_s = total resistance of circuit at last series point of controller, ohms
R_a = total resistance of circuit at first parallel point of controller, ohms
R_p = total resistance of circuit at last parallel point of controller, ohms
r_1 = total external resistance in resistors at first series point of controller, ohms
r_s = total external resistance in resistors at last series point of controllers, ohms (= 0)
r_a = total external resistance in resistors at first parallel point of controller, ohms
r_p = total external resistance in resistors at last parallel point of controller, ohms (= 0)
s = number of series points on controller
p = number of parallel points on controller

Formulas for two motors in series-parallel connection:

Series connection

$$R_1 = \frac{E}{I_1} \qquad r_1 = R_1 - 2r$$

$$R_2 = \frac{E - b(E - I_2R_1)}{I_1} \qquad r_2 = R_2 - 2r$$

$$R_3 = \frac{E - b(E - I_2R_2)}{I_1} \qquad r_3 = R_3 - 2r$$

Continue until

$$R_s = \frac{E - b(E - I_2R_{s-1})}{I_1} = 2r \text{ and } r_s = R_s - 2r = 0$$

Parallel connection

$$R_a = \frac{E - b\left(\frac{E}{2} - I_2 r\right)}{2I_1} \qquad r_a = R_a - \frac{r}{2}$$

$$R_b = \frac{E - b(E - 2I_2R_a)}{2I_1} \qquad r_b = R_b - \frac{r}{2}$$

$$R_c = \frac{E - b(E - 2I_2R_b)}{2I_1} \qquad r_c = R_c - \frac{r}{2}$$

Continue until

$$R_p = \frac{E - b(E - 2I_2R_{p-1})}{2I_1} = \frac{r}{2} \text{ and } r_p = R_p - \frac{r}{2} = 0$$

As an example, it is desired to determine the starting resistances to be used for an 18 ton car, with two 40 h.p. 600-volt railway motors each having a resistance of 0.60 ohm, and series-parallel controllers having five series and four parallel points. The maximum current per motor in any particular case depends upon the weight of the car and load, the desired acceleration and the allowable motor current. The difference between the maximum and minimum values of current is determined by the number of controller steps, which in turn depends on the allowable variation from the mean acceleration. As a trial, take 60 and 79.5 amp. as the minimum and maximum currents. The corresponding values of tractive effort per motor, as read from the characteristic curve, are 1000 and 1460 lb., respectively, which will produce accelerations of 0.99 and 1.50 m.p.h.p.s., respectively. (See formula, page 151.) The change between steps, amounting to 0.51 m.p.h.p.s., is found to be well within the limit of 0.6 m.p.h.p.s. mentioned on page 328.

Series

$$\text{Then } b = \frac{T_1I_2}{T_2I_1} = \frac{1460 \times 60.0}{1000 \times 79.5} = 1.102$$

$$R_1 = \frac{E}{I_1} = \frac{600}{79.5} = 7.547 \text{ ohms}$$

$$r_1 = R_1 - 2r = 7.547 - 2 \times 0.60 = 6.347 \text{ ohms}$$

$$R_2 = \frac{E - b(E - I_2R_1)}{I_1} = \frac{600 - 1.102(600 - 60 \times 7.547)}{79.5}$$

$$= 5.507 \text{ ohms}$$

$$r_2 = R_2 - 2r = 5.507 - 2 \times 0.60 = 4.307 \text{ ohm}$$

$$R_3 = \frac{E - b(E - I_2R_2)}{I_1} = \frac{600 - 1.102(600 - 60 \times 5.507)}{79.5}$$

$$= 3.810 \text{ ohms}$$

$$r_3 = R_3 - 2r = 3.810 - 2 \times 0.60 = 2.610 \text{ ohms}$$

$$R_4 = \frac{E - b(E - I_2R_3)}{I_1} = \frac{600 - 1.102(600 - 60 \times 3.810)}{79.5}$$

$$= 2.399 \text{ ohms}$$

$$r_4 = R_4 - 2r = 2.399 - 2 \times 0.60 = 1.199 \text{ ohms}$$

$$R_5 = \frac{E - b(E - I_2R_4)}{I_1} = \frac{600 - 1.105^*(600 - 60 \times 2.399)}{80^*}$$

$$= 1.200 \text{ ohms}$$

$$r_5 = R_5 - 2r = 1.200 - 2 \times 0.60 = 0.0 \text{ ohm}$$

* Here I_1 = 80 amp., the maximum current to satisfy the condition $r_5 = 0$.

Parallel

$$R_a = \frac{E - b\left(\frac{E}{2} - I_2 r\right)}{2I_1} = \frac{600 - 1.102\left(\frac{600}{2} - 60 \times 0.60\right)}{2 \times 79.5}$$

$$= 1.944 \text{ ohms}$$

$$r_a = R_a - \frac{r}{2} = 1.944 - \frac{0.60}{2} = 1.644 \text{ ohms}$$

$$R_b = \frac{E - b(E - 2I_2R_a)}{2I_1} = \frac{600 - 1.102(600 - 2 \times 60 \times 1.944)}{2 \times 79.5}$$

$$= 1.232 \text{ ohms}$$

$$r_b = R_b - \frac{r}{2} = 1.232 - \frac{0.60}{2} = 0.932 \text{ ohm}$$

$$R_c = \frac{E - b(E - 2I_2R_b)}{2I_1} = \frac{600 - 1.102(600 - 2 \times 60 \times 1.232)}{2 \times 79.5}$$

$$= 0.640 \text{ ohm}$$

$$r_c = R_c - \frac{r}{2} = 0.640 - \frac{0.60}{2} = 0.340 \text{ ohm}$$

$$R_d = \frac{E - b(E - 2I_2R_c)}{2I_1} = \frac{600 - 1.064^*(600 - 2 \times 60 \times 0.640)}{2 \times 72.2^*}$$

$$= 0.300 \text{ ohm}$$

$$r_d = R_d - \frac{r}{2} = 0.300 - \frac{0.60}{2} = 0.0 \text{ ohm}$$

* Here $I_1 = 72.2$ amp, the maximum current to satisfy the condition $r_d = 0$.

If the equations for R_s and R_p, when using the assumed values for I_1 and I_2, do not equal $2r$ and $\frac{r}{2}$, respectively, or approach those values within reasonable limits, it will be necessary to make new assumptions as to limiting currents and recalculate. These limits are not necessarily the same in series as in parallel, but the average current must be kept the same if it is desired to retain a uniform rate of straight-line acceleration.

Following this calculation, it usually will be necessary to make some compromise adjustments in the external resistance actually to be used, first, to fit standard resistance grids, and second, when required by the design of the controller to use the same leads for both series and parallel points.

The foregoing applies, as stated, to a single pair of motors. Should the equipment consist of four motors in pairs permanently connected in parallel, or any other number or combination of motors, the values for current, tractive effort and motor resistance must be modified accordingly

Calculation of Starting Resistance Steps by Graphical Method. The following graphical method of determining starting resistances follows the same principles as the foregoing, and is shown in detail by Prof. A M. Buck in Bulletin No 90 of the University of Illinois, where a proof of its correctness is demonstrated.

As in the foregoing example, it is desired to determine the starting resistances to be used for an 18 ton car, with two 40 h p. 600-volt railway motors each having a resistance of 0.60 ohm and series-parallel controllers having five series and four parallel points. The maximum current per motor in any particular case depends upon the weight of the car and load, the desired acceleration and the allowable motor current. The difference between the maximum and minimum values of current is determined by the number of controller steps, which in turn depends on the allowable variation from the mean acceleration. As a trial, take 60 and 79.5

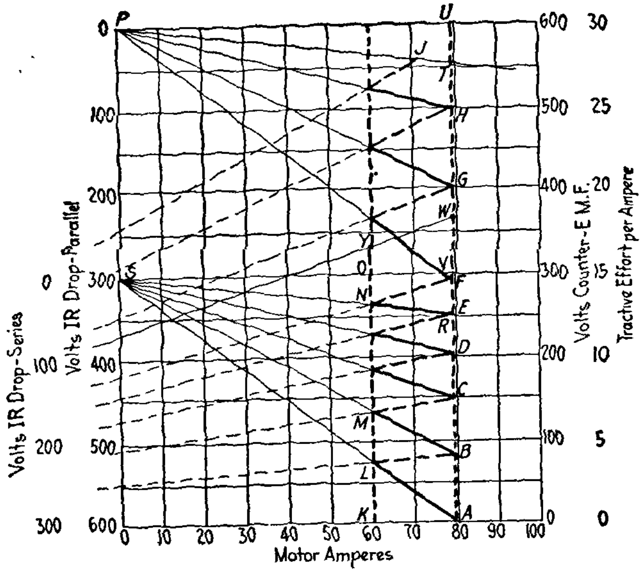


FIG. 17.—Graphical method of calculating starting resistance steps.

amp. as the minimum and maximum currents. The corresponding values of tractive effort per motor, as read from the characteristic curve, are 1000 and 1460 lb., respectively, which will produce accelerations of 0.99 and 1.50 m.p.h.p.s., respectively. (See formula, page 151.) The change between steps, amounting to 0.51 m.p.h.p.s., is found to be well within the limit of 0.6 m.p.h.p.s. mentioned on page 328.

Referring to Fig. 17, draw a diagram between motor amperes and motor volts to a scale of not more than 50 volts per inch and in about the proportions shown. If the line e.m.f. is 600 volts, then when the two motors are in series, each will be taking 300 volts, less the resistance drop. Draw the lines *SR* and *PT* so that the ordinates RV and TU represent the IR drop in one motor at any current I. Here RV and $TU = IR = 79.5 \times 0.60 = 47.7$ volts. Draw the line SA to represent the IR drop per motor when the external resistance is so chosen as to bring the motors (in series) to a standstill at 79.5 amp. When the current has fallen to 60 amp., the IR drop in one motor is represented by the ordinate NO, the drop in its external resistance by LN, and its counter-e.m.f. by KL. If the resistance is then reduced so as to increase the current to 79.5 amp., the counter-e.m.f. will be increased from KL to AB. To determine the location of the point B and succeeding similar points, plot to any convenient scale the points Y and W, which are the values of *tractive effort per ampere* for the particular values of current chosen. The tractive efforts are read from the characteristic curves of the motor under consideration. In this case the tractive effort per motor at 60.0 amp. is found to be 1000 lb. and at 79.5 amp. 1460 lb., and the corresponding tractive efforts per ampere are 16.66 and 18.36, as plotted at Y and W, respectively, on the 60 amp. and 79.5 amp. ordinates. The straight line WYX is drawn, through W and Y, intersecting the current axis prolonged at X, 131 amp. to the left of zero. The intersection has been omitted in Fig. 17 to save space. The location of this point may be determined accurately by the use of the following formula:

$$OX = \frac{T_1 I_2 - T_2 I_1}{T_2 - T_1}$$

Where OX = distance to left of zero, in amperes

T_1 = tractive effort per ampere for minimum current I_1

T_2 = tractive effort per ampere for maximum current I_2

$$\text{In this case } OX = \frac{(16.66 \times 79.5) - (18.36 \times 60.0)}{18.36 - 16.66} = 131 \text{ amperes}$$

If then the straight line XLB is drawn through X and L, intersecting AU at B, the ordinate AB will represent the counter-e.m.f. developed when the current has been increased from 60 amp. to 79.5 amp. without changing the speed. The ordinate VR gives the IR drop in one motor, and BR the drop external to this motor; the latter, divided by the current, determines the new value of external resistance per motor. The IR drop will then decrease along the line BS as the current falls off, until, at point M, the current must be increased again. The same construction is repeated until the two motors are in series without resistance. The current which will be obtained when the last point of resistance is cut out may be readily determined, since the IR drop in the motor alone is plotted as SR. When the last line radiating from X is drawn, it will intersect this line at some point as E. The abscissa determines the current, which, in this case, is 80.0 amp.

In changing to parallel, it is only necessary to move the axis of reference for IR drop to P, the ordinate for 600 volts, and continue the construction from that point in exactly the same manner as for series. The diagram, having been so constructed, may be used as follows to calculate the proper external resistances for the equipment:

Point on contr	From Fig. 17				Resistance per motor			Total external resistance
	Point	Volts C.E.M F.	*IR* drop	Current	*Total	Motor	External	
1	*A*	0	300	79 5	3.77	0 60	3 17	6 34
2	*B*	81	219	79 5	2.76	0 60	2 16	4 32
3	*C*	149	151	79 5	1.90	0 60	1 30	2 60
4	*D*	205	95	79 5	1.20	0 60	0 60	1 20
5	*E*	252	48	80 0	0.60	0 60	0 0	0 0
6	*F*	291	309	79 5	3 89	0 60	3 29	1 64
7	*G*	404	196	79 5	2 47	0 60	1 87	0 93
8	*H*	498	102	79 5	1 28	0 60	0 68	0 34
9	*J*	557	43	72 0	0 60	0 60	0 0	0 0

* Total resistance = *IR* drop ÷ current

The diagram may be used for any value of line potential without any further change than shifting the origins for the *IR* drop. For different current limits it is necessary to take new points of tractive effort per ampere, thus getting a new location for *X*, but a small variation in current limits may be made without relocating this point, and the error will not be great. As the resistors must be used both for the series and the parallel connections, a certain amount of adjustment usually must be made in the values of resistance found for definite current limits. The method of doing this is to continue the *IR* drop line corresponding to the actual resistance until it intersects the corresponding dash line originating at *X*. The limiting current values will then, of course, not be those selected at the outset, but will be as indicated by such intersections.

Should the equipment consist of four motors in pairs permanently connected in parallel, or any other number or combination of motors, it is obvious that the values for current, tractive effort and resistance must be modified accordingly.

Starting Resistance with K Controllers. The following table shows the division of resistance as commonly used with various K type controllers:

DIVISION OF TOTAL EXTERNAL RESISTANCE, K CONTROLLERS
(Per cent on each step)

	K10, K11, K12, K31, K36	K13, K14	K6, K29	K28	K35, K40	K42, K47	K44	K49, K51, K63
R1 -R2	47 2	47 5	27 4	22	26 6	50 7	16 6	41 6
R2 -R3	33 3	25 0	26 2	18	20 9	27 8	12 48	32 8
R3 -R4	11 2	8 75	25 0	27	15 8	8 38	11 00	25 6
R4 -R5	8 3	7 5	8 3	33		7 06	9 67	.
R5 -R6		6 25	7 05		20 9	6 06	8 55	.
R6 -R7		5 0	6 05		15 8			.
R7 R8							12 48	.
R8 -R9						..	11 00	.
R9 R10							9 67	...
R10 R11							8 55	.

Current Capacities for Resistance Grids. Having obtained a set of values for the resistance steps, the final operation is to determine the current-carrying capacities for these steps. Knowing the time spent on each controller point, the current, and the time for each complete cycle of the speed time curve, it is possible to determine the root mean square value of the current flowing through each step of the resistance. The resistance element that is used should be able to carry this value of current continuously without overheating. Also it should have sufficient thermal capacity to withstand the peak loads without dangerous heating. The capacities obtained *by this method are safe only on the assumption that the acceleration is automatic.* If the equipments are accelerated by hand, the capacities should be increased, the amount of increase depending upon the type of service It is obvious that the first grid to be cut out can have a lower continuous capacity than the last. Good practice is to allow from 15 per cent of the one-hour capacity of the motors as the capacity of resistance elements for the first step, up to 40 or 50 per cent for the last step. For locomotive service the capacities should be somewhat higher, ranging from 20 to 50 per cent. The following tables show the resistance and capacity of various resistance grids.

WESTINGHOUSE RESISTANCE GRIDS

8-in. cast iron			8-in. alloy			5-in. alloy		
Style	Ohms	Amp.	Style	Ohms	Amp.	Style	Ohms	Amp.
46,029	0.015	100.0	49,174	0.20	27.3	108,400	0.015	63.0
46,030	0.02	86.5	49,175	0.16	30.4	108,401	0.025	49.0
46,032	0.03	70.8	49,176	0.12	34.5	108,402	0.04	38.0
46,033	0.04	67.2	49,177	0.10	38.7	108,403	0.06	31.5
46,034	0.05	54.8	49,179	0.08	43.3	108,404	0.12	22.4
46,035	0.06	50.0	49,180	0.06	50.0	91,389	0.17	18.8
46,036	0.07	46.3	40,181	0.05	54.8			
46,037	0.08	43.3	49,182	0.04	67.2			
46,083	0.10	38.7	49,183	0.03	70.8			
			49,184	0.025	86.5			

Installation of Grid Resistances. The following should be taken into consideration with respect to the position occupied by the grid resistance: the grids should lie in the longitudinal plane of the car—this insures a maximum of ventilation, when it is necessary to expose the grid resistance to wheel wash, a splash guard similar to the one shown in Fig. 18 should be used; where it is necessary to place the grid frames close together, due to lack of room, sufficient space should be allowed between the top of grids and car underframing for making connections; clearance between bottom of grids and top of rail should be not less than 10 in., except that for low floor cars this figure can be reduced to 7 in.

A method of insulating grid frames for 600 volts is shown in Fig. 18. For 1200 volts the same arrangement is used with the addition of insulated bolts between the hangers and the straps to which the grid frames are attached. Fig. 18 also shows a cross-section of the bolt arrangement. The heavy black lines indicate the insulating washers and tube, while the dotted cross-sectional pieces are porcelain insulators. On equipments requiring more than five grid frames, it will be necessary to use heavier mounting straps or more hangers. The use of three hangers is preferable to heavy straps.

To satisfy the National Board of Fire Underwriters' requirements, it is necessary to observe certain regulations regarding insulation

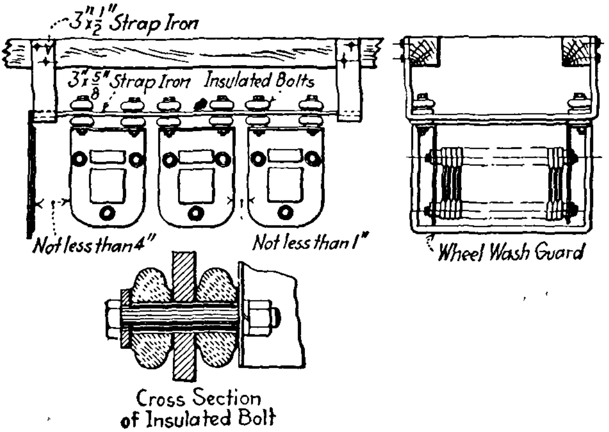


FIG. 18.—Installation of grid resistance.

as a fire protection. When the car underframing is composed entirely or partly of wood over the place where the grid resistance is hung, the regulations call for a fireproof non-metallic insulation at least 1/4 in. thick, covering a surface which extends 8 in. beyond the resistance on all sides. The cable insulation must be removed for a distance of at least 6 in. from the terminals of the grid resistance and supported in a manner to give a rigid construction and to prevent vibration.

Cables for Car Equipment. Wires in cables for car equipment usually have 7 strands for sizes smaller than No. 1 B. & S. gage, while No. 1 B. & S. and larger have 19 strands. In the following table, 1-Motor, 2-Motor, etc., indicate that the wire carries the current of a single motor in the former and of two motors in the latter case. The numbers are American or Brown & Sharpe gage sizes, except where shown in circular mils.

SIZES OF CABLE FOR CAR EQUIPMENTS

Nominal rated h.p. of motor at 600 volts	Trolley and ground		Motor wires		Resistance wires		
	2-Motor	4-Motor	1-Motor	2-Motor	1-Motor	2-Motor	4-Motor
25	5	2	7	5	7	7	6
40	4	1	6	4	7	6	4
50	4	0	6	3	7	6	3
65	3	00	5	2	7	5	2
75	1	000	4	0	6	4	2
100	0	0000	3	00	6	3	0
125	00	300000	2	000	5	2	00
140	000	350000	1	0000	5	1	00

Power-operated Control

Following are some of the advantages of power-operated control. These are secured with simplicity, reliability and low maintenance cost.

(1) *Multiple-unit operation is made practicable.* Thus the motive power may be distributed in small units without loss of efficiency, and since all the wheels thus may be made driving wheels a maximum tractive effort is made possible. A maximum closeness of speed adjustment is secured and strains on mechanical transmission and draft gear are reduced to a minimum.

(2) All circuit-breakers and other controlling devices which carry the main motor current are removed from the platform and placed beneath the car. This reduces the possibility of danger to passengers and stimulates more careful manipulation of the control. Platform weight and space economy are effected.

(3) Switches are operated with heavy contact pressure and reliable overload protection is secured.

The multiple-unit or power-operated type of control was designed primarily for the control of motors in a service requiring that cars be operated singly or with several cars coupled together in a train and operated simultaneously. When several cars are coupled together in a train, the train connections are so arranged that the motors on all of the cars may be controlled from either end of any car by a single operator, the cars being coupled in any desired relation and with either end of any car connected to any other car in the train. Although designed for train operation, power-operated control is widely used for the control of single-car equipments where the motors have a capacity of 50 h.p. or greater, owing to the size and weight of hand type control used with motors of large capacity. It is also coming into greater use for smaller equipments because it removes the power circuits and circuit-breaking apparatus from the car platform thereby removing a great source of annoyance and danger to passengers.

Power-operated control systems consist in general of two parts, the first consisting of a series-parallel motor controller which may be composed of a number of switches or circuit-breakers, sometimes called contactors, whose function is to effect the different electrical combinations of the motors and regulate the starting resistance in circuit with them. There is also a reverse switch for the motors, called the reverser, which is actuated by the same means used to operate the circuit-breakers or contactors. The second part comprises master controllers which operate the motor controlling contactors and reversers through the medium of train wires extending throughout the length of train so operated. A single operator may simultaneously effect similar combinations upon the several motor cars composing the train, thus giving to the train the same advantages of large tractive effort and rapid acceleration obtained in single-car operation.

The two systems of power-operated or multiple-unit train control in operation differ in the means employed to actuate the contactors and reversers, in one the contactor being operated electrically by the line current, while in the electro-pneumatic system the contactor or unit switch is closed by compressed air and opened by a spring, the necessary air valves being actuated electrically by the master controller through a train cable using storage battery or line current. Each system aims to produce the same result, that is, to duplicate the performance of a car operated singly, but the systems differ in the means adopted to secure this end.

Power-operated or multiple-unit control is divided into two general classes according to the method of acceleration, *i.e.*, hand-operated or automatic, the former being wholly under the control of the operator as is the case with the hand controller, while the latter is effected by the current limiting relays and interlocks on the contactors or unit switches.

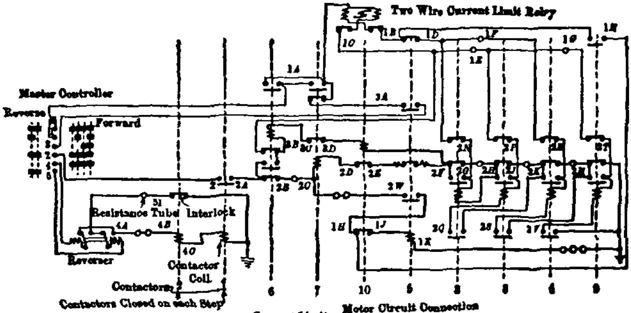
General Electric Power-operated Control

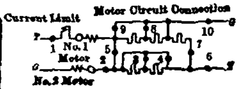
General Electric Contactor Control employs magnetically operated contactors or circuit breakers which are opened and closed in proper sequence by means of train wires which are energized through a master controller.

The Master Control, Fig. 19, comprises those parts which switch the control current operating the motor-control apparatus. The master controller is operated by hand, and is located in the vestibule at either end of the car. The motor control is local to each car, and current for this circuit is taken directly from the trolley or third-rail, through the contactors, starting resistance and reverser to the motors, thence to the ground. Where it is necessary to operate with a gap in the third-rail system, it is sometimes customary to install a train line so that any car may supply the motor current for the other cars of the train. The master control includes train wires made continuous throughout the train by means of couplers between the cars. On each car the operating coils of the motor control are connected to the train line through a cut-out switch, the train wires being energized in proper sequence by the hand-operated master controller on the platform. Current for the master control is taken directly from the trolley or third-rail through the master controller, which is being operated by the motorman, to the train-line, and thus to the operating coils of the contactors forming the motor control on all cars of the train.

The Master Controller is similar in design to the original cylinder controller in that it contains a movable cylinder and stationary contact fingers through which current is supplied in proper sequence to the different wires of the train line, for energizing the operating coils of the motor control. The value of the current required is very small, not exceeding 2.5 amp. for each car in the train. The master controller is provided with two handles, one for operating and one for reversing the train movement.

The Operating Handle is provided with a button which must be kept down except when the handle of the controller is in the off





Contactors Closed on each Step

Steps	Contactors 1	2	3	4	5	6	7	8	9	10
1st	●	●					●			
2nd	●	●	●				●			
3rd	●	●	●				●	●		
4th	●	●	●	●			●	●		
5th	●	●	●	●			●	●	●	
6th	●	●			●					
7th Pre.	●	●			●	●				●
7th Fin.	●	●				●				●
8th	●	●	●			●				●
9th	●	●	●			●		●		●
10th	●	●	●	●		●		●		●
11th	●	●	●	●		●		●	●	●

FIG. 19.—Sprague General Electric contactor control.

position, as releasing this button permits an auxiliary circuit to open, cutting off the supply of current to the master controller, thus de-energizing the train-line and opening the motor-control apparatus. This button is intended to serve as a safety appliance in case of physical failure of the motorman.

The Reverser Handle is connected to a separate cylinder which establishes control connections for throwing the electrically operated reverser into either forward or reverse position when the master-controller handle is on the off point. The operating circuit for the reverser is so interlocked that unless the reverser itself corresponds to the direction of the movement indicated by the reverser handle of the master controller, the line contactors cannot be energized.

The Contactor is a switch operated by solenoid coils, and each contactor may be considered as the equivalent of a finger and its corresponding cylinder segment in the hand-operated *K*-type controller. It consists of an iron magnet frame with an operating coil and two main contacts, one fixed and the other connected to the movable finger. These main contacts open and close in a moulded-insulation arc chute provided with a magnetic blowout. Interlocks are provided for making the necessary connections in control circuits to insure proper sequence in operating the different contactors. All of the contactors usually are mounted in a box placed beneath the car, this box being provided with a sheet-steel cover lined with insulating material.

The Reverser is a switch, the movable part of which is a rocker arm operated by two electromagnets working in opposition. The coils receive their energy from the master controller through the train-line, and the connections are such that only one coil can be operated at a time. Leads from the motors are connected to the main reverser fingers, and by means of copper bars on the rocker arm, the proper relations of armature and field windings are established for obtaining forward or backward motion of the car.

Circuit Couplers between Cars are so designed as to give a corresponding connection of train wires, this being secured by means of proper mechanical design of plug and sockets, it being rendered impossible to insert the plug in the socket improperly.

Sprague General Electric Automatic Multiple-unit Control provides for the acceleration of the train at a predetermined value of current in the motor, this feature being provided without preventing the manual operation of the master controller at less than the predetermined current if desired. The operation of the contactors is controlled from the master controller, but is governed by a notching or current-limit relay in the motor circuit, so that the accelerating current of the motors is substantially constant. This is accomplished by having small auxiliary interlocking switches on certain of the contactors, the movement of each connecting the operating coil of the succeeding contactor to the control circuit. The contactors are energized under all conditions in a definite succession, starting with the motors in series and all resistance in circuit; the resistance is subsequently cut out step by step; the motors are then connected in parallel with all resistance in circuit, and the resistance again cut out step by step. The progression can be arrested at any point, however, by the master controller, and is never carried beyond the point indicated by the position of the master controller. The rate of the progression is governed by the current-limit relay, so that the advance cannot be made at a rate so rapid that the current in the motors will exceed the prescribed limit. One of these relays is provided with each car equipment, so that while the contactors on each car of a train are controlled from the master controller for the application and removal of power, the rate of progression through the successive steps is limited by the relay on each car independently, according to the adjustment and current requirements of that particular car.

Protection Against Failure of Power. Another automatic protection for individual cars is provided by a second or potential relay, which has its coil connected to the lead from the collecting shoes of the respective car. The contacts are so connected in the contactor circuits, that in case of failure of power to any car (such as would be caused by passing over a dead section of rail), this relay is de-energized and causes the control circuits on that car to be thrown back to the series position with resistance in circuit. When power is re-applied the control progresses step by step to its former advanced position.

Train-wire Circuits. There are five circuits leading from the master controller, and five corresponding train wires. The five circuits comprise one for forward direction, one for reverse, one each for series and parallel, and the fifth for controlling the acceleration. When the master controller is moved to its first forward point a direction wire is energized which throws the reverser to its forward position, if it is not already so thrown. The reverser is electrically interlocked so that it cannot be manipulated when the motors are taking current. When the reverser has moved to the proper position, connections are made by it from the direction wire, through the forward reverser-operating coil, to the coils of the contactors which control the main or trolley leads to the motors.

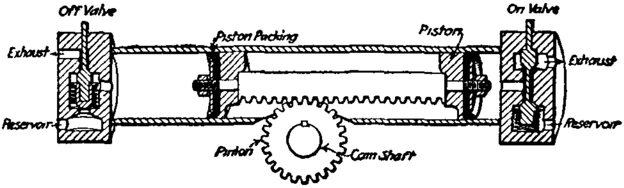


Fig. 20.—Type PC main engine and magnet valves.

General Electric Cam Control. This is a later design of multiple-unit control than the contactor type and is a more direct application of the principle of the drum controllers. In certain features it resembles both the drum controllers and also the contactor control. That is, instead of the main controller shaft of the drum controller with segments mounted thereon and turned by hand, there is substituted in the new control a cam shaft which is operated by means of compressed air. (See Fig. 20.) This cam shaft operates the contactors by means of cams. It has been possible with this design to assemble the contactors, reverser, line breaker relays, etc., in one box beneath the car body. The electric control circuits are greatly simplified and positive serial action of the contactors has been obtained without electrical interlocks. It also weighs less than the older type of contactor control.

Air-operated Line Circuit Breaker and Reverser. The line breaker and reverser are provided with individual air cylinders and magnet valves. These magnet valves control the air inlets and outlets of cylinders which in turn close the line breaker and operate the reverser. The operation of these valves is governed through train wires which are energized through the master controller.

Air-operated Cam Shaft. The contactors instead of having individual magnet valves and air cylinders similar to the line breaker and reverser are all operated by means of cams mounted on a horizontal shaft which carries a pinion at one end. This pinion

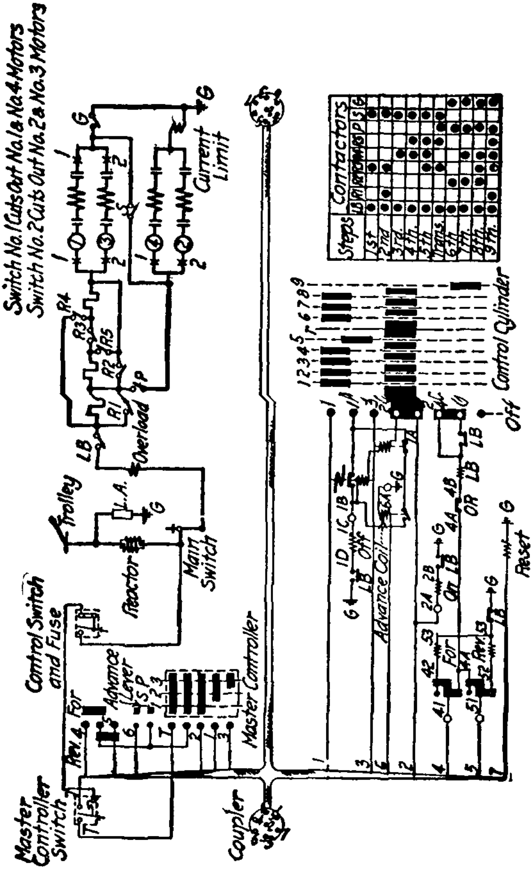


FIG. 21.—General Electric type PC-5 control.

engages with a rack as shown in Fig. 20, which is moved back and forth by a double-acting air piston. Air is admitted or released from either end of the air cylinder by means of magnet valves which are energized through the master controller and train circuit.

Controller Air Cylinder. (See Fig. 20.) The ends of the controller air cylinder are fitted with magnet valves and are designated as the "on" and "off" ends. The "on" end is the end of the cylinder at which air is admitted to turn the controller "on" and likewise the "off" end is the end at which air is admitted to turn the controller off. When the magnet valves are de-energized the normal position of the "on" valve is closed to the reservoir and open to the exhaust. The normal position of the "off" valve when the magnet valve is de-energized is just the reverse, closed to the exhaust and open to the reservoir. When the car is stopped the magnet valves are both de-energized, that is, air at reservoir pressure is admitted at the "off" end, and the "on" end is open to the exhaust; thus the piston is forced in the direction which turns the motor controller "off," *i.e.*, toward the "on" end. When the master controller is turned on and the reverser throws, the line breaker closes and both the "on" and "off" magnetic valves are energized, that is, air is admitted to the "on" side of the piston and exhausted from the "off" side and the rack revolves the cam shaft making the first motor connections. The piston will continue to move in the same direction if both magnetic valves are kept energized, but if the current flowing in the main motor circuit should exceed a predetermined amount it will operate a relay which de-energizes the "off" magnet valve, thus closing the exhaust port and admitting air from the main reservoir, which in turn will equalize the pressure on both sides of the piston and stop its movement. When the current in the main circuit falls below this predetermined value the "off" magnet valve is again energized and the motor controller moves on to the next point.

Master Controller. There are three points on the master controller; the first point gives the first point series, the second all series positions and the third all parallel positions. There are five steps in series on the motor controller, all of which except the first are obtained automatically by moving the master controller to the second notch. Likewise there are four steps in parallel on the motor controller, all of which are obtained by moving the master controller to the third point. The progression from one point to the next will only take place as fast as the notching relay, which energizes the "off" magnet valve, operates.

Advance Lever. A third handle is located on the master controller cap plate. This handle is known as the advance lever and is used to advance the motor controller point by point if in emergencies the car does not accelerate at the current for which the notching relay is set.

Westinghouse Power-operated Control

Classification According to Source of Current Supply for Valve Magnets. In the Westinghouse power-operated or multiple-unit control the two classes (automatic acceleration and hand-operated acceleration) each may be again divided according to the source of current supply for the valve magnets. That is, the source of this current may be the line or a storage battery.

Hand-controlled Acceleration Current for Control Circuits Taken from Line. (See Fig. 22.) The system having hand-controlled acceleration and taking current for the control circuits from

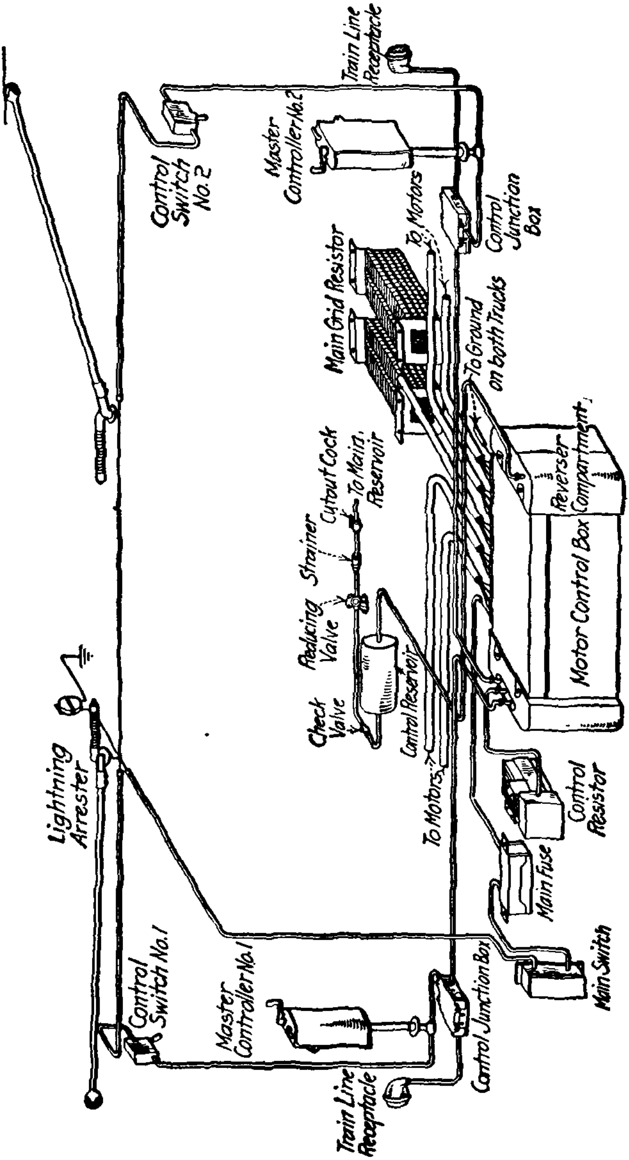


FIG. 22.—Typical layout of electro-pneumatic control.

the line is the simplest type and is in most general use on interurban and city surface lines for both train and single car operation. It is known as type HL. It is in effect the same as the hand-operated drum type of control, except that instead of combining the various circuit-breaking contacts upon a single drum, operated by hand, it divides these contacts into separate contactors and actuates them by means of compressed air taken from the air brake reservoirs and controlled by magnet valves and train-line wires from a small master controller. The system is made up of two general parts, namely, the auxiliary control system and the motor control system.

Auxiliary Control System. The auxiliary control system comprises a set of train wires, a control resistance, electrically operated valves operating the pneumatic unit switches, a master controller and auxiliary protective devices. Current for the control circuits is taken directly from the trolley or third rail through the master controller to the control resistance, the latter being arranged with low voltage taps for the valve magnet circuits.

Master Controller. The master controller is similar in design to the drum controller in that it contains a movable cylinder and stationary contact fingers are used to supply current in proper sequence to the different wires of the train line for energizing the operating coils of the motor control. The value of the current required is very small, not exceeding 2 amperes for each car in the train. The master controller is provided with two handles, one for operating and one for reversing the direction of train movement.

Reverse Handle. The reverse handle of the master controller is connected to a separate drum which establishes control connections for throwing the electro-pneumatically operated reverser to either the forward or reverse position when the main master controller handle is moved to the first accelerating notch. The control circuits of the reverser are so interlocked that unless the reverser itself corresponds to the direction of the movement indicated by the position of the reverse handle of the master controller, the unit switches on that car cannot be energized.

Control Coupler. Train-line jumpers between cars are so designed as to give a corresponding connection of train wires, this being secured by proper mechanical design of receptacles and jumper heads so that it is impossible to improperly insert the jumper in the receptacle.

Contactors or "Unit Switches." (See Fig. 23.) These are placed in a group, usually being assembled on a common frame and placed in a box beneath the car. The unit switch constitutes a circuit-breaker having a fixed and movable contact and provided with a magnetic blowout, the movable finger being actuated by an air cylinder energized from the brake reservoir and controlled by a magnet valve connected electrically to the train wire system. The switch finger is normally held open by a spring, which is compressed on closing the switch. The high pressure used in closing the switch is made use of to reduce the contact resistance, thus reducing heating and the size of the switch contacts.

Reverser. The reverse switch or reverser consists of an insulated drum carrying two sets of contacts arranged to make contact with stationary fingers when operated forward and backward by a pair of pneumatically operated pistons actuated through electrically operated valves by the auxiliary or master controller. No magnetic blowout is required as the reverser does not operate when carrying current on account of a mechanical interlocking between the main and reverse drums of the master controller. This reverser switch usually is mounted at one end of the switch group, but in some installations it is enclosed in a separate sheet-iron box.

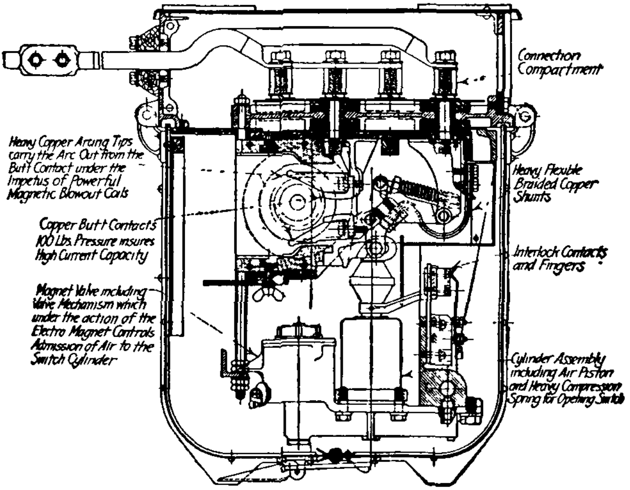


FIG. 23.—Electro-pneumatic switch unit.

Overload Trip. An overload trip is mounted on one end of the box containing the unit switches. It is actuated by a coil in the motor circuit and is arranged to open the control circuits of several switches ahead of the motors and main circuit resistance when an excessive overload on the motors occurs.

Control and Reset Switch. The control and reset switch is a small single-pole double-throw canopy switch used for disconnecting the control circuits from the line and for closing the control circuit to the reset coil of the overload trip, which thus can be reset from the car platform.

Motor Cut-out Switches. The motor cut-out switches consist of knife switches arranged in a manner similar to those on a hand or drum controller and are mounted in a box on one end of the switch group.

Control Resistance. The control resistance consists of a number of flattened tubes wound with resistance ribbon and supported in an iron-clad frame with all live parts protected from the weather.

The several tubes are connected in series for the full trolley voltage, and low-voltage taps are taken off for the magnet valve circuits. Fig. 24 shows the main and control circuits for a four-motor equipment of 60-h.p. motors with HL control, using a ten-switch group.

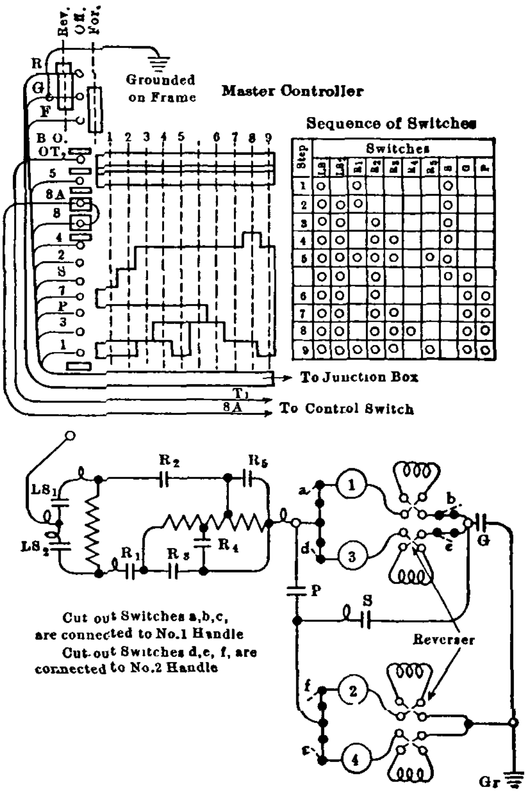


FIG. 24.—Scheme of circuits, Westinghouse hand-operated multiple-unit control, line current operation, four motors.

Westinghouse Automatic Control. Westinghouse automatic multiple-unit control differs from the hand operated type in that the acceleration of the motors is made on a predetermined current which is governed by a secondary master controller (called a sequence switch) and the setting of a current limit relay in the motor circuits, instead of being entirely under the control of the operator. A typical schematic diagram, shown by Fig. 25, is for heavy duty with two 190-h.p. motors in subway service.

Master Controller. The master controller is essentially the same as for HL control except that it has fewer notches and is arranged with a safety device to set the brakes in case of possible physical failure of the motorman. The emergency or "dead man's" application can be secured pneumatically, or electrically if desired when electro-pneumatic air brakes are used. The main handle of the master controller has an upward movement whenever the motorman's hand is removed; this releases the fingers on the upper positions of the reverse drum and cuts power off the control supply to certain switches and further releases a pin valve, which causes the

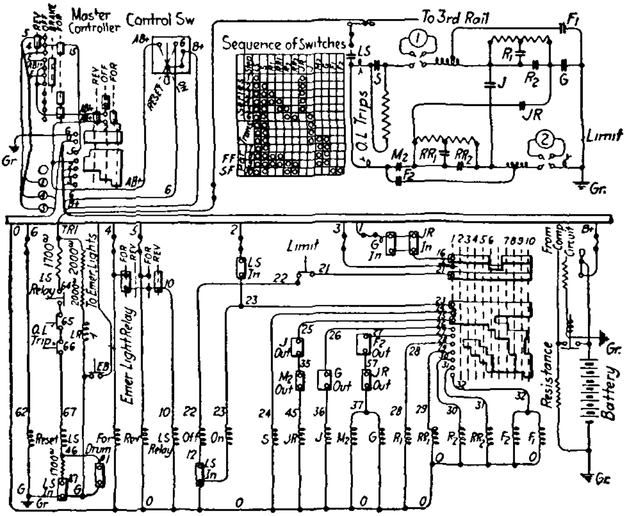


FIG. 25.—Main and control circuits, Westinghouse battery operated automatic electric-pneumatic control for two 190 h.p. motors, train operation.

air brake train pipe to be vented and brakes to be applied. The master controller has three working notches and an off position. The scheme of operation consists of the following progression: the reverse handle is first moved to correspond to the desired direction of the train; the main handle is then depressed and moved from the off to the succeeding notches to give the speed desired. With the main handle depressed, the off position serves as a coasting position of the controller, as it then opens all motor switches but does not apply the brakes. The first or switching position establishes connection with the train line so that the reverser is operated to the desired position, and the unit switch group closes the circuit to the motors with all resistance in. The second or full series position of the master controller handle introduces the current-limiting relay which permits the progressive picking up of the several unit switches as the current falls below the predetermined value. With *the controller on this notch, no further progression is made after* the current relay has permitted the closing of the proper unit switches to effect full series position connection of the motors with starting resistance entirely cut out. The third notch on the master controller establishes parallel connections of the motors by again bringing the current relay into activity, which permits the progressive cutting out of starting resistance until full parallel operation is attained, with full line voltage upon the motors.

In former designs, the master controller was constructed to have but one handle, which was notched to right or left of a central "off" position for forward or reverse running, respectively. The dead man's feature was then obtained by a spring which caused the handle to return to the central off position if the operator's hand was removed; this energized an emergency train brake magnet valve which set the brakes.

Unit Switches. The group of switches is similar to that used in the hand-operated type of multiple unit control, with the addition of a sub master controller (called a sequence switch), which is automatically advanced under the control of the limit relay to close the unit switches progressively so as to obtain a uniform acceleration at practically constant motor current.

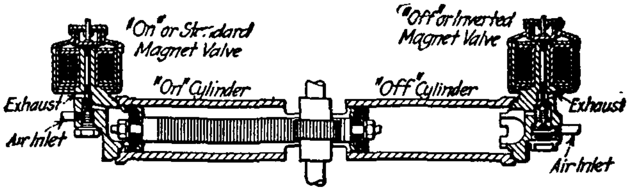


FIG. 26.—Cross-section of differential air cylinders.

The Sequence Switch is essentially an electrically controlled, pneumatically operated drum that carries contacts arranged for energizing the valve magnet coils of the switch group. It consists of two air cylinders with the pistons connected by a rack that transmits the movement of the pistons through bevel gears to the drum carrying the contacts. (See Fig. 26.) Compressed air is admitted or released from the cylinders by means of a magnet valve at each cylinder. The magnet valves are similar in general construction to those in the switch group, but the operation of the "off" valve is different in that when this valve coil is energized air is released from the cylinder, and when it is de-energized air is admitted. The "on" valve admits air to its cylinder when the coil is energized and releases air when it is de-energized. If both magnet valves are energized simultaneously, movement of the drum results, since air is admitted to one cylinder and released from the other. With neither magnet energized, the air in the "on" valve magnet cylinder is at atmospheric pressure, while that in the "off" valve magnet cylinder is at reservoir pressure, thus holding the pistons and drum in the "off" position. Energizing the "on" magnet balances the pressure in the two pistons but causes no movement. Thus, to move the piston and drum in successive steps, it is only necessary to energize the "on" magnet from the control supply circuit and the "off" magnet through the limit relay contacts. The sequence switch may be mounted directly on the switch group or apart from it.

The Limit Relay is of the solenoid type, with a coil connected in series with the main motors, in which operates a plunger that carries contacts for closing the control circuit to the sequence switch. The relay plunger lifts when the main motor current reaches a certain predetermined value and interrupts the control current to the "off" valve magnet, thus halting the progression of the sequence switch drum. As the motor speed increases, the current falls until a value is reached where the limit relay plunger again drops, closing the operating circuit to the sequence switch. This performance is repeated until the control has reached the notch indicated on the master controller.

Line Relay. The line relay consists of a coil connected between the line and the ground and provided with a plunger, to which is attached a contact disk. When line voltage is available, the coil of the line relay is energized, lifting the plunger and causing the contact disk to close a circuit across a pair of contacts. These contacts are directly in series with the auxiliary control circuit of certain contactors, and hence, if the line current fails for any cause, this auxiliary contact is opened, thus opening the main power circuits. Should the line switch be opened or should the current be cut off from the line for any reason, the plunger of the line relays drops, the auxiliary control circuit is opened and all the contactors are opened. When the contactors are opened by the temporary cutting off of the control current for any reason when the master controller handle is partially or wholly notched up, the limit relay governs the progressive reclosing of these switches.

Line-operated Automatic Control. For the line-operated automatic control, type AL, a control resistance is used to obtain a low voltage supply of control current. The control and reset switch and control resistance are the same as those used for the hand-operated control, type HL.

Battery Control. Battery control may be either the hand-operated or automatic type. The former is known as type HB and the latter as type AB. When a battery is used to supply the auxiliary control current for multiple unit control, the control resistance is replaced by a storage battery, supplying current at from 14 to 32 volts. On account of the low voltage used, a plug switch replaces the canopy type control and reset switch used on line current control equipments and the insulation of the entire auxiliary control circuit can be greatly reduced. Battery current for the auxiliary control circuits is particularly adaptable to high voltage systems, either alternating current or direct current, on account of the elimination of the high voltage from the master controller. This type of control is also employed on elevated and subway systems, and in trunk line service where close headway between trains is essential.

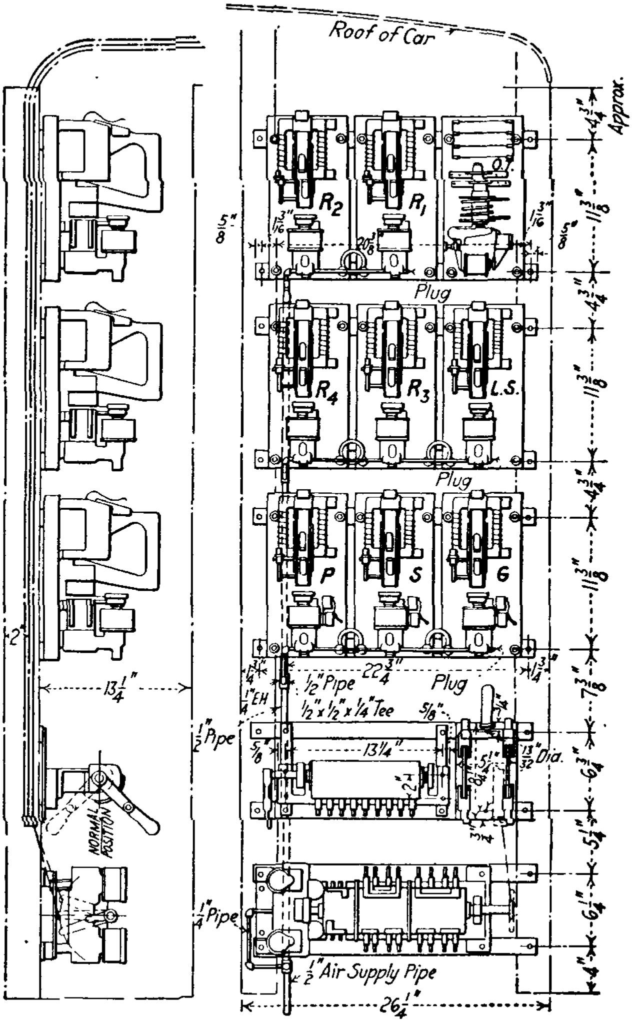


FIG. 27.—Cabinet control arranged for installation in car vestibule.

The use of a low voltage control which is independent of the line voltage eliminates various causes for incorrect operation, making this type of control particularly applicable for this class of service. The storage battery used for the control can also be used for emergency lighting, as well as for signal and brake systems. In large systems, where storage batteries can be economically maintained, the benefits of a low voltage control system are most evident. Unless it is desired to charge the storage battery from an outside source at the end of a run, it can be charged automatically from the line on a direct-current system or by a small motor-generator set on an alternating-current system, the amount of charge being regulated by means of an adjustable battery charging resistance and a battery charging relay.

Cabinet Control. Some engineers are of the opinion that the elements of a power-operated control system might be more advantageously arranged than in the group mounting as usually supplied by the manufacturers, both from the viewpoint of the space occupied and the place of installation, the latter especially considering ease of maintenance. There is an advantage in the usual group mounting in that the manufacturer properly assembles the various units and makes the various electrical interconnections, thus relieving the user of that responsibility. However, the proponents of the so-called "cabinet" mounting of control apparatus are willing to accept such responsibility in exchange for other advantages. The various switching elements of the control apparatus are shipped separately, and installed in a "cabinet" or compartment especially designed and built into the car body for the purpose. The cabinet may be built into the bulkhead back of the motorman's position, as suggested by Fig. 27 (see also Fig. 18, p. 506), or it may consist of special compartments under the car, arranged for the most economical use of the space there available, as well as for the most convenient inspection and maintenance. It has been suggested that the space beneath longitudinal seats might be used for all control switching apparatus except the line switch which breaks main circuits, which probably should be kept in its usual location under the car.

Single-phase Series Motor Control

The single-phase series motor (see pages 231 to 238) is controlled by adjusting the electromotive force applied to the motor. The three methods of making this adjustment are: (1) By resistance in series as in ordinary direct-current series motor control. (2) By autotransformer. (3) By induction regulator. The latter two are more efficient than the first because by their use the heavy resistance losses which would occur in the resistors by the first-named method are saved. The second is a widely used method and is generally known as the tap potential control (see Fig. 28) and the function of

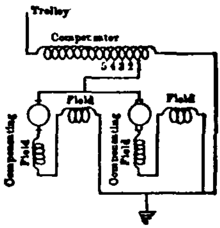


FIG. 28.—Tap control for a.c. motors.

the control, whether it be hand-operated K type or power-operated, is to successively connect the motor terminals to transformer taps of increasing potential while starting.

Auto-transformer. The auto-transformer or single-coil step-down transformer is wound for the trolley potential. Upon its grounded side there are several taps brought out to facilitate starting. It is customary to connect two motors in series when operating alternating current in order to approximate 500 volts input and reduce the size of the contact surface required in contactors. Owing to the fact that the alternating-current motor characteristic is more drooping than that of the direct-current series motor, fewer starting points are required for alternating-current control. As each point on the controller with potential tap control constitutes a running point at full efficiency, it is not necessary to use series-parallel connection of motors as is done with direct-current motors. If while passing from one compensator tap to the next without opening the circuit the gap were bridged by a connection of very low impedance, a heavy current would flow in the portion of the compensator winding thus short-circuited. To hold this current at a low value an inductance or a non-inductive resistance (termed, in this application, preventive inductance and preventive resistance, respectively) is placed in the circuit. Fig. 29 shows the application of the inductance (preventive coil).

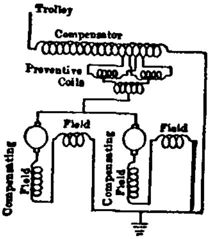


FIG. 29.—Tap control for a.c. motors, with preventive coils.

Reversing. Reversal is effected by reversing the series field windings as in direct-current control.

Hand-operated Control. The hand-operated control is identical with the type K controller used with direct-current motors, except that advantage is taken of the fact that alternating-current arcs of considerable size can be broken in the air without the aid of the magnetic blowout. Where cars are to be operated from both alternating current and direct current trolley with the same equipment, the hand-operated control is provided with magnetic blowout for direct current running, using the same controller without the magnetic blowout for alternating current running, unless the motor equipment be of large capacity.

Power-operated Control. The master controller is similar to the direct current master controller. Its office is to energize the train wires in proper sequence. Current of proper potential is obtained from transformer taps. The contactor is similar to the direct current contactor, except that it is designed for operation on alternating current and therefore has its magnetic parts laminated. The contactors connect the motors with transformer taps or with starting resistance grids, depending upon the character of the supply current.

Combined Alternating and Direct Current Control. (Fig. 30). Owing to the fact that alternating current series motors are wound for potentials of approximately 225 volts per motor, and that such motors in order to meet the exactions of commercial operation must in some cases also be adapted to run with 600 volts direct current, some additional features in control are required to perform this double service of alternating current and direct current control. When both alternating current and direct current running are to be accomplished with the same control it is necessary to

Sequence of Switches D.C. Operation.

Step	LS1	LS2	M1	M2	S1	S2	P	J	JR	K1	K	K2	K3	K4	RR1	RR2	RR3	RR4	RR5	G
Off																				
1	O		O						O											
2	O		O		O				O											
3	O		O		O	O			O											
4	O		O		O	O			O	O										
5	O		O		O	O			O	O					O					
6	O		O		O	O			O	O	O				O					
7	O		O		O	O			O	O	O				O	O				
8	O		O		O	O			O		O	O			O	O				
9	O		O		O	O			O		O	O				O	O			
10	O		O		O	O			O			O	O			O	O			
11	O		O		O	O			O			O	O				O	O		
12	O		O		O	O			O				O	O			O	O		
13	O		O		O	O		O					O					O		
14	O		O		O	O		O										O		
15	O		O		O	O		O												
16	O	O	O	O	O	O														O
17	O	O	O	O	O	O														O
18	O	O	O	O	O	O				O					O					O
19	O	O	O	O	O	O				O	O				O	O				O
20	O	O	O	O	O	O				O	O	O			O	O	O			O
21	O	O	O	O	O	O				O	O	O	O		O	O	O	O		O
22	O	O	O	O	O	O							O	O				O	O	O

Sequence of Switches A.C. Operation.

Step	CB	M1	P	J	S1	S2	RX	O	I	II	III	IV	V	VI	VII	VIII	IX	X	XI	XII
Off	O																			
1	O	O	O		O	O														
2	O	O	O		O	O														
3	O	O	O		O	O														
4	O	O	O		O	O			O											
5	O	O	O		O	O			O	O										
6	O	O	O		O	O			O	O	O									
7	O	O	O		O	O			O	O	O	O								
8	O	O	O		O	O				O	O	O	O							
9	O	O	O		O	O					O	O	O	O						
10	O	O	O		O	O						O	O	O	O					
11	O	O	O		O	O							O	O	O	O				
12	O	O	O		O	O								O	O	O	O			
13	O	O	O		O	O									O	O	O	O		
14	O	O	O		O	O										O	O	O	O	
15	O	O	O		O	O											O	O	O	O

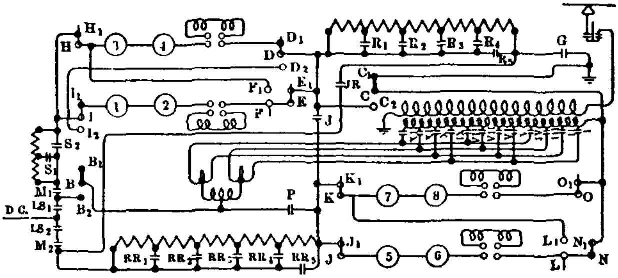


FIG. 30.—Circuits and sequence of switches, New Haven twin-motor locomotive (eight motors), alternating and direct current operation.

effect the following changes when changing from alternating current to direct current and *vice versa.*

Change main line fuses or circuit breakers.
Change lightning arresters.
Change motor field winding connections.
Change transformer taps to resistance taps.
Incidental changes in car wiring connections.

To effect these changes with a minimum amount of delay, all the necessary contacts are concentrated upon one cylinder in an auxiliary controller, so that a single movement will effect all changes simultaneously.

Retaining Coil. In changing from alternating current to direct current trolley and *vice versa*, it is necessary to guard against the possibility of wrong connections upon the car for the current received, that is, to prevent disaster should connections be made for 600-volt direct current operation and accidental contact be made with high voltage alternating trolley. To guard against this, the main switch of the direct current, alternating current car equipment is provided with a retaining coil so designed that it will open when the motor current is interrupted. Where alternating current and direct current trolley sections adjoin, a dead section may be left between the two for a length not exceeding a car length, so that a car may pass from one section to the other at full speed, in which case the main car switch opens on the dead section through lack of power to operate the retaining coil, and will reset automatically for alternating current or direct current operation as the case may be, after leaving the dead section.

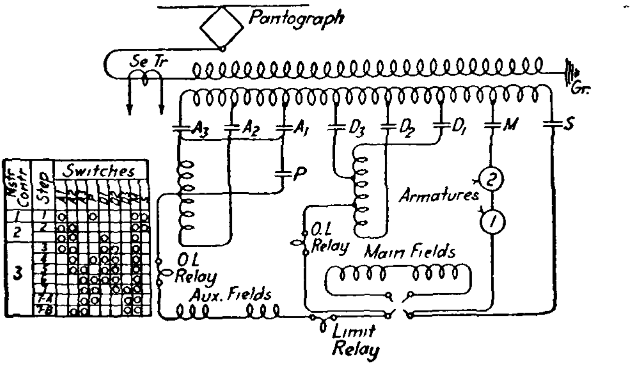


FIG. 31.—Control circuits for doubly-fed motors, Paoli Electrification, Philadelphia.

Connections to Air Compressor. The air compressor is geared to a single-phase motor wound for operation with 500 volts alternating current or direct current, and it is customary to wind the motor fields in two parts connected in multiple for alternating current and in series for direct current running, these changes being effected through the medium of the commutating switch.

Control of Doubly-fed A.C. Series Motors. The doubly-fed series alternating current motor is described on page 234. The circuits of a typical control system for two such motors are shown by Fig. 31.

Induction Regulator. Due to great weight, low power factor and lack of simplicity as compared with the tap potential method, the induction regulator method is at a disadvantage in controlling alternating current traction motors. The induction regulator is a special type of transformer, so arranged that one winding may be adjusted relative to the other, mechanically. It is commonly built like an induction motor having a coil-wound rotor and a very short air gap which is permitted since there is little motion and negligible bearing wear. The rotor winding is generally the primary. The primary is connected across suitable taps of the transformer, and the secondary, in series with the motor, is also connected across suitable transformer taps (see Fig. 32). The electromotive force induced in the secondary of the regulator due to the current in the primary may be adjusted by turning the coils relative to each other, and it may be made to buck or boost the electromotive force applied to the motor directly from the transformer by any amount within the range of the regulator. Thus the net electromotive force applied at the motor terminals is adjusted. On the single-phase locomotive of the Dessau-Bitterfeld line of the Prussian-Hessian State Railway gradual change of voltage is obtained by means of an induction regulator which covers a range of one step on the main transformer. The first position of this regulator lowers the transformer voltage by one-half step. On turning the induction regulator the voltage reduction gradually drops to zero; that is to say, the resultant voltage is the mean value between the two successive transformer taps. As the turning is continued the regulator delivers a rising additional voltage which gradually reaches the value of one-half step and thus equals the next higher step on the transformer. In this way the voltage changes from step to step are made without shock. The switching from one step of a transformer to the next is automatically obtained by means of a switch which is coupled to the induction regulator. The small single-phase motor which is used for turning the induction regulator is cut out automatically as soon as the voltage of the traction motor has reached its highest or lowest value as the case may be. This small motor and the various contactors in the motor circuit are operated by low voltage current taken from the main transformer.

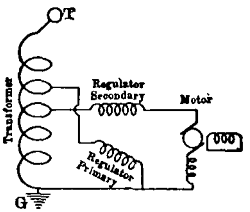


FIG. 32.—Induction regulator control.

Three-phase Induction Motor Control

The three general methods of controlling the speed of a three-phase induction motor are (1) rheostatic; (2) changing the number of primary poles; (3) cascade operation. The first method is used in connection with the other two, at least during the straight line acceleration period.

Reversing. A three-phase induction motor may be reversed by interchanging two of the three leads which supply the energy to the primary windings of the motor.

Rheostatic Control. In this method the proper tractive effort during the straight line acceleration period is secured by adding resistance to the secondary winding of the induction motor. For railway work the windings of the secondary are brought out to slip rings and the added resistance in the form of grids or liquid is external to the motor circuit. The grid resistance is cut out in steps as the motor speeds up and is short-circuited at the last step. Where liquid resistance is used, its value is reduced steadily, thus securing a steady acceleration. The electrodes which dip into the liquid are held stationary and in the process of reducing the resistance, the level of the liquid is raised, the influx being regulated by a valve controlled by the engineman. Ordinarily the secondary winding of the motor is three-phase; the external control resistance is star connected and is customarily kept balanced throughout adjustment. With a view, however, to simplifying apparatus this external resistance is sometimes unbalanced. Such unbalancing is used in the control of the four motors which drive each of the 115-ton locomotives operating through the Cascade Mountain tunnel on the Great Northern Railway. Each motor is three-phase, 25-cycle, 500-volt, 475-h.p. with a gear ratio 4.26 driving 60-in. wheels. The following, relative to this particular installation, is from a paper by Dr. Cary T. Hutchinson, A.I.E.E., 1909: "The control system of each motor is separate; the circuits branch from the transformer and are independent through the resistance. There are 14 contactors in each motor circuit, 56 in all. . . . . Iron grid resistances are provided for each motor; there are thirteen steps in the control, but in order to reduce the number of contactors to the lowest possible point an unsymmetrical system is used. A change is made in the resistance of one phase only, in passing from step to step. This arrangement in effect treats the three-phase circuit as a single-phase circuit; on each step of the control the torque is the average of the three values of the torque of the separate circuits. The principal advantage of this is that 56 contactors do the work that would otherwise require 128, thus effecting a great gain in the simplicity of the control apparatus."

Because of the resistance loss in the added external resistance in the secondary circuit the rheostatic method of induction motor control is inefficient. The net mechanical energy output of the motor is equal to the energy input to the secondary of the motor, diminished by this secondary copper loss at that instant.

Changing the Number of Primary Poles. (See page 239 for method of determining synchronous speed.) A motor may be arranged so that the number of its primary poles may be changed, thus giving more than one efficient running speed. The number of poles may be changed by re-grouping the coils of the motor primary winding or by using an independent primary winding for each number of poles desired. The re-grouping method is generally preferable because in its application all the motor primary windings are used.

Cascade Operation (Also referred to as **Tandem Control** or **Concatenation**). A great many variations are possible in concatenation. In railway work two motors are used. Two motors may be operated in direct concatenation to give half speed and then operated in parallel to give full speed and the advantages thus gained will be analogous to those gained in series-parallel control of direct current motors. Concerning concatenation the following useful comparison is drawn by B. G. Lamme: "This corresponds to two or more direct current shunt motors with their armatures connected in series. If both armatures are wound for equal speeds, then the tandem or series connections give half speed, as with the alternating current motor. If one direct current armature is wound for a higher speed than the other, which corresponds to two alternating current motors with different numbers of poles, then four speeds may be obtained corresponding to the windings connected cumulatively, differentially and to each motor used separately. This corresponds to the four speeds of the alternating current combination where two motors have different numbers of poles and are connected in tandem to give speeds corresponding to the sum or difference of the number of poles or are operated separately."

For concatenation of two motors for railway service, the rotors of the motors are mechanically connected, the primary of the first motor is connected to the supply, the secondary of the first motor is connected to the primary of the second motor and secondary of the second motor is connected to an external resistance at start. As the concatenated set speeds up the resistance in the secondary of the second motor is reduced and finally this secondary is short-circuited just as in the rheostatic method of control outlined above. If the motors have the same number of poles and are operated in direct concatenation they may, after having reached their normal maximum speed (synchronous speed of the set minus the slip) be separated and each, having resistance inserted in its secondary, may have its primary connected to the supply. From this point the resistance in the secondary circuits may be reduced as in the ordinary rheostatic method of control outlined above, finally leaving the secondaries short-circuited and the motors operating in full parallel on the line. Two running speeds using both motors are thus secured.

Cascade-single Control. This method is similar in operation to the method just outlined except that the second motor (termed the auxiliary in this use) is cut out entirely beyond the period of concatenation.

Cascade-parallel-single Control. This method is between the two previous methods (parallel and cascade-single). The motors have a different number of poles or the same number of poles with different gear ratios. This method is similar in operation to the parallel method except that during acceleration beyond concatenation the motor with the greater number of poles reaches synchronism and is cut out before the motor with the fewer poles reaches its maximum running speed. If the motor having the fewer poles be cut out when the one having the greater number of poles approaches synchronism the train will operate at a running speed between that for the set in concatenation and that for the motor having the fewer poles when running free with its secondary short-circuited.

Combination of Concatenation and Changing the Number of Poles. If several running speeds are desired this may be accomplished by changing the number of poles in one or both motors in the manner previously outlined and then proceeding with one of the methods of concatenation.

Synchronous Speed of Two Concatenated Motors; *Motors in Direct Concatenation.* Two concatenated motors are said to be in direct concatenation when they are so connected that they tend to start in the same direction. The synchronous speed of a set so connected is given by:

$$S = \frac{f \times 120}{p_1 + p_2}$$

in which S = synchronous speed, revolutions per minute
f = frequency of supply, cycles per second
p_1 and p_2 = the number of poles per motor, respectively.

Motors in Differential Concatenation. Two concatenated motors are said to be in differential concatenation when they are so connected that they tend to start in opposite directions. The synchronous speed of a set so connected is given by:

$$S = \frac{f \times 120}{p_1 - p_2}$$

OVERHAULING AND INSPECTION OF CONTROL APPARATUS

(A.E.R.E.A. Miscellaneous Methods and Practices)

Type K Controller. The equipment should be given a thorough *overhauling for every 60,000 miles of service as follows:* The controller should be taken apart, thoroughly cleaned, defective parts replaced, wood scraped and shellaced and other parts of the controller painted with insulating paint. The controller should then be given a break-down test of not less than 1500 volts alternating current for 5 seconds. With controllers in good condition, periodical inspections on a basis of from 300 to 500 miles service—depending upon the conditions of operation—will be sufficient and the most economical. Careful attention should be given to the fit of the controller and reverse handles on spindles.

Inspection of GE Type M Multiple-unit Control Apparatus. *Master Controller.* Examine the master control fingers, trying tension of fingers and polishing cylinder and fingers to secure good contact. Scrape carbon dust from arc-deflector division plates. Wipe dust off inside of cover. Examine controller throughout for loose connections and see that each finger is adjusted to make good contact. Lubricate fingers and cylinder with small amount of vaseline. Press down button on controller handle to see that the auxiliary fingers make proper contact, and polish up the same when necessary. Oil sparingly the shaft bearings of cylinder and reverser switch. Wipe off excess lubricant, especially around the blow-out coil. When finished, try on each point to see if contactors pick up properly, also throw reverser two or three times. Blow out controller with air hose.

Contactors. Scrape arc chutes clean of carbon and copper dust and examine the tips, filing up any that are blistered to make sure of good contact on all. Press up under contactor fingers to see if bottom tip wipes or travels on top tip at least one-eighth of an inch. If tip is worn thin or so as to give less than one-eighth of an inch wipe renew the tip. Try all screws which secure the tips to the contactor arms, making sure that all screws are absolutely tight. Examine for loose connections and try all screws and bolts to make sure that they are all tight. A long screw-driver arranged to turn with a wrench at bottom is desirable for this purpose. Inspect interlock fingers and polish up tips to insure good contact. If interlocks are bent out of shape, bend back into such shape as to make proper contact. Examine all shunts to see that none are partly broken or any in such condition that they will not have full carrying capacity, thus preventing danger of burning the hinge pins. See that contactors will lift sharply on each point and drop freely when controller is shut off. Before shutting the contactor box take air hose and blow out the contactors and box thoroughly clean.

Reversers. Try all screws to finger and connections to see that they are tight. Examine all fingers, filing fingers and contacts to insure a good contact. Feel tension on each finger, and renew if finger has been heated and softened. Make sure that each finger will make square and firm contact with segments. Wipe off all segment blocks, and if there is any indication of shorting across between segments remove segment block and replace with one that has been cleaned up and varnished. Throw rocker by hand a few times to see that the fingers will not catch. Throw reverser electrically a few times to see that it throws with strength and promptness without undue arcing and without rebound. When reverse finger is not making good contact it sometimes can be detected by tapping the finger with a screw-driver handle or by placing a thin strip of paper between the finger and the contact and noting the way the paper pulls.

Overhauling Type M Multiple-unit Control Apparatus. Overhauling should be on a 60,000-mile basis as follows:

All coils removed from the contactor, reverser and circuit-breaker; boxes thoroughly cleaned and painted with an insulating paint. Interior of boxes cleaned and painted; contact strips between coil frames inspected for loose contacts; all working parts thoroughly inspected and worn parts replaced when necessary; wires inside of contact box thoroughly painted and when reassembled given an insulation test of 1500 volts alternating current. It would then seem that periodical inspections on a basis of from 600 to 900 miles of service would be satisfactory.

The following points are suggested as requiring attention at such times. Examine for broken shunt straps and broken hinge pins. See that interlocks are properly adjusted and that small arcs do not form between the fingers and disks, thereby burning finger and disk, which would eventually cause a defective contact at this point and a dead car. Clean the disk and finger with fine emery cloth. Keep the arc chutes and plates clear of all copper caused by contactors breaking current. See that all connections are tight. See that springs are not broken and are in good order insuring good contact when closed. See that plungers do not bind and that contactors break free when the current is thrown off. Contact plates should not be worn so low that screws holding them are burned. Blow out contactor box with compressed air. Note condition of wiring in the box. Clean the master control cylinder and use a small amount of vaseline on the fingers. See that the handle is of proper fit and works perfectly free. The adjustment of controller should be looked after very carefully, as there are no adjustment screws on the contact fingers. Note condition of throttle. Clean throttle disks and fingers and see that adjustment nuts are not loose. Do not clean throttle plunger unless it shows signs of sluggishness. Great care must be taken when cleaning plunger. Clean reverser and note adjustment and condition of plates and fingers and that the reverse throws in properly. Use no oil or grease on contactor or reverser finger or plates. A great deal depends on the close adjustment of interlocks. All bearings on contactors and interlocks must be made loose. When a contactor box becomes coated inside with a yellow coating caused by the burning of copper, short circuits are very likely to occur if this is not cleaned off.

Overhauling Electropneumatic Control Apparatus. Overhauling should be on a 60,000-mile basis as follows:

Clean the drum and adjust fingers of master switch; inspect cab switch terminals and see that they are held rigidly and no strands of wire are broken. Repair, clean and carefully adjust line relay, limit switch and battery relay. Limit switch should be adjusted with ammeter. Take apart, clean, scrape and shellac drums of motor cut-out switch and reverser; replace any parts that will not make the mileage and adjust the finger tension. Strip switch groups of all magnets, switch arms and moving parts, replace worn parts when necessary. Replace worn or broken arc shields; adjust all magnet valves to operate at proper voltage; replace defective shunts; adjust and clean all interlocks and interlock fingers; examine all insulation and make as good as new; examine piston leathers and see that they are flexible and replace those badly worn. Storage batteries should be cleaned of sediment and acid strength adjusted. Grid diverters should be cleaned, the insulation renewed where necessary, and all connections tightened. Control jumpers should be tested by passing 7 amperes of current through them for 3 minutes, at the same time giving jumper the same motion that it has when in service. Clean and adjust circuit-breaker; thoroughly blow out all piping and air chambers connected with the control.

On **short period inspection** the following is the practice on a road having inspection periods based on a 600-mile service.

Master Switch. Clean and lubricate every tenth inspection.

Cab Switches. Inspect terminals every inspection day.

Close jaws of cab switch to fit tight each inspection day.

Line Relay and Limit Switches. Clean with crocus cloth each tenth inspection. Inspect connections each inspection day.

Motor Cut-off Switch and Reverser. Inspect finger tension and oil drum contactor each second inspection and feel the terminals to see if the wires are O.K.

Inspect interlocks each twelfth inspection.

Oil reverser switch toggle each tenth inspection.

Circuit-breaker and Switch Group. Clean armature and valves each tenth inspection. Inspect contacts each inspection day. Clean arc chutes each inspection day. Blow out with compressed air all switches and grid diverters each third inspection. Inspect all grid diverter connections and oil all pistons each inspection day, see that all terminals are tight and inspect wires. Wipe off insulators. Inspect shunts and battery connections each inspection day. Add distilled water to take care of evaporation when necessary. Test specific gravity each thirtieth inspection day. Test battery relay and inspect terminals of battery switches each inspection day.

Maintenance of Controller Fingers and Contacts. The Westinghouse company's instructions under this heading call attention to the importance both of lubrication and contact pressure, as follows:

Lubrication. Large capacity fingers (¾ and 1 in. in width), such as are used in the K type of controller, are made of copper and slide over copper contact segments. Both finger and segment, being of the same composition and comparatively soft, will wear excessively unless properly lubricated. The quality of lubricant used varies somewhat with the climate and temperatures, but vaseline will be satisfactory for summer and for moderate winters; engine oil is good for very cold winters. It is erroneous to use large quantities of lubricant with the idea that the more used the longer it will remain on the contact segments, as the surplus soon wipes or burns off and accumulates on arc barriers, fingers and drum castings, collecting copper dust and dirt, with a resulting tendency toward insulation failure. The contact segments also become sticky and dirty. The best practice is to spread the lubricant as smoothly over the segment as possible with a cloth, operate the fingers over the segment several times, and then wipe around the finger and the segments to remove any surplus. Where non-arcing duty is performed, considerably less lubricant can be used, and it should be of a lighter grade.

Contact segments and fingers become roughened by arcing and should be smoothed up carefully with emery cloth or a file before lubrication is applied. A wire-drawn contact surface should be carefully smoothed and wiped off. The safe mileage between inspections depends on local conditions, and should be determined by experiment for each service.

Small fingers ¼ in. and ⅜ in. wide, such as used in the control circuits of multiple-unit apparatus, are usually made of different material from the contact segments, thus causing less cutting. The arcing is very slight, so that the wear is much less, and the need for lubrication is not so urgent as on the copper fingers. However, the same general methods should be followed.

Contact Pressure. The safe current load on a finger depends on the width of the contact surface, the pressure at the point of contact, and the mass and radiation of the finger and segment. The capacity for a given width increases with the pressure, but too heavy pressure causes excessive wear and stiff drums. Average practicable finger pressure for general service (copper fingers on copper segments) are listed below. For different contact materials these values may be increased somewhat.

Size of finger	Pounds pressure
One inch	8
Three-fourths inch	6
One-half inch	4
One-fourth inch	2

By means of a small spring balance and a wire stirrup the pressure is easily checked, and inspectors soon become accustomed to the feel of a finger with correct pressure. Pressure is varied by changing the bend in the flat finger spring. After bending, see that the finger is making contact along its full width. Most fingers have an adjustable stop, which limits the drop of the finger tip when it leaves the contact, but this stop does not vary the finger pressure; its sole purpose is to prevent stubbing. The drop should be set at 1/16 in. to 1/8 in. or enough to allow the finger to lift entirely free from the stop when the finger is on the contact; this allows full pressure at the contact surface. The lift and pressure should be checked in all positions of the drum, as an eccentric drum or one having worn bearings and shaft may have good finger pressure in one position and weak pressure in another position.

The considerations mentioned above are equally important when installing new fingers or contact segments. A new finger should preferably be ground in with emery cloth to give a contact area at least 1/8 in. width along the contact line, and the finger should make contact over at least three-fourths of its breadth.

SECTION VI

CURRENT COLLECTING DEVICES

Trolley Wheels. The first essential of a good trolley wheel is that it should be in perfect balance, and run perfectly true. The necessity for this, as well as for good lubrication, is due to the location at the end of a long and flexible pole, and to the high speeds; at 30 miles per hour a 4¼ in. wheel (2¾ in. diameter at contact) runs at about 4250 revolutions per minute; at 60 miles per hour a 6 in. wheel (4½ in. diameter at contact) runs at about the same speed. As a simple test for balance on commercial wheels, it is suggested that wheels to be tested be mounted in harps and the wheel held against a rapidly moving belt. If the balance be only slightly imperfect, it will be found impossible to hold the wheel against the belt.

The two designs or types of trolley wheels in most general use are the alloy (Fig. 1) and the "Ideal" (Fig. 2). The alloy wheel is cast bronze, and should be of a composition to combine ample conductivity with sufficient hardness to insure maximum mileage and minimum wear on the trolley wire. Among the formulas which have been used by various companies are the following·

	Per cent		Per cent
Copper	90.0	Copper	91.0
Tin	8.0	Tin	7.0
Zinc	2.0	Zinc	1.5
		Flux	0.5

	Per cent		Per cent
Copper	91.38	Copper	96.0
Tin	6.5	Tin	3.0
Zinc	2.0	Phosphor tin	1.0
Lead	0.12		

The so-called "Ideal" trolley wheel is somewhat cheaper than the alloy wheel, and has given good service. It is built up with a forged copper center with steel flanges pressed on either side, as shown in Fig. 2.

For lubrication the bronze and graphite bushing is in almost universal use, often supplemented by an oil well in the hub, as shown in Fig. 1. In some cases, this oil well has been very considerably enlarged, as in the special type shown in Fig. 3.

Dimensions of various sizes of trolley wheels, as standardized by the manufacturers, are shown in the table on page 367. In general the smaller wheels are used in city service where the maximum speeds are comparatively low. Some years ago there appeared a distinct tendency toward large trolley wheels for heavy and high-speed service, as high as 10 in. diameter having been used in some instances. The great weight of such large wheels, the high base spring tension required, and the resulting likelihood of increased damage to overhead construction when the wheel left the wire, caused a reaction to the present tendency to limit the diameter to about 6 in.

The average mileage obtained from trolley wheels in city service, as reported by 22 companies to the 1922 Committee on Equipment, A.E R E A , was 8272 miles; in interurban service, as reported by 20 companies, 5035 miles. Of 21 companies reporting definite data

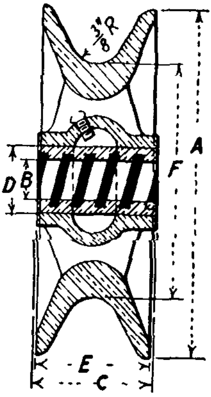


FIG. 1 —Standard alloy trolley wheel

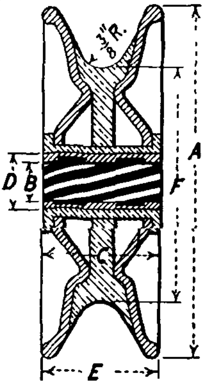


FIG. 2.—Ideal trolley wheel.

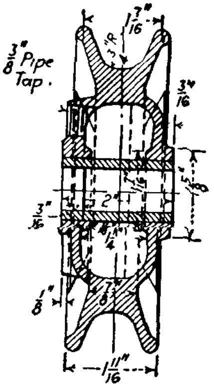


FIG 3 —Special form with large oil well.

on the life of trolley wheel bushings as compared to the life of the wheel, experience varied from 1/8 to 3 times the life of the wheel, the average life of bushing being the same as that of the wheel; eliminating four cases which seemed to be unusual (1/8, 2, 3, and 3), the average of the bushing lives reported by the remaining 17 companies was 75 per cent of the life of the wheel.

The Standard Handbook for Electrical Engineers gives the following for approximate life of trolley wheels:

City service 25 miles per hour maximum	11,000 miles
Suburban service 35 miles per hour maximum ...	6,000 miles
Interurban service 50 miles per hour maximum .	3,500 miles
High-speed service 60 miles per hour maximum	2,000 miles

DIMENSIONS OF TROLLEY WHEELS, FIGS. 1 AND 2

Form no	Type of wheel	Weight in lb	Dimensions, in inches					
			A Out-side diam.	*B* Diam axle pin	*C* Length hub	*D* Diam bush-ing	*E* Width at rim	*F* Diam. groove bottom
5	Alloy	2	4 1/4	1/2	1 1/2	7/8	1 3/8	2 3/4
6	Alloy	2	4 1/4	1/2	1 1/2	7/8	1 3/8	2 3/4
31	Alloy	2 1/4	4 1/4	1/2	1 1/2	7/8	1 3/8	2 3/8
32	Alloy	2 1/4	4 1/4	5/8	1 1/2	1	1 3/8	2 3/8
33	Alloy	2 3/4	4 1/2	1/2	1 1/2	7/8	1 7/16	3
40–A	Ideal	2 1/8	4 1/2	1/2	1 1/2	7/8	1 1/2	2 5/8
34	Alloy	2 1/2	4 1/2	5/8	1 1/2	1	1 7/16	3
40–B	Ideal	2 1/8	4 1/2	5/8	1 1/2	7/8	1 1/2	2 5/8
36	Alloy	2 3/4	5	1/2	1 1/2	7/8	1 7/16	3 1/2
37	Alloy	2 1/2	5	5/8	1 1/2	1	1 7/16	3 1/2
38	Alloy	3	5	5/8	2	1	1 7/16	3 1/2
61	Alloy	3	5	3/4	1 1/2	1 1/4	1 7/16	3 1/2
41	Alloy	3	5 1/2	1/2	1 1/2	7/8	1 7/16	4
42	Alloy	3	5 1/2	5/8	1 1/2	1	1 7/16	4
43	Alloy	3 1/4	5 1/2	5/8	2	1	1 7/16	4
21	Alloy	3 1/2	5 3/4	1/2	3	7/8	1 1/2	4 1/4
46	Alloy	3 1/2	6	1/2	1 1/2	7/8	1 7/16	4 1/2
·47	Alloy	3 3/4	6	5/8	1 1/2	1	1 7/16	4 1/2
60–A	Ideal	3 1/8	6	5/8	1 1/2	1	1 29/64	4 1/2
48	Alloy	3 1/2	6	5/8	2	1	1 7/16	4 1/2
60–B	Ideal	3 3/4	6	5/8	2	1	1 29/64	4 1/2
49	Alloy	3 3/4	6	1/2	3	7/8	1 7/16	4 1/2
63	Alloy	3 3/4	6	3/4	1 1/2	1 1/4	1 7/16	4 1/2

Trolley Harp Details. Trolley harps are of malleable iron, the weight being kept as low as possible consistent with strength and wearing qualities. Limiting dimensions depend upon those of the trolley wheel to be carried. In most cases the harp is riveted to a steel rod which is inserted into the pole and riveted fast. The trolley axle pin is of case-hardened cold rolled steel, and the contact springs and washers are of spring bronze. The current should be conveyed from the wheel to the harp through the springs and washers—not through the axle pin. It is most important, therefore, to see that contact washers and springs are functioning properly, and that the latter are securely riveted into good electrical contact with the harp. See Fig. 4 for details of one design

Current Carrying Capacity of Trolley Wheel. The following values of current carrying capacities of trolley wheels when traveling at various speeds are from the Standard Handbook for Electrical Engineers. The pressure between trolley wheel and trolley is from 20 to 40 lb.

Speed, miles per hr	5	10	20	30	40	50	60
Current capacity, amperes	1000	850	650	550	400	300	200

Trolley Wheel Defects. The following consideration of possible defects in trolley wheels is from a paper by C. W. Squier, *Electric Railway Journal:* The greatest trouble with trolley wheels arises from the difficulty of securing satisfactory lubrication and of conducting the current from the wheel to the harp. Some of the most frequent defects are as follows: sides bent, broken or chipped; double groove; holes in sides; flat spots; bushing hole too large; burned wheels; sides loose and rim worn off.

Bent, Broken or Chipped Sides. Bent, broken or chipped sides usually result from wheels coming off the wire and striking some

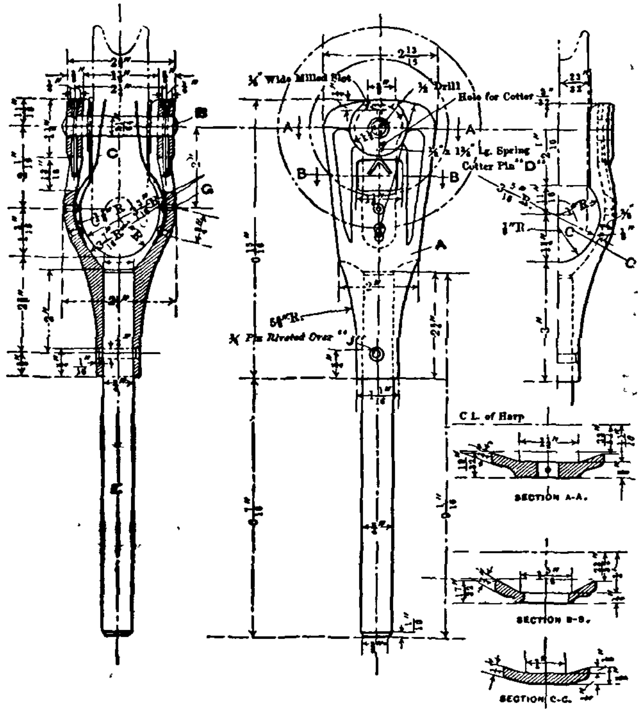


FIG. 4.—Sample trolley harp details.

part of the overhead construction. The number of such troubles can be reduced by proper attention to the lubrication of the trolley stands so as to insure their free swiveling, by making sure that the wheels stand perpendicular to the car roof so as to make proper contact with the trolley wire, by keeping the side bearings properly adjusted so as to prevent wear by the swaying of the car when operating at high speeds, by renewing the trolley bushings and axles before they become unevenly or excessively worn, and by keeping the trolley tension properly adjusted.

Double Grooves or Holes in Sides. Double grooves or holes in the sides arise from the condition that the wheel is not following the wire properly, and this may be due to improper maintenance of the overhead construction. These defects also occur frequently on cars that are operated from one end on lines with frequent curves, caused by the wire riding the flange while rounding a curve. If the harps are not straight in the poles so that the wheel is maintained at an angle to the wire, or if the side springs are weaker on one side of the car than on the other so that the car body does not rest level when loaded, the trolley wheels are liable to wear unevenly. When a wheel is found wearing to one side it can be made to wear straight in some cases by reversing it in the harp.

Flat Spots. Flat spots are caused by the sliding of the wheel on the wire on account of imperfect rotation. Flanges frequently become slightly bent and rub against the harp, or the side springs and washers may be too tight or exert too great pressure against the side of the wheel. Flat spots usually start with a very slight slippage between the wheel and the wire, but when a spot is once started it increases rapidly.

Bushing Hole Too Large. Oversized bushing holes are caused by lack of lubrication or by shunts and springs which have become so worn and loose that their pressure is too light to properly conduct the current to the harp. A slight burning or pitting results from the carriage of the excessive current through the bearing and axle. This causes the bushing to bind on the axle, and if it is not a very tight fit in the wheel, rotation will take place around the bushing instead of the axle. Under such circumstances only a few trips are required to wear out the bushing hole in the wheel.

Burned Wheels and Sleet Troubles. Burned wheels are caused when the wheel becomes separated short distances from the wire so that rapid arcing and destructive burning take place. Sleet destroys the wheel very rapidly in this manner. At other times it is found that if a wheel is out of balance its centrifugal force as it rotates will break the contact with the wire. The trolley tension on cars found with burned wheels should always be tested carefully. In sections where sleet storms are frequent, sleet cutters are used to advantage. On large systems, however, the removal of trolley wheels and the installation of sleet cutters assume enormous proportions and come as an additional task for the shop forces at the very time when they are usually busy with snow equipment. Large city roads usually prefer to depend on the frequency of the service to keep the sleet off the wire, or they install sleet cutters on but a few cars of each line, depending on these to scrape the ice clean and thus prevent excessive burning on the wheels of the other cars.

Sliding Contact Shoe. Several types of sliding contacts, designed to be substituted for the trolley wheel, have been developed. One of these, the Miller trolley shoe, is shown by Fig. 5. The forged steel shoe has its contact surface polished smooth and to about the same cross contour as the standard trolley wheels, but has a longitudinal curvature of much longer radius than the wheel, thus giving a greater area of contact than the wheel. The contact shoe is hung in the special harp and electrically connected to it by a copper shunt cable, as shown in Fig. 5. The sliding contact shoe eliminates trolley wheel bearing troubles, including lubricat on; it makes better contact with the trolley wire; it eliminates vibration due to wheel unbalancing, and therefore should reduce arcing and noise; it is claimed that its life is greater than the trolley wheel, and that by its use there is less total wear on trolley wire, due to reduction in wear due to arcing; one of the disadvantages noted is increased difficulty in backing cars. H. Savage, of the Detroit United Railway, reports life of the sliding contact shoe as about 2½ times that of the standard trolley wheel. He says that an examination of trolley wire where contact shoes had been used continuously for 23 months showed that the wear on one-way wire was very slight, the wire having a brownish color, slightly glazed; while on wire in two-way operation the wear was more noticeable, but not excessive. The brownish color and glaze do not appear on the two-way wire. He also found that on wire where wheels and slides were both used the wear was much more noticeable than if slides or wheels alone were used. This was true also for a short time after all trolley wheels were replaced by slides. It has been pointed out, in connection with trolley wire wear, that the sliding contact is likely to cause considerably more wire wear at low than at high speeds; this without doubt is partly due to the fact that the coefficient of friction is greater at low than at high speeds, as well as to the generally larger currents taken through the contacts at low speeds.

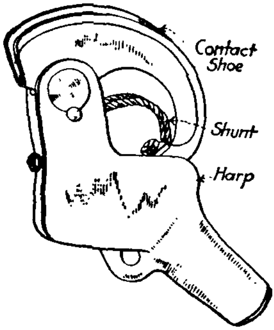


FIG. 5.—Miller trolley shoe and harp.

Trolley Base. The functions of a trolley base are to provide so flexible a support for the trolley pole that it will have free movement both laterally and vertically while exerting the pressure necessary to keep the trolley wheel in contact with the wire. A freely swiveling base is most essential in order that the trolley wheel may follow the overhead line. To provide this extreme sensitiveness, roller and ball-bearing bases have come into general use. Both types operate satisfactorily when provided with properly hardened parts. By using ball bearings a lower base can be obtained than with roller bearings, but roller bearings give greater wearing surfaces. Various pressure spring arrangements are in use. Some bases have springs in tension and others in compression, some bases have but a single spring, while others have a battery of springs. The following brief consideration of possible trolley base defects is from a paper by C. W. Squier, *Electric Railway Journal:*

Broken or Weak Springs. A spring which is in tension breaks most frequently at the end loops or at the bend where the loop joins the first turn of the spring. The number of such breakages can be reduced by keeping the parts over which the springs hook in good condition so that the loops have a maximum amount of bearing surface. By the use of a magnifying glass one may often discern small cracks in new springs at the bend from the loop to the first turn of the spring. These cracks are evidently due to the method of manufacture, and while such springs are strong enough to withstand heavy strains, they will ultimately break at these fissures. Springs are also weakened from the gradual loss of their ability to resist elongation or to the compression which follows the slow accumulation of a permanent set. In older types of bases the full power of the springs may not be available for producing tension at the trolley pole because of excessive friction in such parts as the cross-pin that forms the up-and-down bearing for the trolley pole, the cross-head in its guide, the side rod bearings which carry the trolley spring pressure and the trolley springs themselves on their guides. It is very difficult to keep these parts lubricated, as rain forces the oil out on the roof of the car and the windows and sills often become bespattered and soiled from oil. In the later designs of stands an effort has been made to provide bearings which do away with the necessity for constant oiling at friction points, and the designers have also endeavored to provide for a uniform pressure of the trolley wheel on the wire at different elevations. Tension springs are sometimes weakened by overstretching them during installation. One bad practice, for example, is to force a screw-driver or other sharp tool between the spirals of the spring and to use this as a lever to hook the eye over its post. This is likely to force the spirals apart to such a distance that they will not come back to their original position.

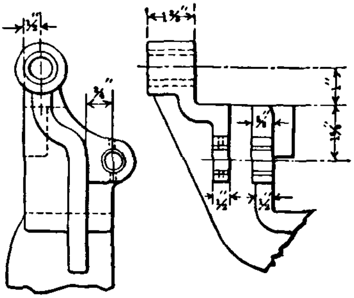


Fig. 6.—Bushings in trolley base yoke.

Worn Bearings and Pins. Worn bearings at various parts of the base are a constant source of trouble. In a great many cases the desire to keep down the weight has left insufficient material to permit the boring and bushing of worn bearings. Some roads have gone to the expense of making patterns of new castings which are provided with sufficient material to allow all wearing parts to be bushed. Fig. 6 illustrates one method of bushing a yoke to take care of excessive wear.

Nut, Bolt, Ball and Roller Troubles. All nuts should have lock washers, and if difficulty is then experienced, cotter keys should be added. Where ball bearings are used, the races are usually insulated from the socket casting to prevent them from carrying the trolley current. Notwithstanding this precaution, the frequent discovery of burned ball races and balls shows that they do carry current. This action may take place in several different ways. For instance the contact shunts that carry the current from the socket casting to the base plate occasionally get bent out of position so that they bear on the race and socket casting at the same time. Current then passes across the contact face of the shunt to the ball race and thence through the balls to the base plate, causing the burning of the balls and race. Moisture and dirt collect on the surface of the insulation, and the current then has an easy path to the ball race. Again, contact shunts are sometimes torn off entirely and current then passes over the surface of the insulation from the socket casting to the race. When new ball races are installed care should be taken to see that the lower edge of the ball race does not project beyond or come flush with the contact surface of the socket casting. This precaution will prevent the contact shunts from bearing on the race, for then they cannot touch the edge of the race while still remaining in contact with the face of the socket casting.

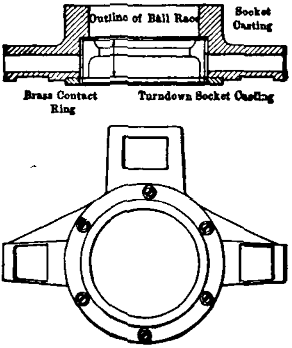


Fig. 7.—Trolley base with contact ring.

Burned or Broken Shunts and Burned-off Leads or Terminals. Of necessity, trolley bases must be made very low. As a result, the distance from the socket casting to the base plate is usually not more than 1 to 1¼ in. This, then, is all the space that is available for contact shunts. It is very difficult to get in this space a spring which is efficient enough to give the necessary current-transmitting pressure of the contact shunt against the socket casting. The springs soon take a permanent set and arcing then takes place between the contact surface of the socket casting and the shunt. As a result, the shunt is burned away or else the contact surface of the socket casting becomes so rough from the arcing that the shunts are torn off. The surface of the socket casting is thus destroyed, and in most cases it is necessary to install a new casting. Fig. 7 shows a brass contact ring which was made to screw to the face of the socket casting. As this ring is of brass, it forms a better conducting surface than the steel casting, and if it becomes burned it can easily be replaced at small expense. This ring also permits the re-use of socket castings that have become slightly burned, since the lower face of the casting can be turned off for the reception of the contact ring. The common form of terminal used on trolley bases consists of a hole in a lug to receive the lead. The hole is provided with set screws to clamp the lead in place. Leads become loose from the working out of the set screws or because the screws are not tightened when a new base is installed. If the screws stay in place, they soon become so rusty that it is almost impossible to remove them when it is necessary to install a new base. If the leads become loose, the arcing thereby initiated soon burns away the lead and terminal. A better form of terminal which can be quickly and easily removed consists of a standard soldered terminal such as is used for the ground leads on motors. If the trolley lead is soldered into this properly, there is no danger that the lead will burn off at this place. This terminal is then bolted to the base plate, where a spot-finished surface is provided to give a good contact surface.

Inspection and Lubrication of Trolley Wheel, Stand, Rope and Retriever. (*A.E.R.E.A. Miscellaneous Methods and Practices.*) Examine wheel and see that bushing and hub or spindle are not unduly worn and that outer rims are not bent, nicked or worn out; see that cotter keys holding spindle in harp are in good condition; see that contact springs and washers are sufficiently tight between harp and hub of wheel to form a good contact, but not so tight as to allow the wheel to slide on the wire. See that the spindle is tight enough in the harp to form a good contact, and that the spindle holes in the harp are not too badly worn to prevent this contact. Examine the harp and see that it is not loose on the pole and that rivets holding same are tight. Examine pole for cracks, bends or flaws, and see that it alines the wheel properly with the wire, and if not, loosen the clamp bolts and turn with a pipe wrench until wheel is in proper alinement, leaving the wheel on the wire during the operation. See that clamp bolt and nut holding same and base bolts are all tight and in good condition; examine springs and see that they have sufficient tension, allowing sufficient space between the coils for compression when the pole is pulled down to the roof of the car. In bases with more than one spring, see that they are equalized on each side, both sides given the same tension. Give springs sufficient tension so that the wheel will have pressure of about 20 to 25 lb. against the wire in city service and from 35 to 40 lb. on high speed. Make this test where the wire is of standard height. It can be done by using a hook scale, or by hanging an old brake-shoe or other weight to the trolley rope. See that trolley board is securely fastened to the roof of the car.

Trolley Lubrication. Lubricate trolley wheels at each inspection and wipe all surplus oil from wheel hub after lubricating. Lubricate bases when necessary. This can be determined by swinging the pole from side to side below the wire. If the base operates freely no lubrication is necessary. Great care must be taken not to allow any surplus oil or grease to reach car roof while lubricating the bases and wheels.

Trolley Rope. See that rope has a firm fastening with the harp; that it is not chafed or showing signs of wear where it comes in contact with the hood; that it has not been broken and that it has no unnecessary knots in it.

Trolley Retriever. Trip the retriever and see if it operates properly; that the tension in the retriever spring is not so severe as to break the rope or pull the trolley down so severely as to damage the hood or roof; see that rope works freely when resetting, and that it is of such length that it will not pull trolley from the wire where the wire is high, such as at railroad crossings, etc.

Extra Trolley Pole. See that all interurban cars are supplied with an extra pole, fully equipped, and in good condition.

Trolley Pole and Pressure between Trolley Wire and Trolley Wheel. Poles of such length that the longitudinal axis of the pole makes an angle of from 35 to 45 deg. with the axis of the trolley on tangent track are in most common use and the most common length is 12 ft. Height of car, height of trolley wire, alinement of track and other local conditions may require a longer or shorter pole between the limits 10 ft. and 16 ft. A light, flexible pole is preferable to a heavy one, as it adapts itself to irregularities in the overhead system and does comparatively little damage to that system when the trolley wheel jumps from the trolley wire. The pressure between trolley wire and trolley wheel should be such that the trolley wheel will follow the trolley wire with as little wear as possible. High-speed service requires a greater pressure than low-speed service. When the proper pressure has been decided upon, the adjustment should be made by measurement. This adjustment has been commonly made by hanging a weight at the end of the trolley pole or by fastening a spring balance to the trolley rope and applying the required tension through the spring balance, then adjusting the trolley stand spring till the wheel is of the proper height to just touch the trolley wire. Adjustment by use of the weight has been found to be the most satisfactory.

In operating practice, pressure between the trolley wheel and wire varies from 16 to 40 lb., pressures from 16 to 25 lb. being usually considered sufficient for city operation, while 35 lb. is generally used in high-speed interurban operation. As the alinement of trolley wire and condition of track approaches perfection, the trolley wheel pressure approaches the minimum, and no more pressure should be used than is required to hold the trolley on the wire at the required speeds with properly lubricated trolley bases.

Trolley Poles for Double Truck Cars. The general practice is to use two trolley poles on double truck cars when operated double-end, the trolley base usually being mounted directly over the center of the truck, although less offset in trolley wire is required when the trolley base is mounted between the truck center and the end of the car. With two trolley poles there is less likelihood of the trolley wheel being pulled off the wire from rope friction over the rear end of the car or a defective retriever; with two trolleys the car is not entirely disabled by the failure of one; trolley wheel replacements are made more easily when the base is located nearer one end of the car; and the offset required in trolley wire on curves is less as the trolley base is brought nearer the car end. With short double truck cars in city service, where the distance between truck centers is not great, and the trolley wheel may be easily observed from the rear platform, a single trolley pole may be used to advantage.

Third Rail Collector. The third rail collector has the greatest current collecting capacity of any current collecting device used in electric railway work. In the section on electric traction of the Standard Handbook for Electrical Engineers, it is stated that tests have been made which indicate that current may be collected at

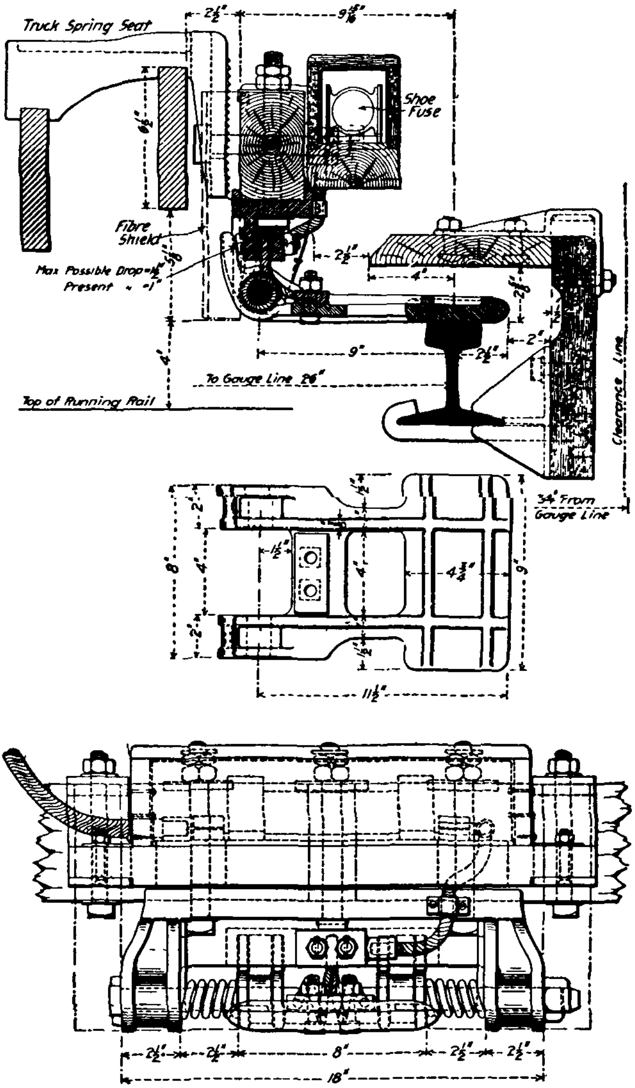


Fig. 8.—Third rail collector, New York subway.

the rate of 2000 amp. from a single shoe at a speed of 35 miles per hour and 500 amp. at a speed of 70 miles per hour. The two general classes of collectors are that in which the contact pressure is furnished by gravity and that in which the contact pressure is furnished by a spring. The shoes are generally made of wrought iron or cast iron and in some cases a steel wearing surface is used.

Fig. 8 shows the details of the third rail collector designed for operation on overrunning protected third rail in the New York subway. Contact pressure is furnished by a spring. The shoe is arranged with a stop by which its downward movement is limited. The maximum possible drop is 1¼ in., but this may be reduced by shims. Connection from the shoe to the supporting brackets is made by a flexible cable wound around the shaft so that it will not be affected by the motion of the shoe.

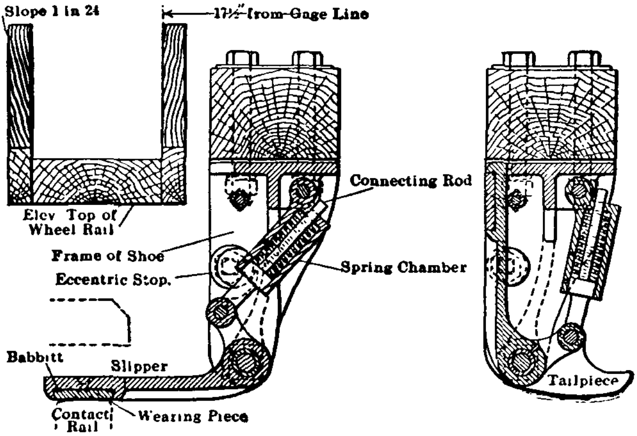


FIG. 9.—Third rail collector, Philadelphia and Western.

Fig. 9 shows the automatic third rail collector for use on overrunning third rail of the Philadelphia and Western Ry. The pressure between the shoe and third rail is about 10 lb. and is furnished by a spring which also serves to hold the shoe in the inoperative position. A tool-steel insert in the shoe is used for the wearing piece. This piece is held in place by two rivets and babbitt. During the winter the shoe is slanted slightly downward in order to make contact with the far side of the third rail, as that part of the rail is usually free from sleet. This slanting is done by means of the eccentric stop, the four quarter turns of which will also take care of a 1.5-in. reduction in diameter of the car wheels. There is also a rack adjustment on the car trucks for raising and lowering the collector beams. The automatic folding and unfolding of the shoe is obtained in the following manner: Where the shoes should either open or fold, as the case may be, two parallel metal strips which are sloped in opposite directions are placed along the tracks and set several inches apart. If the shoe is open, it will bear against the rising strip until it assumes a nearly vertical position,

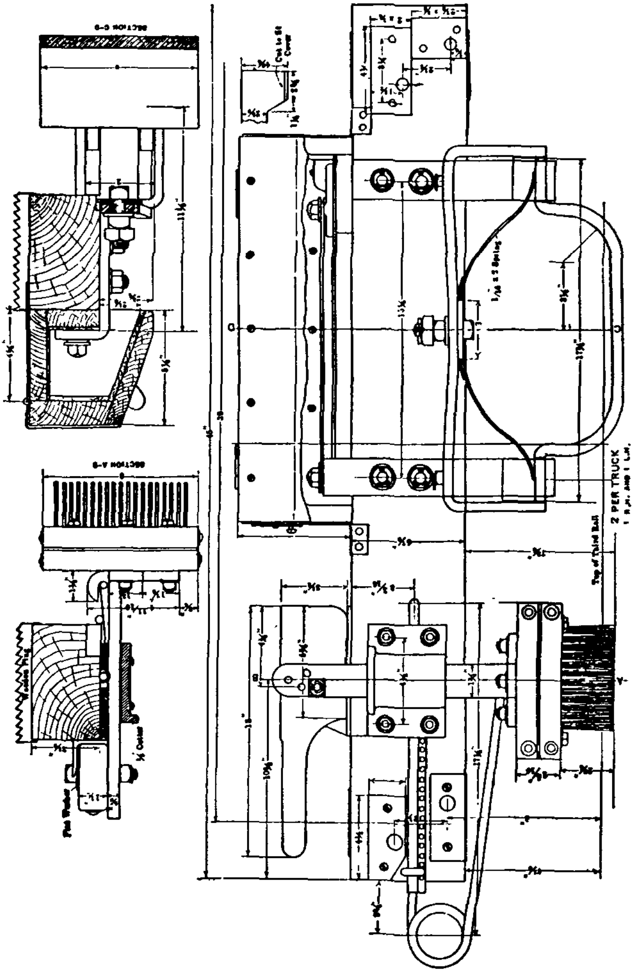


FIG. 10.—Third rail collector and sleet brush, Brooklyn.

and the spring then acts to close it. The rising of the shoe, however, is coincident with the lowering of a tail-piece which is at right angles to the shoe. When the shoe is to be opened, this tail-piece will be within range of the other inclined strip, so that it will be forced upward while the shoe moves downward to make contact with the conductor rail. The eccentric stop and the tail-piece prevent the shoe from dropping when unsupported by the third rail.

Fig. 10 shows a third rail collector and sleet brush of the Brooklyn Rapid Transit Co. This collector was designed to eliminate all links and castings, it is of wrought iron, and it allows a maximum height variation of 2.5 in. The wrought iron shoe replaces a link-suspended casting which was too fragile. Copper shunts were used on the casting, but the flat springs were found to be sufficient conductors on the type illustrated. The sleet brush is made of No. 23 B. & S. gage wire, which is renewed about every season.

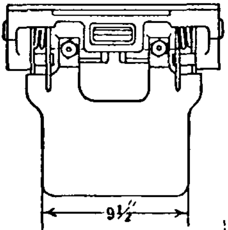


Fig. 11 shows a third rail collector for use with either over-running or under-running third rail. The tension or pressure of the shoe on the rail is

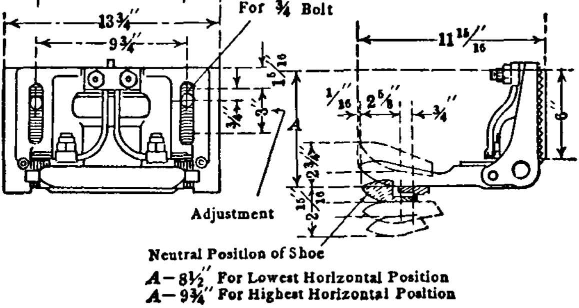


FIG. 11.—Third rail collector for over- or under-running.

regulated by the position of the spring in relation to the pivot or bearing points and may be set at any reasonable amount.

Pantograph Collector and Bow Collector. The pantograph collector differs from the bow collector essentially in the movement and position of the contact shoe. The shoe of the pantograph collector (Fig. 14) is controlled by a pantograph structure and moves in a vertical plane through the center of the mechanism as it rises and is depressed in action, while the shoe of the bow collector (Fig. 13) is trailed in a manner similar to that of the common trolley wheel and moves in a vertical curve. When used in high voltage operation, both types are insulated from the car by being mounted on porcelain insulators. Both types are generally placed in action and removed therefrom by means of a compressed air mechanism and in high voltage service this is often arranged to lower the collector automatically when the high voltage distributing box is opened. The scheme of the pneumatic operation of the low pantograph on New York Central locomotives is shown in Fig. 12. Near each master controller in the cab there is a valve by means of which the pantograph shoe may be raised or lowered. When air is applied, the shoe is lifted so as to make contact with the overhead rail. When air is released, the shoe drops; also if the shoe runs off the rail, it is tripped automatically and drops. Moving the handle forward operates the pilot valve, by means of which a slide valve is thrown to admit air from the reservoir to the cylinder of the contact shoe device. Pulling the handle back operates another pilot valve and the slide valve is thrown over to connect the air chamber of the contact device to the exhaust. The handle will spring back to the middle position from either direction. There are two of these overhead contact shoes which are controlled in common by either valve in the cab.

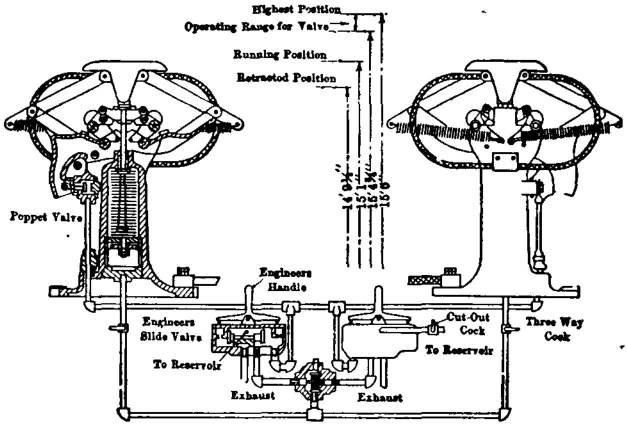


FIG. 12.—Low pantograph collector, New York Central.

Either pantograph or bow may be used when high speed, voltage or amperage or overhead construction will not permit satisfactory wheel trolley operation. In many places a pantograph is used in suburban service, but in city service the overhead construction usually permits only a wheel trolley. This is provided for by equipping each car with a pantograph in the middle and a wheel trolley pole at each end. Both the pantograph collector and the bow collector are built with many variations in the details of construction. The pantograph collector is built either with the simple pantograph and rigid contactor or with the pantograph base to follow great changes in trolley wire height and having the contactor spring-supported to follow minor unevenness. The bow collector may have a framework or pole or a pantograph base to follow variations in trolley wire height and from this the contactor is generally carried on a lighter secondary trailing structure made flexible to follow minor unevenness. The following, relative to successful current collector operation, is by Otis Allen Kenyon, *Elec. Ry. Journal:*

The successful operation of any system depends upon the ability of the contactor to keep in contact with the wire. Failure to do this produces arcing and hammering. The arcing destroys both the collector shoe and the wire, while the hammering accelerates the wear, kinks the wire and breaks the fastenings. The difficulty of keeping the collector in contact with the wire increases very rapidly with the speed, and the minimizing of this difficulty is one of the important problems in high speed operation. The variations in the position of the wire over the track and the swaying of the car require the collecting device to be so constructed as to adapt

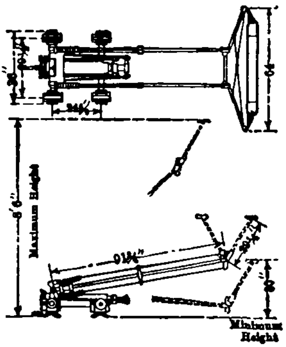


FIG. 13.—Bow collector.

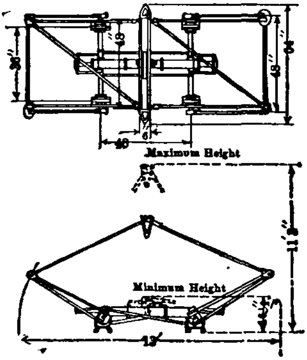


FIG. 14.—Pantograph collector.

itself to all such variations, thus forming a flexible connection between the wire and the car. At low speeds it is comparatively easy to design such a device, since all variations in position take place so slowly that the inertia of the collector does not prevent it from following the wire. However, at high speeds the effect of every little variation is exaggerated and tends to produce vibration. Large variations in position, such as are caused by passing under bridges, etc , when made gradually, have little or no effect on vibration. The three principal causes of vibration are:

(1) Unevenness of the contact wire, due either to imperfect suspension or non-uniform wear.

(2) Vibration and swaying of the car or locomotive.

(3) Inertia of the contact device.

Contact shoes have been made of copper, steel, aluminum and various alloys in attempts to get a considerable life of shoe together with small wear on the trolley wire. A shoe of "U"-shaped section containing lubricant in the form of grease has been found to reduce wear and singing. Aluminum shoes are light, consequently they have comparatively little inertia which makes possible a collector structure of high natural period of vibration.

Wear of the wire and shoe can be traced to two causes, mechanical and electrical. The mechanical causes are due to pressure and cannot be entirely eliminated. Non-uniformity of pressure is the most serious trouble. The springs can be so designed as to make the pressure constant for any position of the collector. However, the speed at which the collector changes its position affects the pressure to an extent which can only be controlled by reducing the inertia of the mass to be moved. Throughout Europe the average pressure of the contact shoe against the wire is about 11 lb. The electrical causes are secondary, being due to arcing, which is caused by insufficient contact surface. Arcing is especially destructive in that once started the surface is left more predisposed to arcing than before and conditions rapidly go from bad to worse. The

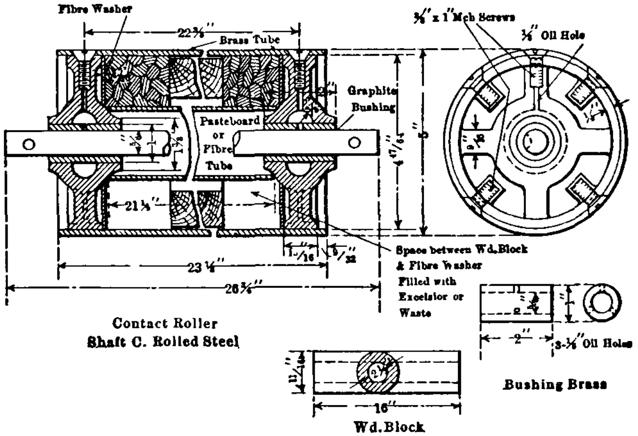


FIG. 15.—Roller trolley, Key Route, California.

cause is due to imperfect contact and vibration and the resulting wear is a function of the current rather than of the power. For a given power, the higher the voltage the less destructive is the arc. Soot deposited on the contact wire by steam locomotives increases the wear to an astounding extent. The Swedish commission found that the wear of the shoe on a soot-covered wire is about ten times as rapid as on a clean one. The aluminum shoes on the various types of collectors ran about 1500 miles on soot-covered wires. It is estimated in the report that on clean wires they would run at least 12,500 miles. The practice of the Swedish tramways indicates an average life of such shoes, when used in tramway service, of 12,500 miles before renewal, but if left until worn out they will run as high as 62,000 miles.

Collecting Capacity of Pantograph. Tests were conducted by the General Electric Company at Erie, Pa., in July, 1923, in which currents exceeding 5000 amperes were successfully collected by pantograph slide at speeds approximating 60 miles per hour, and with no sparking. These tests were made at 850 and 1500 volts, no appreciable difference resulting on account of voltage. The

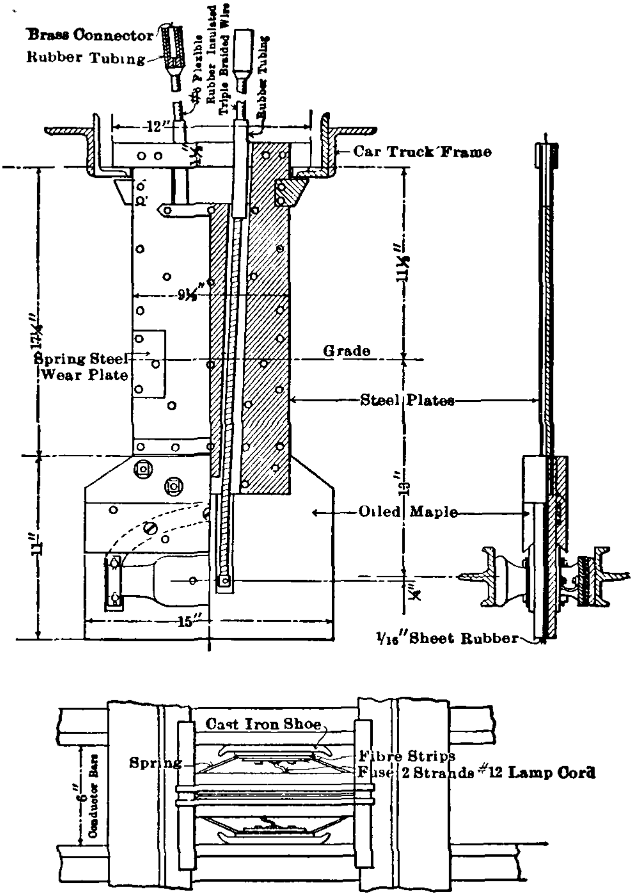


FIG. 16.—Slot Plow, Washington.

pressure of the pantograph against the trolley was 30 to 35 lb. The working conductor was two 4/0 grooved wires suspended from the catenary messenger by clips spaced 15 ft. apart on each wire and alternating. Such currents are greatly in excess of those required even in heavy locomotive practice, but the tests demonstrated the possibilities which could be met with the pantograph slide and with the working conductor in excellent condition.

Roller Trolley. The roller trolley is good in current collecting capacity, but it has more inertia than the pantograph shoe. For these reasons it is suited to service at heavier currents and at lower speeds than those to which the pantograph shoe is particularly adapted. Fig. 15 shows a roller trolley used successfully on the Key Route, California. The roller is mounted on a pantograph frame and weighs, complete with spindle, 28 lb. The wearing surface is a tube of non-arcing brass, supported on a wooden roller. The height of the trolley wire above the head of the rail varies from 14 ft. 6 in. to 22 ft., but by the pantograph the pressure of the roller against the wire is kept nearly constant at about 34 lb. The average mileage of the rollers is 55,000.

Top Contact Collector (Oerlikon Collector). Current is collected from either a trolley wire suspended from an inverted catenary or a trolley wire stretched tightly over insulators at the side of the track. The collector consists essentially of a curved hinged arm which sweeps over nearly a semcircle in a plane transverse to the longitudinal axis of the track. This collector is supported on insulators on the side of the car top. Normally the arm rests on top of the trolley wire with a pressure of about 1.5 lb. On cross-overs and in tunnels, where the trolley wire is carried over the track, the arm swings toward the center of the car, and is depressed, making contact progressively from the top around to the side and then underneath the trolley wire. In addition, the saddle which carries the arm is movable laterally, increasing the radius of action.

Slot Plow. The slot plow is arranged to collect current from conductor bars which are located about 6 in. apart and a foot below the track rails. It consists of a flat frame made of insulating material suspended from the car and carrying the necessary conductors from the conductor bar to the car. This frame extends through the 5/8-in. or 3/4-in. slot between the slot rails and carries a contact shoe at either side of its lower end. It insulates the contact shoes from each other. The contact shoe is generally made of chilled cast iron and is held against the conductor bar by a flat steel spring by which it is fastened to the frame. The frame carries thin wear plates of hard steel for protection against excessive wear in rubbing over the slot rails. Electrical connection between the contact shoes and the upper end of the plow is by fuse connectors and rubber-insulated conductors. These fuse connectors are connected to the contact shoes and serve to protect the plow from damage by short circuit from shoe to shoe. Fig. 16 shows a slot plow used by the Capital Traction Co., Washington, D. C. The suspension of the plow from the truck must admit of free lateral movement in order that the plow may not be damaged by truck play and unevenness of track and slot rails. Where the car moves between a conduit section and a trolley section, the plow is removed or replaced by a man stationed in a pit at the junction of the two sections.

SECTION VII

TRUCKS

General Classification. Trucks are of two general classes, namely, the single truck and the double or swivel truck, the latter sometimes called the bogie truck.

Single Truck Cars are intended for city street service and where the maximum speed does not exceed 25 miles per hour. The two axles of the single truck are more or less rigidly alined by side frames, therefore the wheel base must be short in order to negotiate sharp curves. This limits the length of car to a maximum of about 30 feet overall.

Double Truck Cars are equipped with two distinct trucks joined together by the car body framing. Each swivel truck consists of two or more axles centered in common side frames which are joined together by cross pieces. The central cross members usually form a transom surrounding a spring supported bolster. The bolster carries a center plate on which the truck swivels and upon which the car body rests, and also side bearing plates which maintain the body level. The usual type in electric service carries two axles; for very high speed service or for very heavy cars, three-axle trucks may be desirable.

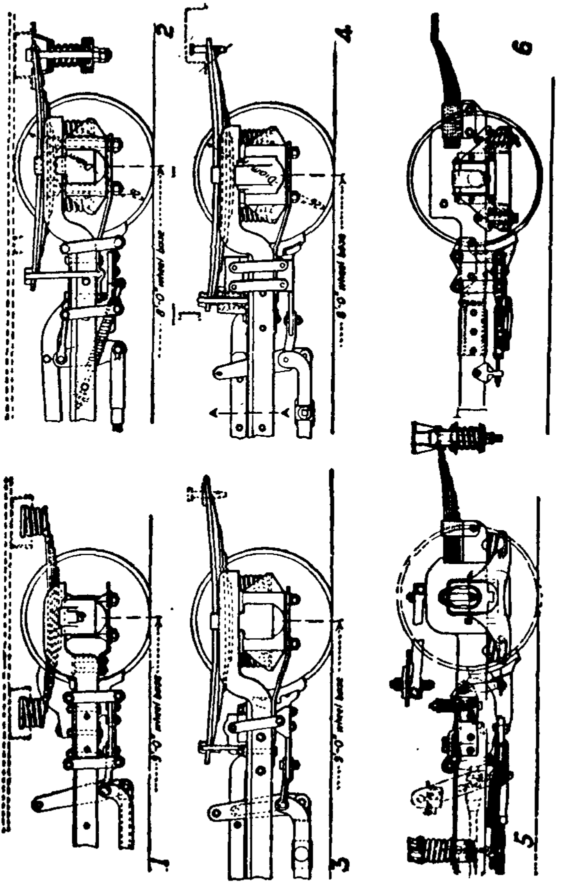
Design of Trucks. It has come to be generally recognized that definite engineering principles should be applied in the design of trucks for electric railway service, and that weight of trucks is not a prime essential to easy riding qualities. A number of variations in the type and arrangement of springs and in the general design and arrangement of truck parts are discussed in the following paragraphs. In addition to those matters of general design, the following points have been mentioned by Norman Litchfield as those which should be considered in connection with truck design: weight of car and of load, standing; shifting of weight from side to side, due to centrifugal force; shifting of a portion of the load from the rear to the front truck in braking, and vice versa in acceleration; similar shifting of a portion of the load from one pair of wheels to the other pair in the same truck under similar conditions; the forces set up by the motor during acceleration; the flywheel effect of rotating parts; the forces set up by friction of brake shoes on the wheels during braking; and the distortional effect in rounding curves.

Developments in Truck Design. Reductions in weight and in maintenance, together with improvement in riding qualities, have been the outstanding objectives in the development of truck design, principally due to the developments in light weight car construction. The return to the single truck design with the Birney type of safety car, and the light weight construction demanded, produced a marked development in the single truck. The principal differences in present types of construction found in trucks built by various manufacturers are in methods of constructing the side frame and in the arrangement of springs to produce easy riding. In addition, the layout of brake rigging and design of brake hangers for the purpose of eliminating chattering and reducing maintenance represent important variations in construction. Combined with the arrangement of springs to provide vertical flexibility, various methods and devices are utilized for the purpose of allowing lateral movement of the bolster and at the same time limiting the amplitude of its oscillations. Further refinements in design by various manufacturers are primarily for the purpose of increasing rigidity and durability and decreasing maintenance costs. Rather wide departures from conventional designs have been made by several operators who have felt the need for more fundamental developments than were available in the products of the equipment manufacturers, and this work is at present in process of development; Fig. 14 is an example.

Single Trucks. In the construction of single trucks a number of designs are in use. The demand for reductions in weight and the requirements of minimum maintenance and easy riding are reflected in the variations in design. There is a tendency to avoid built-up side frames and to secure the rigidity and freedom from maintenance offered by a one-piece side structure, as well as to avoid heavy castings wherever possible and to substitute lighter weight forgings. For the purpose of obtaining minimum weight, one type of construction utilizes two deep channels of high grade steel assembled so as to form a closed box structure filled with wood and welded together along the upper corners. Of the one-piece side frames, one type is machined from a solid flat billet, another is forged to shape, and in another the side frames are pressed out of a comparatively thin plate, providing reinforcing flanges and ribs at the critical points.

The application of brake rigging to the single truck is by several arrangements designed to eliminate chattering and insure a true and square contact between the shoe and the wheel. The method of employing two parallel link hangers, applied either directly to or adjacent to the side frame so as effectively to guide the shoe in its movement back and forth, as well as the design with a center hanger back of the brake beam and a single brake hanger for each shoe head at the side, are both widely used.

The arrangement of springs is important in connection with the results obtained in the operation of single trucks. With axles supported squarely in the truck frame at the journals, the requirement for negotiating sharp curves existing in most cities limits the allowable wheelbase to between 8 and 9 ft. as a maximum. Since the car body of a single truck car extends considerably beyond these lengths, the overhang each side of the wheel line is comparatively large and results in a tendency toward teetering and nosing in operation. Various methods are employed to overcome this condition, practically all of which center around the general idea of increasing the effective spring base by the extension of the truck side frame a considerable distance beyond the wheel line at each end, to act as a support for springs. By the use of overhanging half-elliptic springs secured at one end to the truck side frame, or quarter elliptic springs rigidly secured to the side frame close to the pedestal, the same effect is obtained. Fig. 1 shows a simple spring arrangement with a half-elliptic spring fastened rigidly to



FIGS. 1, 2, 3, 4, 5, 6.—Types of spring arrangements, single trucks.

the journal box, and short helical springs at either end extending to the car body support. In this construction the effective spring base is equal to the wheelbase of the truck. Fig 2 illustrates the next step in providing a more flexible spring construction. In this arrangement the truck side frame is supported on helical springs mounted on extended ears on the journal box, and the half-elliptic is, as in Fig. 1, again supported on the journal box, but its inside end is fastened by a long link to the truck side frame instead of to the body. The outside end of the half-elliptic is connected to the body through a long helical suspension spring, which allows considerable lateral and longitudinal motion. The long link employed for fastening the inside end of the half-elliptic to the truck side frame also contributes to a comparatively free movement. Fig. 3 illustrates the next step in the development of this type of truck, in which the loose and flexible connection to the body at the outside end of the half-elliptic is eliminated, and the link connecting the inside end of the latter to the truck frame is materially shortened. The final development in this design to produce easy riding and at the same time limit excessive oscillation is shown in Fig. 4, where the construction is similar to Fig. 3, except that a short compression spring is interposed in the link connection to the truck frame for

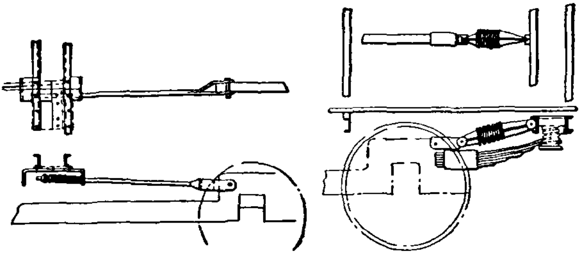


FIG. 7.—Driving connections between single truck and car body.

the purpose of reducing the strains imposed on the parts at this point and giving a little added vertical flexibility. In Fig. 5 the side frame is carried on short helical springs resting upon ears extending from the sides of the journal box. The main body supporting spring consists of a stiff quarter-elliptic rigidly fastened to the side frame adjacent to the pedestal. The connection to the body is usually made through a short helical spring, flexibly connected to the end of the quarter-elliptic. A flexible device for absorbing lateral strains between the body and truck and for maintaining the body and truck in alinement is also shown in Fig. 5, as well as the details of the quarter-elliptic spring fastening to the side frame, the brake hanger arrangement, and the journal box construction. Still another method of supporting the side frame on short helical springs set on extension ears on the side of the journal box, is shown in Fig. 6. The quarter-elliptic spring is fastened with a strap and projects under a lip on the pedestal frame for firm anchorage. This type of truck employs a centrally supported brake-beam hanger with the brake hangers supported from gusset plates bolted to the side frame. The brake hanger is of the half-ball type, in which a semi-ball and socket arrangement with a compression spring on the connecting bolt is utilized to compensate for wear.

Two arrangements for transmitting the driving force between the single truck and its car body are shown by Fig. 7, in addition to the one shown in Fig. 5.

The Radiax truck is an example of a single truck in which the two axles are not carried rigidly parallel, but are allowed some motion, so as to adjust themselves more nearly radially in passing around sharp curves. Such construction permits a longer wheel base to be used than would be possible with a rigid truck on the same radius track curves. The principal feature of the Radiax truck is the swing link suspension of the truck from the journal boxes which is illustrated by Fig. 8, the double bolt arrangement tending to return the axles to their original alinement when on straight track.

Swivel or Double Trucks. In this type of truck, modern construction shows side frames forged in one piece, made of a steel casting, built up in the conventional diamond arch bar construction, or built up of structural shapes. The forged side frame has the advantages of a one-piece rigid structure, in which by proper design the metal can be distributed accurately in proportion to the stresses imposed. This type gives a very rigid truck and freedom from maintenance trouble on the side frame structure. In the application of cast steel to modern trucks, the M.C.B. equalizer bar type of truck, as shown in Fig. 9, lends itself to the elimination of the heavy pedestal castings, which were used with earlier trucks of the cast steel type. As here shown, the truck side frame and transom have been cast in one unit, giving an extremely rigid structure from the standpoint of maintenance of alinement, and also allowing the brake rigging and spring suspension parts to be conveniently applied.

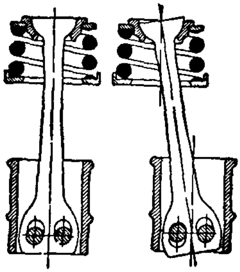


Fig. 8.—Swing link of the "Radiax" truck.

The arch bar type of truck gives both low weight and first cost. In this type of truck high grade heat treated steels of either high carbon or alloy composition are being used to increase the structural strength of the built-up side frame. In some cases the arch bar is of the conventional flat section, while in other trucks a channel-shaped member is used for the bar in order to obtain the maximum sectional rigidity with minimum weight (Figs. 10 and 11).

In the spring suspension and brake rigging arrangement, there is a variation over a considerable range of design. The customary full-elliptic spring under the bolster appears in a number of types, but various features have been added for the purpose of obtaining improved riding performance. The heavy spring plank formerly used under such springs has in some cases been entirely eliminated, the full-elliptic spring is carried in a swinging hanger or swing link provided with an appropriate socket at the lower end to form a seat for the lower spring band, and the two hangers or stirrups are tied together across the truck with a comparatively light structural

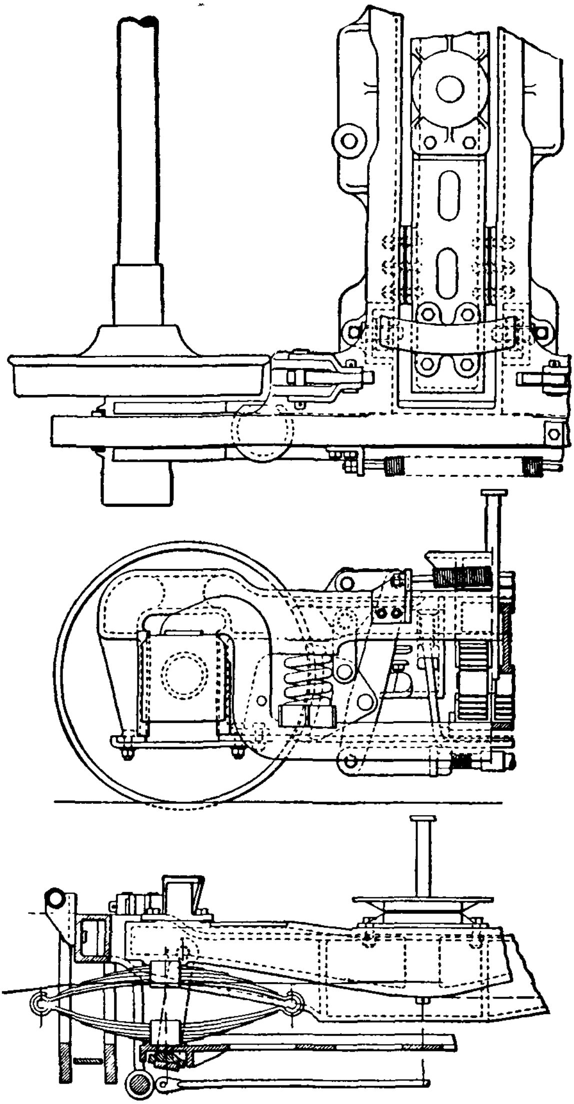


FIG. 9.—M.C.B. type swing bolster swivel truck.

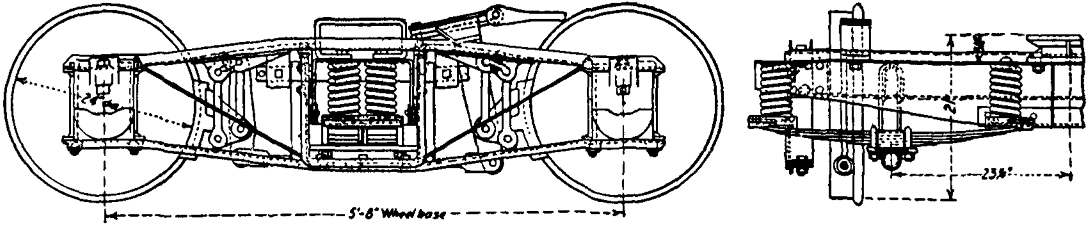


FIG. 10.—Arch bar type swivel truck with half-elliptic bolster springs supported from long link.

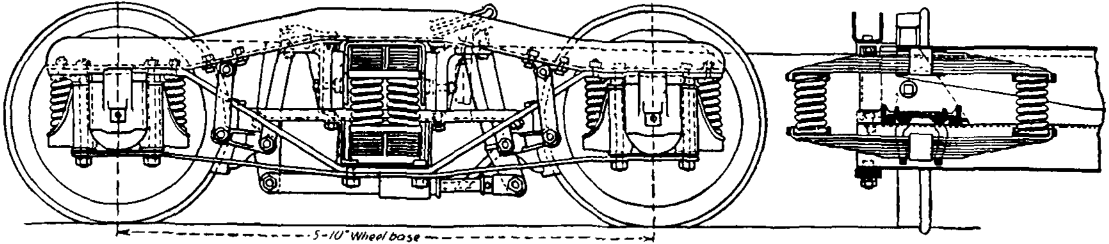


FIG. 11.—Modified arch bar type swivel truck with helical springs at journal boxes and combination half-elliptic and helical springs under bolster.

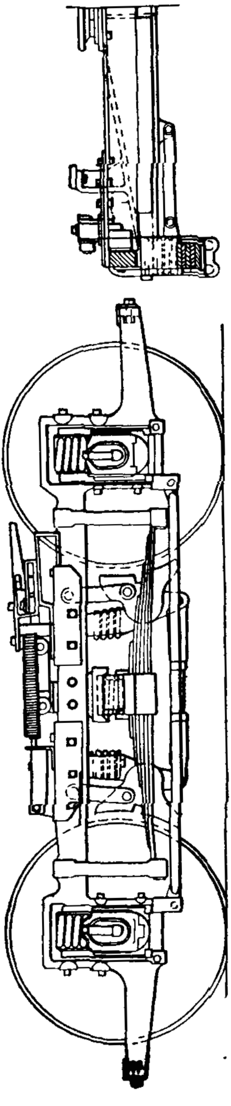


FIG. 12 —Swivel truck with forged steel side frames and longitudinal half-elliptic springs carried in swinging links

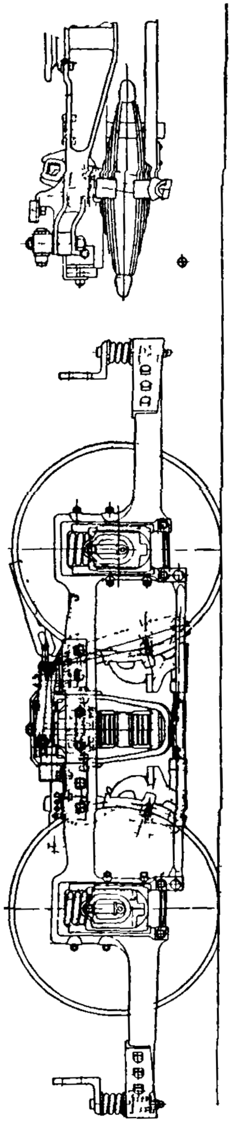


FIG. 13.—Swivel truck with forged steel side frames and elliptic springs under swinging bolster.

member. Light helical springs mounted on top of the full-elliptic and set into upper and lower spring caps which are provided with lugs which come in contact after the capacity of the helical is exceeded are utilized for the purpose of absorbing road shocks and vibrations while the car is operating under light load. This light action spring goes out of operation when the spring cap lugs come in contact, and from that point on the total load is carried by the

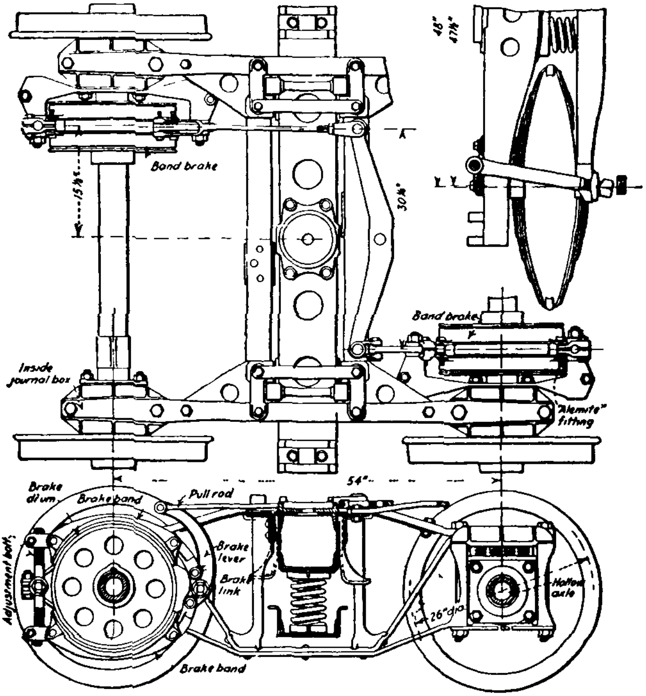


FIG. 14.—Minneapolis light-weight truck with band brakes and hollow axles

full-elliptic. Bolster guide links fastened between the truck transom and the bolster are utilized in some trucks of this general type for the purpose of preventing the bolster from rubbing against the transom sides. In addition to the simple arch bar type of construction with the bar resting directly on the journal box, some recent trucks utilize a type of construction in which helical springs are provided between the arch bar and projecting ears on the sides of the journal box. The side frame, instead of being clamped down around the box by the long bolts between the top and bottom bars, is clamped to tubular struts which ride through openings cored in the journal box structure, so that the equivalent of a pedestal construction is provided.

The use of half-elliptic springs carried longitudinally in links or stirrups hung over the top of the side frame has been customary for a number of years, as shown in Fig. 12. The light-load helical spring described above is utilized with this construction and is imposed between the top of the half-elliptic and the truck bolster. The design of link in which the half-elliptics are carried is such that the amplitude of the lateral swing is rapidly checked by the angularity which these stirrups assume. In modern trucks the entire theory of securing easy riding from the standpoint of absorbing lateral oscillation seems to be the provision of relatively short suspension links in the spring system, which are comparatively free at the beginning of an oscillation, but which quickly assume an angularity that checks the amplitude of the swing.

Another type of spring arrangement which has been used to a considerable extent, particularly in trucks of the modified arch bar construction, is a type in which a half-elliptic spring mounted crosswise is carried independently at either side of the truck in a long stirrup supported from the transom. At the ends of the half-elliptic, comparatively long helical springs are set up vertically and directly support the truck bolster. This same general type of construction is represented in Fig. 13 by two half-elliptics mounted crosswise at either side of the truck, one fastened to the truck bolster above and the lower fastened in short swinging links carried on a comparatively large structural member extending across the truck. This lower structural member is used to tie the two sides of the truck together, and the usual transoms above are largely made of wood. Between the ends of the half-elliptic springs described, vertical helical springs are set so as to hold the two half-elliptics a considerable distance apart.

Electric Railway Axles

Standard Specifications for all types of electric railway axles are contained in the A.E.R.E.A. Manual. The following information is an outline of some of the more important points covered by the complete specifications.

Process of Manufacture. The steel shall be made by either the open hearth or electric furnace process, except that the open hearth hammered steel axles may be made by only the open hearth process.

Heat Treatment. 1. Quenched and Tempered Carbon Steel Axles: For quenching and tempering, the axles, immediately after forging, shall be allowed to cool to a temperature below the critical range under suitable conditions to prevent injury by too rapid cooling. They shall then be uniformly reheated to the proper temperature to refine the grain (a group thus reheated being known as a "quenching charge") and quenched in some medium, under substantially uniform conditions for each quenching charge. After quenching they shall be uniformly reheated to the proper temperature for tempering or "drawing back" (a group thus reheated being known as a "tempering charge"), and allowed to cool uniformly.

2. Annealed Carbon Steel Axles. The procedure to be followed in annealing shall consist of allowing the axles, immediately after forging, to cool to a temperature below the critical range under suitable conditions to prevent injury by too rapid cooling. They shall then be uniformly reheated to proper temperature to refine the grain (a group thus reheated being known as an "annealing charge") and allowed to cool uniformly.

Turning. Quenched and Tempered Carbon Steel Axles and Annealed Carbon Steel Axles: The forgings shall conform to sizes and shapes specified by the purchaser. Unless otherwise specified

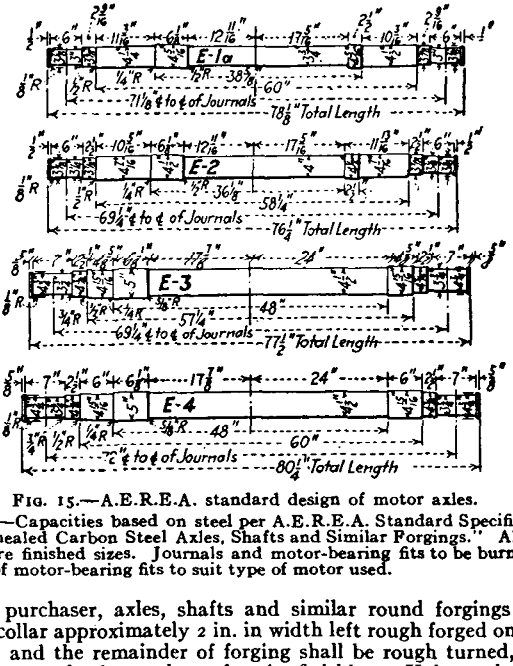


FIG. 15.—A.E.R.E.A. standard design of motor axles.

NOTE.—Capacities based on steel per A.E.R.E.A. Standard Specification for "Annealed Carbon Steel Axles, Shafts and Similar Forgings." All sizes shown are finished sizes. Journals and motor-bearing fits to be burnished. Length of motor-bearing fits to suit type of motor used.

by the purchaser, axles, shafts and similar round forgings shall have a collar approximately 2 in. in width left rough forged on each forging, and the remainder of forging shall be rough turned, with an allowance of ⅛ in. on the surface for finishing. Unless otherwise specified, all axles, shafts and similar forgings shall have ends faced, with an allowance of ⅛ in. on each end for finishing, and shall be centered for 60 deg. lathe centers, with clearance for points.

Chemical Composition. Quenched and Tempered Carbon Steel Axles and Annealed Carbon Steel Axles: The steel shall conform to the following limits in chemical composition:

Carbon	Not over 0.60 per cent
Manganese	0.40 to 0.70 per cent
Phosphorus	Not over 0.05 per cent
Sulphur	Not over 0.05 per cent

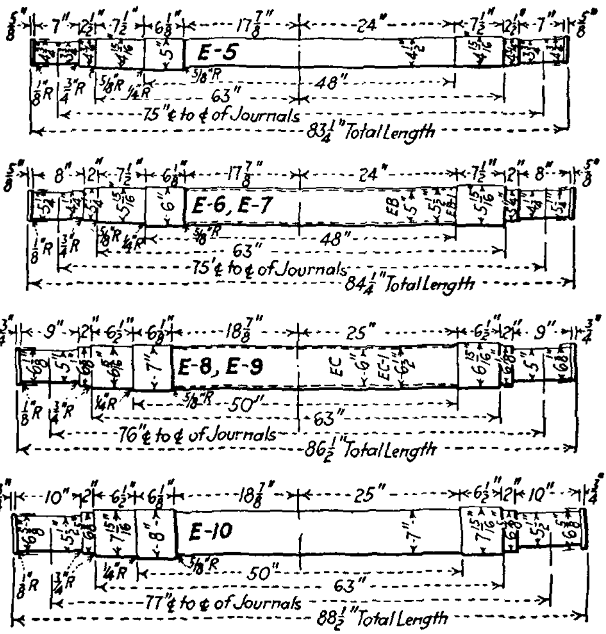
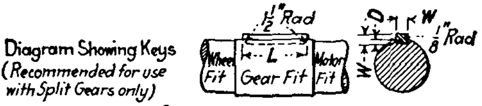


Fig. 16.—A.E.R.E.A. standard design of motor axles.

Note.—Capacities based on steel per A.E.R.E.A. Standard Specification for "Annealed Carbon Steel Axles, Shafts and Similar Forgings." All sizes shown are finished sizes. Journals and motor-bearing fits to be burnished. Length of motor-bearing fits to suit type of motor used.



Summary of Axle and Gear Data

Type	Journal	Dia. Motor Fit	Dia. Gear Fit	Dia. Wheel Fit	Wheel Hub Length	Wheel Hub Rough Dia.	Distance between Hubs	Centers of Journals	Max. Capacity (pounds)	Length Gear Hub	[illegible]	Extension of Gear Hubs Wheel Side	Extension of Gear Hubs Motor Side	Key W"	Key D"	Key L"
E-1a	3 × 6	3 1/4	4 5/16	4 1/4	4 1/2	7 1/4	48	71 1/8	13000	4 5/8	6 1/2	1	1/8	3/4	3/8	4
E-2	3 1/4 × 6	4	4 1/2	4 7/16	4 1/2	7 1/4	48	69 1/4	13000	4 5/8	6 1/2	1	1/8	3/4	3/8	4
E-3	3 3/4 × 7	4 1/2	5	4 15/16	5 or 5 1/2	8 1/4	48	69 1/4	16000	6 1/8	7	1	1/8	1	1/2	5
E-4	3 3/4 × 7	4 1/2	5	4 15/16	5 or 5 1/2	8 1/4	48	72	15000	6 1/8	7	1	1/8	1	1/2	5
E-5	3 3/4 × 7	4 1/2	5	4 15/16	5 or 5 1/2	8 1/4	48	75	14000	6 1/8	7	1	1/8	1	1/2	5
E-6	4 1/8 × 8	5	6	5 15/16	6	10 1/4	48	75	18000	6 1/8	8	1	1/8	1	1/2	5
E-7	4 1/4 × 8	5 1/2	6	5 15/16	6	10 1/4	48	75	22000	6 1/8	8	1	1/8	1	1/2	5
E-8	5 × 9	6	7	6 15/16	6	10 1/4	50	76	27000	6 1/8	9 1/2	1/8	1	1	1/2	5
E-9	5 × 9	6 1/2	7	6 15/16	6	10 1/4	50	76	31000	6 1/8	9 1/2	1/8	1	1	1/2	5
E-10	5 1/2 × 10	7	8	7 15/16	6	11 1/4	50	77	38000	6 1/8	10 1/2	1/8	1	1	1/2	5

Fig. 17.—Table of axle and gear data for standard A.E.R.E.A. axles.

Tensile Tests. After final treatment, the forgings shall conform to the following minimum tensile properties. When applied to axles, the diameter indicated shall be taken as the finished diameter of the motor bearing seat.

	Quenched and tempered carbon steel axles		Annealed carbon steel axles	Hammered steel axles
	Up to 4 in. in diameter or thickness	Over 4 in. up to 7 in. in diameter or thickness		
Tensile strength, lb. per sq. in.	90,000	85,000	80,000	80,000
Elastic limit, lb. per sq. in.	55,000	50,000	40,000	40,000
Elongation, in 2 in., per cent	Not under 20.5	Not under 20.5	20	20
Reduction of area, per cent	39	39	32	25

A *heat-treated axle* is one which is allowed to cool after forging, is then reheated to proper temperature, quenched in some medium, and then reheated to the proper temperature to refine the grain. The exact treatment of a particular steel depends upon its chemical composition. If the chemical composition of an existing axle be known, the axle may be properly heat-treated and then turned down to a smaller size for service. The purpose of heat treatment of

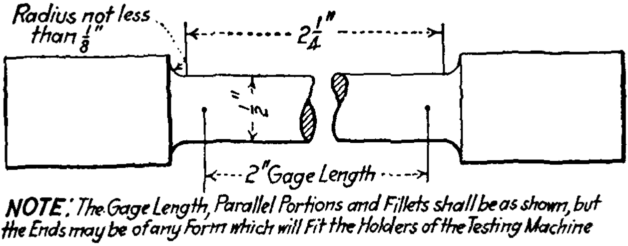


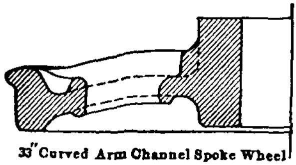
FIG. 18.—A.E.R.E.A. standard tensile test specimen for axle steel.

axles is to relieve the internal stresses set up by improper cooling, also to so rearrange the structure of the steel that it will resist the action of repeated blows which, though each would have an effect within the endurance of the original untreated steel, such continued repetition would cause the untreated steel to fail.

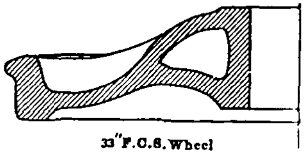
Axle Breakage. The principal reasons for removal of axles are excessive wear and the occurrence of cracks or breaks. A few electric railways that are operating high speed service are removing axles after they have reached a certain mileage limit regardless of appearance on inspection, experience having shown that axle breakage is most frequent after this point has been reached. This mileage limit varies with various companies from 200,000 to 500,000. Many electric railways specify a wear allowance, and axles are either scrapped or turned down to smaller diameters when this wear is reached. These wear allowances vary from $\frac{3}{16}$ in. to $\frac{1}{2}$ in. of diameter of bearing fit. One company stamps on the end of each axle the date that it is placed in service. After four years of service, all interurban axles are removed and annealed, regardless of the mileage which they have made. Careful inspection is made of wear, and tests are made for flaws. Most of the broken axles on this system have occurred close to the wheel. The fillet at this point is tested for flaws or cracks by gouging up a groove in the fillet with a small diamond pointed chisel; if it transverses a flaw, a square chip will be thrown off. Another company found that many axle breaks start at the corner of the keyway at the base of the hole drilled to start the keyway cut; this has led to a change in the practice of cutting keyways, so that they are now milled at the end instead of being drilled.

In testing axles for cracks, the general practice is first to place the axle in a lathe and rotate it so that its trueness can be determined. After this a hammer test of some form is usually applied. One test consists of painting the axle with a coat of black lead and oil over the entire surface. The axle is then wiped clean and again painted with a coat of white lead and oil, after which the axle is struck with an 8-lb. sledge on each end. If there are cracks, flaws, breaks or other defects, the oil and black lead will show through the white paint. Many railways use this test except that the first coat of black lead and oil is omitted. Others report that if the axle is carefully cleaned and then struck several sharp blows on the ends with a sledge, the oil which remains in any crack will be forced out and thus indicate the position of the crack. Another method of testing axles without removing the wheels consists of immersing the axle with wheels mounted in a tank of hot oil. After removal, the oil is carefully wiped off, and the axle is covered with whiting. The axle is then supported on a block at the center with the wheels hanging free, and is struck several blows with a heavy sledge. Wherever there is a crack, a fine thread of oil works up from the crack under the vibration, and can be seen against the white. Several roads report that they clean axles very carefully and then go over them with a magnifying glass, while on one road the test consists of taking a very fine cut with a sharp lathe tool over the section where the crack is most likely to develop; if the axle has a crack, it is then apparent to the operator.

CHART SHOWING TYPES AND WEIGHTS OF 33-IN. CAST IRON WHEELS RECOMMENDED FOR CITY AND INTERURBAN SERVICE BY THE GRIFFIN WHEEL CO.



33" Curved Arm Channel Spoke Wheel



33" F.C.S. Wheel

Weight of car	City service, 2½-in. tread		Light inter-urban service, 3-in. tread		Weight of car	Heavy inter-urban service, 3½-in. tread	Heavy inter-urban service, M.C.B. Flg. & Trd.
	Spoke	F.C.S.	Spoke	F.C.S.		F.C.S.	F.C.S.
32,000	440	....	490	....	55,000	640	670
36,000	460	....	510	....	60,000	660	690
40,000	480	....	530	510	65,000	680	710
44,000	500	480	550	530	70,000	700	730
48,000	520	500	570	550	75,000	720	750
52,000	540	520	590	570	80,000	740	770
56,000	560	540	610	590	85,000	....	790
60,000	580	560	......	610	90,000	....	810
64,000	600	580	....	630	95,000	....	830
68,000	620	600	....	650	100,000	....	850

NOTE: Weights of wheels in the above table are for cars operated under normal conditions.

Effect of Dimensions on Weight of Cast Iron Wheel. Considering cast-iron wheels of the same general design, a 36-in. wheel will weigh approximately 100 lb. more than a 33-in. wheel, and a 30-in. wheel will weigh approximately 75 lb. less than a 33-in. wheel. Increasing the width of tread affects the weight of wheel approximately as follows:

Diameter of wheel, in.	Increase in weight of wheel per inch increase in width of tread, lb.
36	60
33	50
30	40
24	32
20	24

Wheel Grinding. Chilled cast-iron wheels which would have been removed from service because of worn flat spots, slight chipping, worn tread, or sharp flange have been made to give from 25 to 40 per cent more service by grinding. By close attention to the grinding out of flat spots track and truck maintenance costs have been reduced in addition to securing the increased mileage. Whether the wheel is worn out or another grinding will be economical may be decided from a comparison of the worn diameter

WEIGHTS AND DIMENSIONS OF F.C.S. (CAST-IRON) WHEELS RECOMMENDED BY THE GRIFFIN WHEEL CO.

Service*	Diam. of wheel, in.	Weight of wheel, lb.	Weight of car, lb.	Wheel dimensions, inches (see Fig. 19)														Number of brackets
				b	c	d	e	f	g	h	i	j	k	l	m	n	o	
A	26	360	17,000 to 38,000	5	4½	1⅝	4¼	4¼	.....	.....	⅞	1⅝	¼	⅝	1	2½	3 23/32	12
A	30	410	36,000 to 44,000	5	4½	1⅝	4¼	4¼	.....	.....	15/16	1⅝	¼	⅝	1	2½	3 23/32	12
A	33	485	36,000 to 44,000	5	4½	1⅝	4⅜	4⅜	¾	23/32	¾	1 9/16	¼	⅝	1 1/32	2½	3 23/32	15
A	33	510	40,000 to 48,000	5¾	5 7/16	⅞	4⅝	4⅝	13/16	¾	13/16	1 9/16	¼	⅝	1 1/32	2½	3 23/32	15
B	33	535	46,000 to 56,000	6	5 15/16	⅝	4⅝	4⅝	15/16	⅞	15/16	1 9/16	¼	⅝	1 1/32	2½	3 23/32	15
C	33	560	44,000 to 54,000	6	5 15/16	1⅛	4⅝	4⅝	15/16	⅞	15/16	1 9/16	¼	¾	1⅛	3	4 9/32	15
D	33	640	50,000 to 60,000	6 7/16	5 15/16	1 3/16	4⅝	4⅝	1 1/16	1	1 1/16	1⅝	¼	⅞	1 3/16	3½	4 15/16	15
E	33	730	60,000 to 70,000	6⅝	6 15/16	⅞	5 1/16	4⅞	1⅛	1 1/16	1⅛	1⅝	¼	1	1 15/64	4 11/32	5⅝	15
E	33	800	70,000 to 80,000	6⅝	6 15/16	⅞	5 1/16	4⅞	1¼	1 3/16	1¼	1¾	¼	1	1 15/64	4 11/32	5⅝	15
E	33	850	80,000 to 100,000	6⅝	7 15/16	⅞	5 9/16	5⅝	1⅜	1 5/16	1⅜	1⅞	¼	1	1 15/64	4 11/32	5⅝	15
E	36	800	60,000 to 70,000	6¾	5 15/16	⅞	4¾	4¾	1	1	1⅛	1⅞	¼	⅞	1 3/16	3½	4 15/16	18
E	36	925	80,000 to 90 000	7	7	⅞	5¼	5⅛	1⅜	1 3/16	1¼	.....	.....	1	.....	4 11/32	.....	16

* A = City. B = Heavy City. C = Light interurban. D = Ordinary interurban. E = High speed interurban.
F.C.S. wheels made from either double plate or single plate design.

or circumference with the original depth of the chill. A high abrasive such as emery or alundum in the form of a block or wheel has been used for the grinding. The best practice is to grind both mating car wheels, thus maintaining the same diameter for both. A method of grinding out flat spots employed on many roads is to

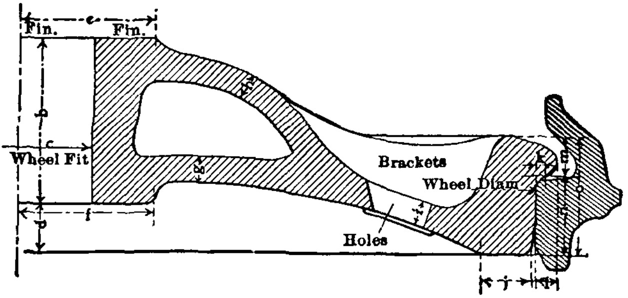


FIG. 19.—Griffin Wheel Co. F.C.S. cast-iron wheel. See p. 400 for dimensions.

use a brake-shoe of abrasive material until the flat spot is ground out. This method calls for close watching in order that the grinding shoe may be removed as soon as the flat spot is eliminated, and so additional flat spots will not develop. A method which has been used with success is to jack up the end of the car which carries the flat wheel and spin that wheel against a block of the abrasive while the other pair of wheels is blocked. An adaptation of this method is to place a block of emery, suitably retained with approach inclines at each end of the emery block, on the rail. A similar device, with the emery replaced by a system of rolls, is placed on the opposite rail. The car is run up on these blocks with the wheel to be ground resting on the emery block and the mating wheel on the rollers. All motors but the one driving the axle are cut out, and with the car blocked in position, the motor is run, grinding the wheel against the emery.

The use of the abrasive in the form of a grinding wheel in a grinding machine is most satisfactory. A separate grinding wheel is used for each of the mating car wheels. The car wheels and grinding wheels are revolved in opposite directions and the dust is conducted away by a blower. The car wheels are revolved at 10 to 15 r.p.m. The grinding wheels are from 14 to 18 in. in diameter by from 2 to 2½ in. in thickness and have a peripheral speed of about 5000 ft. per minute. In some machines the feed is automatic. The power required depends upon the rapidity of grinding, but for ordinary work it amounts to about 10 h.p. There are two general types of grinding machine; the floor grinder and the pit grinder. In using the floor grinder the axle carrying the pair of wheels to be ground is removed from the truck and mounted in bearings in the grinder, a sprocket wheel is fastened to the axle and it is revolved by a chain driven by the motor which drives the wheels. In the pit grinder the end of the car carrying the flat wheel is raised, a short piece of rail is removed from beneath each of the wheels to be ground, the grinding wheels are adjusted to the wheel treads in place of the rails removed and the car motor driving the pair of wheels being ground is run with the controller on the first notch and a water rheostat is used in place of the regular grid resistor. If trailer wheels are to be ground a sprocket wheel is attached to their axle and is driven by the grinder motor. The removal of ordinary flat spots requires from 10 to 40 minutes actual grinding.

It has been found advantageous to grind chilled cast iron wheels after mounting them and before putting them into service. Such grinding will correct for possible defects and thus aid in securing the most satisfactory wheel service. Wheels may thus be brought to the same diameter, wheels out of round or warped may be trued, and in case the axle has not been turned perfectly, the wheels bored not in the center, or the wheel improperly pressed onto the axle, grinding the wheel will make the wheel tread, flange and journal coaxial.

Opinion varies as to what conditions warrant the purchase of a wheel grinder. A majority of master mechanics, in a canvass made by the *Electric Railway Journal* in 1923, indicated that on roads having 100 pairs or more of cast iron wheels, the economies obtained would pay interest on the investment in a grinder. Individual opinions varied from 50 pairs to 500 pairs of wheels. Wheel grinders vary in price, depending on the degree of efficiency, and a purchase should be made commensurate with the actual requirements of the railway.

Wheel truing brake shoes have the advantage that the wheels are ground without removing cars from service. Frequently it has been found that wheel truing brake shoes do not leave the wheels concentric with the journal. The method is slow compared to pit or floor type grinders. In this method some companies pay particular attention to the increased coefficient of friction of the brake shoe and proportion the brake leverage accordingly.

The main advantage of the pit grinder as compared with the floor type grinder lies in the saving of expense in jacking up cars, removing and replacing wheels, and handling wheels in and out of the grinder. There is also a reduction in time that the car is withheld from service. Some of the disadvantages given are that the pit grinder does not produce as satisfactory a job as the floor type grinder, that the wheel treads are not ground concentric with journals, and that the driving machinery is in the way of pit work. Some master mechanics feel that a sufficiently rigid construction could not be obtained from a pit grinder, and for that reason the floor type grinder is best. Many roads grind their wheels before placing them in service, in which case the floor type grinder is of advantage. A provision for removing or caring for the dust removed from the wheels is important, and several railways are using an exhaust ventilating system. This improves working conditions considerably, so that the job does not become so annoying or unfavorable for the workmen.

Solid Wrought Carbon-steel Wheels for Electric Railway Service

(*A.E.R.E.A. Standard Specification*)

Process. 1. Steel for wheels shall be made by the open hearth process.

Discard. 2. A sufficient discard shall be made from each ingot to secure freedom from injurious pipes and undue segregation.

Chemical Composition. 3. The steel shall conform to the following limits in chemical composition:

	Acid steel	Basic steel
Carbon	0 60 to 0 80 per cent	0.65 to 0.85 per cent
Manganese	0 55 to 0.80 per cent	0 55 to 0 80 per cent
Silicon	0.15 to 0.35 per cent	0.10 to 0.30 per cent
Phosphorus, not over	0.05 per cent	0 05 per cent
Sulphur, not over	0 05 per cent	0.05 per cent

Sample for Chemical Analysis (Ladle Analysis). 4. An analysis of each melt of steel shall be made by the manufacturer to determine the percentages of the elements specified in Par. 3. This analysis shall be made from a test ingot taken during the pouring of the melt. The chemical composition thus determined, together with such identifying records as may be desired, shall be reported to the purchaser or his representative, and shall conform to the requirements specified in Par. 3.

Check Analysis. 5. A check analysis may be made by the purchaser from a wheel representing each melt. The chemical composition thus determined shall conform to the requirements specified in Par. 3. A sample may be taken from any one point in the plate; or two samples may be taken, in which case, they shall be on radii at right angles to each other. Samples shall not be taken in such a way as to impair the usefulness of the wheel. Drillings for analysis shall be taken by boring entirely through the sample parallel to the axis of the wheel; they shall be clean and free from scale, oil and other foreign substances. All drillings from any one wheel shall be thoroughly mixed together.

Finish. 6. (*a*) The wheels shall be free from injurious defects and shall have a workmanlike finish.

(*b*) Wheels shall not be offered for inspection if covered with paint, rust or any other substance to such an extent as to hide defects.

Workmanship. 7. The wheels shall conform to the dimensions specified within the following permissible variations:

Height of Flange. 7. (*a*) The height of the flange shall not vary from the dimensions specified more than 1/16 inch.

Thickness of Flange. 7. (*b*) The thickness of flange shall not vary from the dimensions specified more than 1/16 inch.

Radius of Throat. 7. (*c*) The radius of throat shall not vary from the dimensions specified more than 1/16 inch.

Thickness of Rim. 7. (*d*) The thickness of rim shall not vary more than 1/4 inch over nor more than 1/8 inch under that specified. The thickness of rim shall be measured from the inner edge of the rim to a base line drawn from the point of tangency of the throat and tread, parallel to the axis of the wheel.

Width of Rim. 7. (*e*) The width of rim shall not vary from the dimension specified more than 1/8 inch.

Thickness of Plate. 7. (*f*) The plate may vary in thickness, but the variation less than that specified shall not exceed 1/32 inch for each 1/8 inch in the thickness of the plate.

Limit Groove. 7. (*g*) When a limit-of-wear groove is specified, its location shall not vary more than 1/16 inch from that specified.

Diameter of Bore. 7. (*h*) The diameter of rough bore shall not vary from that specified more than 1/16 inch over or 1/8 inch under. When not specified, the diameter of rough bore shall be 1/4 inch less than the diameter of finished bore, subject to the above limitations.

Diameter of Hub. 7. (*i*) The diameter of hub shall not be less, but may be 3/4 inch more than that specified. The thickness of wall of the finished bored hub shall not be less than 1 inch at any point for bores 6 inches or under in diameter, nor less than 1 1/4 inches for bores over 6 inches in diameter, unless otherwise specified. The thickness of wall of the hub of any wheel shall not vary more than 3/8 inch at any two points equidistant from the face of the hub.

Length of Hub. 7. (*j*) The length of hub shall not vary from the dimension specified more than 1/8 inch.

Projection of Back Face of Hub. 7. (*k*) The projection of back face of hub shall not vary from the dimension specified more than 1/16 inch.

Black Spots on Hub. 7. (*l*) Black spots deeper than 1/16 inch will not be permitted in rough bore within 2 inches of either face of hub.

Eccentricity of Bore. 7. (*m*) The eccentricity between the axis of the tread and the axis of the rough bore shall not exceed 3/64 inch.

Block Marks on Tread. 7. (*n*) Block marks shall not exceed 1/64 inch in height.

Rotundity. 7. (*o*) Wheels shall be gaged with a ring gage and the opening between the gage and tread, at any point, shall not exceed 1/16 inch.

Plane. 7. (*p*) Wheels shall be gaged with a ring gage concentric with and perpendicular to the axis of the wheel.

(*b*) For all points on the back of the rim equidistant from the center, the variation from the plane of the gage when so placed shall not exceed 1/16 inch.

Tape Sizes. 7. (*q*) Wheels with treads under 3 inches in width shall not vary more than 6 tapes over nor more than 4 tapes under the size specified. Wheels with treads 3 inches or over in width shall not vary more than 9 tapes over nor more than 5 tapes under the size specified.

Mating. 7. (*r*) The wheels shall be mated as to tape sizes and shipped in pairs. The tape size shall be legibly marked on each wheel.

Gages and Tapes. 8. The manufacturer shall provide suitable gages and tapes which shall conform to the contour and dimensions specified.

Branding. 9. Wheels shall be stamped with the manufacturer's mark, date, heat number and serial number, in such manner that each wheel may at any time be readily identified.

Inspection. 10. (*a*) The inspector representing the purchaser shall have free entry, at all times while work on the contract of the purchaser is being performed, to all parts of the manufacturer's works which concern the manufacture of the wheels ordered. The manufacturer shall afford the inspector, free of cost, all reasonable facilities to satisfy himself that the wheels are being furnished in accordance with this specification. Tests and inspection shall be made prior to shipment.

10. (*b*) The purchaser may make the tests to govern the acceptance or rejection of material in his own laboratory or elsewhere. Such tests, however, shall be made at the expense of the purchaser.

10. (*c*) All tests and inspections shall be so conducted as not to interfere unnecessarily with the operation of the works.

10. (*d*) Wheels which show injurious defects while being finished by purchaser, will be rejected and the manufacturer shall be notified.

10. (*e*) Unless otherwise arranged, any rejection based on tests made in accordance with Par. 10 (*b*) shall be reported within ten working days from the receipt of samples.

10. (*f*) Samples tested in accordance with Par. 10 (*b*) which represent rejected wheels shall be preserved for one month from the date of the test report. In case of dissatisfaction with the results of the tests, the manufacturer may make claim for a rehearing within that time.

Standard Wheel Circumference Measure (*A.R.A.*). The measure for tape sizes, referred to in above wheel specifications, is shown in Fig. 20. To use this standard measure, place tape about circumference of wheel, having the brackets in contact with flange. The normal circumference for wheels of each of the different diameters is indicated by the space marked "3." According to the A.R.A. specifications, steel and steel-tired wheels should be rejected if the scribed lines on the head piece fall to the left of the space marked "C" or to the right of the space marked "½"; cast iron wheels should be rejected if the scribed line on the head piece falls to the left of the space marked "1" or to the right of the space marked "5." The continuous markings on the upper side of the tape, as shown in the figure, may be used for mating worn wheels. The linear dimensions shown in figure represent measurements of actual circumference of the wheel and not straight length of the tape. Graduations are spaced ⅛ in. apart with the tape laid flat, and hence a variation of one tape size means approximately a variation of ⅛ in. The tape size of a wheel is determined by the space in which the circumferential measure falls; for example, the space between lines 157 and 158 on the upper side of the tape coincides with the space on the lower side of the tape, representing Tape Size No. 3 for the 33 in. diameter wheel.

Rolled Steel Wheels. The general replies from fifty operating companies to questions regarding solid steel wheels sent out by the Committee on Equipment, A.E.R.E.A., were as follows: The diameters when new range from 30 to 37¼ in. New rims are from 2 to 3¾ in. thick. These are worn down by some companies to as low as ½ in. Others wear down to 1¼ in., while the average considers ¾ in. to be safe. The average number of turnings is three. Solid rolled steel wheels are considered safer than steel-tired wheels for interurban service. They are also held to be more economical, scrap considered.

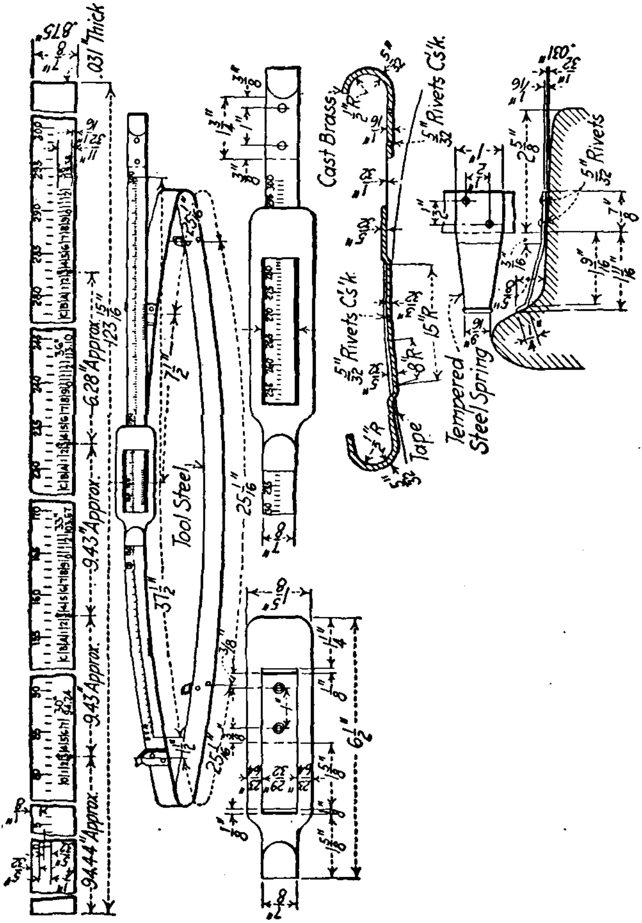


Fig. 20.—A.R.A. standard wheel circumference measure. (Tape gage.)

Wheel Turning. The necessity for turning is most often determined by flange wear. A general statement of the flange problem is as follows: When a train is moving on tangent track, the flanges of the wheels, due to lateral oscillation of the train, strike against the rails alternately on one side or the other; when the train moves on curved track, the wheel flanges are forced against either the inner or outer rail depending on the speed and the amount of superelevation of the outer rail. In both cases the car wheel flanges in contact with the rail are factors in transmitting the various forces set up within the car to the rail. Since the flange pressures produced in traversing curves are the more severe, the pressures arising on tangent track need not be considered. First, the pressure or force between rail and flange sets up internal strains and stresses within the body of the wheel; in the case of a wheel subjected to severe operating conditions, the internal stresses may result in the failure of the wheel. Second, the pressure between the rail and flange coupled with the rotation and sliding of the wheel on the rail results in a grinding or wearing action and in consequent reduction in area and strength of the flange. Third, the rail pressure, when of sufficient magnitude and if acting on a flange materially worn, may cause the failure of the flange. The necessity for turning may also be determined by flat spots on the tread or by the allowable difference between the circumferences of two wheels rigidly fastened to the same axle or geared or linked together. This allowable difference is often taken to be ⅛ inch in ordinary electric railway work, and a consideration of it is of greater importance on driving wheels than on trailers.

New York subway wheels are turned in two Pond center-drive lathes. Of these one is driven by a 20-h.p. motor and the other. which is one of a newer and heavier type of construction, by a 40-h.p. motor. The latter is equipped with air operated tail stocks and both have the cutting tools clamped in the tool post by means of compressed air. The output of the former lathe is twelve pairs of wheels per day, while the latter machine has a capacity of twenty pairs per day. The great stiffness and pulling power of the latter machine permits the finished cut on the wheels to be taken with a forming tool which gives proper contour with one cut.

A. B. Creelman of the Youngstown Municipal Railway reports the average life of steel wheels obtained without turning to be 150,000 miles. This high mileage is obtained by careful attention to various factors affecting wheel wear beginning with the initial mounting of the wheels on the axle. Wheels of exactly the same tread diameter are paired, axles are trued and straightened, and both the wheel and gear seats are turned, if necessary, to secure full bearing. The trucks are squared and brake rigging placed in first class shape with the brake shoes hung to eliminate any side thrust on the trucks when the brakes are applied. In operation the brake rigging is kept in line, and all lost motion is taken up as it develops. Journal brasses are renewed before the side motion or end play becomes excessive. The axles are kept square in the truck frame by rearranging the shims in the journal box pedestals. Wheels that begin to show flange wear on single end cars are removed to the other end of the car, thus balancing wheel wear such as is obtained on double end cars.

Other companies report various mileages between turnings ranging from 20,000 to 65,000 miles, with ¾ in. to 1 in. as average material turned off, and three to five average turnings in life of wheel.

Welding Worn Flanges. Satisfactory results have been obtained by building up worn flanges by means of a welding process, as follows: The wheel with the worn flange is turned, removing 1/4 in. of metal from the tread, and in so doing, 1/4 in. from the new flange at the base. The top of the flange is finished to proper contour, which leaves a right angle groove in the flange about 3/8 in deep, and parallel to the tread, and 1/2 in. deep at right angles to the tread for a 1 in. flange. The flange is then filled out, using a 1/4 in. welding rod, and turned to finish. Most all steel wheels are taken out of service on account of sharp flanges, and usually there will be one sharp flange on each axle, while the other will have become larger than the original. The latter wheel is turned to the same diameter and a new flange formed, unless both wheels have a sharp flange, in which case both flanges are welded. Experience shows that a filled flange will last as long as a turned flange.

Mounting Wheels. Wheels with flanges worn to or below the limit of wear, as shown by the standard gages, should not be remounted. The flange thickness of wheels fitted on the same axle should be equal and never vary more than 1/16 in. In mounting wheels, new or second-hand, the standard wheel mounting and check gage (*Fig. 21*) should be used, pressing on one wheel so that it is the proper distance from the center line of the axle by having the pointer of the gage at the center line of the axle and the end pieces of the gage resting only on the tread and contour of the wheel at the gage line, allowing enough clearance to take care of the tolerance variation of the wheel. Then the other wheel is pressed on in the same manner, and by proper mounting and use of the gage should thus place the wheels at the proper distance from each other and also locate them centrally on the axle. The wheel seats on all axles must be turned to a uniform diameter throughout the entire length of each wheel seat, and must be smooth and free from ridges, so as to provide an even bearing for the wheel fit throughout. The mounting of wheels on axles which have the wheel fit tapered is not permissible. Mounting presses should be provided with recording pressure gages, and all wheels not mounted within the limits given, or wheels that are forced against the shoulder, are to be withdrawn.

Mounting Pressure. The A.R.A. recommended practice is shown in the following table.

MOUNTING PRESSURES, IN TONS

A.R.A. axle	Wheel seat diameter	Cast iron wheels		Steel wheels	
		Minimum	Maximum	Minimum	Maximum
A	5 1/8 in	30	45	45	60
B	5 3/4 in	35	50	50	70
C	6 1/2 in	40	60	60	80
D	7 in.	45	65	65	85
E	7 5/8 in	50	70	70	95

Removing Wheel from Axle. The pressure ordinarily required to remove a wheel from its axle will vary from two to five times that at which the wheel was mounted. The wheel press is often used to apply this pressure which amount generally lies between 150 tons and 300 tons. Heating the wheel and hammering while the pressure is applied are often resorted to in starting the wheel. A drop hammer has also been used alone to start the wheel which was afterward removed by the wheel press.

Shrink Fits. The process of shrinking wheels and gears to axles and tires to wheels is outlined in the following citation of the practice on the New York, Westchester and Boston Railway by R. R. Potter: The parts are bored accurately to a certain diameter in relation to the axle, micrometers being used in measuring. The part is then heated and placed on the axle without pressure. By this method there is no abrasion on the finished surfaces of the axle, wheel or gear due to the process of application. Also, as heat is applied to the part when it is removed, very little abrasion occurs, due to the process of removal. In addition, the parts may be replaced in their original positions a number of times, without loss of holding power. The axles, at the location of wheel seats, are carefully finished and filed as smooth as possible. Wheels and gears as well as tires are bored 0.001 in. smaller than the axle or wheel center per inch of diameter of bore, and then they are heated and slipped into place. After the wheels have been applied and have cooled, they are tested in the wheel press at a pressure of 10 tons per inch diameter of bore. The gears are heated at center with a torch using compressed air and city gas, but the rims are not allowed to go above 300 deg. F., as measured with a thermometer set with the bulb in a special cup which fits between the gear teeth. It is found that this temperature invariably gives sufficient expansion to permit gears to be slipped on without any pressure. Wheels are heated with the same torch until a drop of water at the middle of the spoke will boil freely when dropped onto it. Wheels sometimes require a slight pressure in addition to the heating to move them into place, but this is negligible in comparison with that used in ordinary press fits. When wheels or gears are thus shrunk onto the axles, the microscopic points of metal are not smoothed down as they are when a wheel is pushed on cold, and the result is that these points interlock and make a very close and secure fit.

Holding Power of Steel Tire. In the New York subway where wheels are heated excessively by frequent high acceleration and heavy braking, the highest temperature of the tire of a wheel just taken from service was found to be 145.4 deg. F., and the corresponding temperature of the hub was 87.8 deg. F. After this was investigated, the holding power of steel tires shrunk on with the standard shrinkage allowance of 0.001 in. per 1 in. diameter was calculated. The results are shown by the following table:

Shrinkage allowance	Temp. of tire	Temp. of hub	Radial pressure of tire on wheel center	
			Tire 2⅝ in. thick	Tire 1⅛ in. thick
0.03 in.	60° F.	60° F.	6070 lb. per sq. in.	2600 lb. per sq. in.
0.03 in.	145° F.	88° F.	3970 lb. per sq. in.	1700 lb. per sq. in.

Percentage reduction in holding power caused by wear, 57 per cent.
Percentage reduction in holding power caused by heat, 35 per cent.
Percentage reduction in holding power caused by wear and heat, 92 per cent.

Life of Wheels. The actual mileage obtained from wheels varies through quite a large range, depending on service conditions. The wear of wheels resulting from rolling on the rails is very small. Brake shoes wear away the metal more rapidly, and the loss from chipped and broken flanges, the grinding out of flat spots on chilled iron wheels, and the turning down of steel wheels for sharp flanges, account for much of the wheel consumption. Thus the number of stops made has an important bearing on wheel wear, and likewise the character of the line as regards grades and curves. The amount of special trackwork is also an important factor in the wheel life. With the same depth of wear, the larger the wheel the greater the amount of metal which can be worn off, and hence increased mileage for the life of wheel. In interurban service, with few stops, a greater mileage can be expected from the same type of wheel than in city service.

	Mileage	Average mileage
City service—chilled iron wheels	35,000–60,000	49,000
City service—steel wheels	70,000–240,000	137,000
Interurban service—steel wheels	200,000–300,000	250,000

The above table was compiled from a survey of sixty electric railways by the *Electric Railway Journal*. By very careful attention to inspection and maintenance, the maximum mileages can be increased.

The use of steel wheels is increasing. Many railways are not equipped for proper maintenance of steel wheels and continue the use of chilled iron wheels rather than go to the expense of purchasing new shop equipment. Some railways have found that, due to track conditions, and where motors hang low, they cannot obtain the full wear from steel wheels, and hence the chilled iron wheel may be the more economical. The great advantage emphasized by most users of steel wheels is the freedom from chipped flanges, and from the trouble of removing flat spots from cast iron wheels, as slight flat spots in steel wheels will roll out in a few days running without requiring attention at the shop. Steel wheels usually are shopped on account of worn flanges, and as this condition comes gradually to steel wheels, they can be shopped at convenience and the work bunched to reduce maintenance cost. On the other hand, cast wheels fail one at a time and must be shopped at once. Steel wheels weigh less compared to cast iron wheels, and a slight saving in energy results.

Wheel Defects. The following relative to common wheel defects is from a talk by F. A. Beebe of the Griffin Wheel Co. at the Hartford Shops of the Connecticut Co.:

Thin Flange. A thin flange may be due to one or more of the following causes: (*a*) Wheels improperly mated as to diameters. (*b*) Improperly mounted as to gage. (*c*) Improperly put on axles; that is, the distance from end of journal to gage line on one end of axle being greater than the other. (*d*) Improperly shaped flange. (*e*) Trucks out of true. (*f*) Car riding on side bearings, preventing free movement of the truck, bringing the work of turning on the

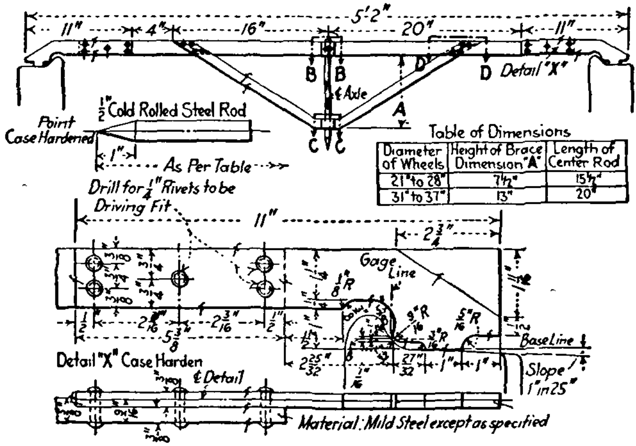


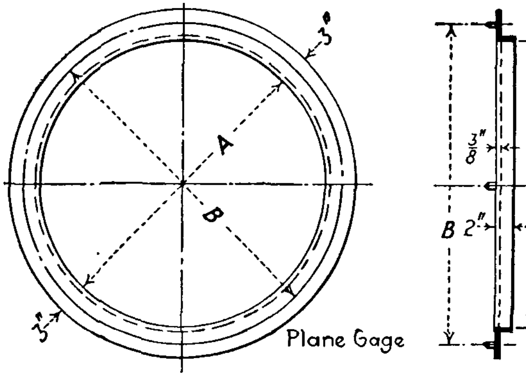
FIG. 21.—A.E.R.E.A. standard wheel mounting and check gage.

flange. (*g*) Track out of gage on curves. (*h*) Improper elevation of outer rail. (*i*) Running car in one direction; that is, when car at the end of the line runs around a loop instead of being turned. (*j*) Sanding one rail, causing one wheel to wear down faster than its mate.

Warped Flange. A cast iron wheel flange warpage of about 1/16 in. is allowable.

Elimination of Flat Spots. (*a*) See that the brake shoes are properly adjusted. (*b*) Braking apparatus adjusted so that in coming to a stop the wheel will not lock. (*c*) See that the percentage of brake pressure is in proper relation to the load on the wheels. (*d*) An important point is the manner of hanging the brake. Most brake hangers are placed below a line passing through the center of the axle, parallel with the trucks, which allows the brake to fall away from the wheel by gravity. This angle may be made too large, so that with badly worn and loose parts the application of the brakes causes the shoe to crowd up against the wheel. This makes a toggle joint, producing excessive pressure on one pair of wheels in the truck causing them to stop rolling while the other pair is still revolving. This is a prolific cause of slid flats and also results in excessive wheel wear.

Wheel Gages. Fig. 21 shows the A.E.R.E.A. Standard Wheel Mounting and Check Gage. This gage is made with a pointer for



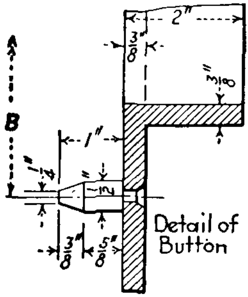


Fig. 22.—A.E.R.E.A. standard plane gage. Measurements A and B vary depending on diameter of wheel to be gaged.

the center line of the axle so that the first wheel may be pressed on the proper distance from the center line of the axle and the wheels will not only be located the proper distance from each other, but will be located centrally on the axle. The end pieces making contact with the wheels are machined so that they rest only on the tread and contour of the wheel at the gage line, allowing enough clearance to take care of the tolerance variation of the wheel. The pointer placed at the center has a thumb screw with which to make adjustments for differences in the diameter of the wheels. In mounting and gaging wheels it is understood that the gage line is at the point on the fillet of the flange ¼ in. below the wheel tread, and wheels should be gaged ¼ in. narrower than the gage of the tracks; the track gage being measured between points ¼ in. below the tops of the rails.

The A.E.R.E.A. Standard Plane Gage for Solid Wheels is shown in Fig. 22. This is used to check the warped condition of wheels by placing it on the back of the rim. The tolerances allowed will be governed by the wheel specifications.

The A.E.R.E.A. Standard Rotundity Gage for Solid Steel Wheels is shown in Fig. 23. As all standard wheels have a taper of 1 in. in 20 in., in checking the wheel, the gage is used that will become tight on the wheel at about the center of the tread. The tolerance governing this is included in the wheel specifications

Limit of Wear Gage. (*A.E.R.E.A. Standard.*) Designs for limit of wear gages for the wheel contours which were adopted as standard by the Am. El. Ry. Eng. Assn. in 1923 are at this writing in preparation by the Equipment Committee of that Association; these probably will be adopted as standard at the 1924 convention, and thereafter be shown in the A.E.R.E.A. Manual. To use the limit of wear gage, it is fitted down over the flange contour, and if the gage touches the tread of wheel and the guard side of the flange contour, and does not encounter the flange contour at any other

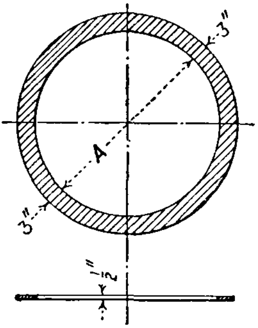


FIG. 23.—A.E.R.E.A. standard rotundity gage. Measurement A varies depending on diameter of wheel to be gaged.

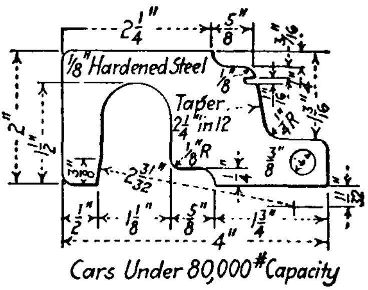


FIG. 24.—A.R.A. standard limit of wear gage.

point of the gage, the flange has been worn to the limit. In this case the flange is too thin or the angle of the flange is too sharp. If the gage touches at the top of the flange and not the tread of the wheel, the flange height is too great.

Fig. 24 shows limit of wear gage for A.R.A. Standard Tread Cast Iron Wheels for cars under 80,000 lb. capacity. Similar gages are used for cast iron wheels on cars of over 80,000 lb. capacity and for steel wheels.

A.E.R.E.A. Standard Wheel Contours. (Fig. 25.) Investigation of the conditions of track, rail sections, etc., brought out that the widths and depths of wheel flanges, measured on the level of the tread, which appear to best cover all ordinary requirements, are 1 3/16 in. width in combination with 7/8 in. depth, 1 1/8 in. and 1 in. widths in combination with 3/4 in. depth, and 1 in. width in combination with the 5/8 in. depth. A wide tread is desirable for minimum wear on track, particularly in special trackwork, and as practically all A.E.R.E.A. standard sections of rail permit the use of a wheel tread 3 in. wide, it is recommended that this should be given the preference to the gradual elimination of the narrower wheel treads. Existing conditions in many localities, however, do not now permit a wheel tread wider than 2½ in., and the latter therefore is also recommended as an alternate standard in connection with the ¾ in. and ⅝ in. depth of flanges. For interurban lines using the ⅞ in. by 1 3/16 in. flange the 3½ in. width of wheel tread is recommended. The taper of wheel tread in common use formerly differed between steel wheels and chilled iron wheels, but for the future the same taper of 1 in 20 is adopted for both. The rounding off of the outer edge of the wheel tread in present practice varies greatly. A certain amount of rounding is desirable to prevent chipping or flowing out of the metal, but it should be confined to the necessary minimum so as to not unduly reduce the

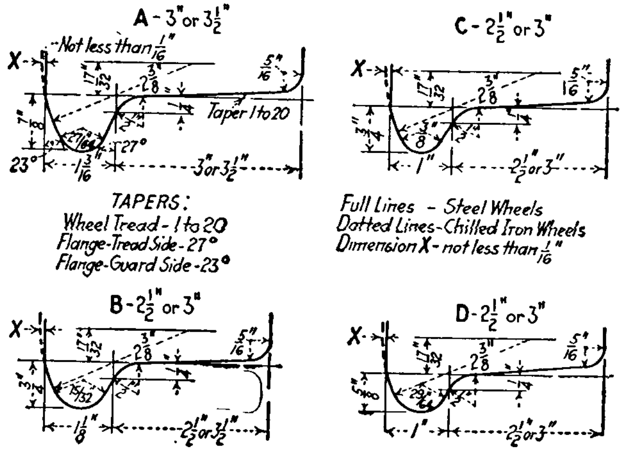


FIG. 25.—A.E.R.E.A. standard tread and flange contours for steel and chilled iron wheels.

width of the effective wheel tread. It has been made 5/16 in. uniformly in all standard A.E.R.E.A. contours.

The curved contours of wheel flanges now in use vary greatly for the same width and depth of flange without apparent regard to the track and wear conditions. For given or assumed fixed conditions that affect the wear on wheel flanges, *i.e.*, dimensions of trucks or running gears, rail sections used, and proportionate amount of straight and curved track of various degrees of curvature, the essential portions of wheel flange contours best suited to these conditions can be theoretically and graphically determined. For straight track of standard girder guard rails and the usual play between wheel gage and track gage, the flanges would have straight sides with a minimum angle from the vertical. On curves this angle increases with the curvature and wheel base until maximum practical conditions are reached. As the same flange must answer in average as well as extreme conditions, the contour must follow a line that takes this into account. The hyperbolic curves produced by the horizontal section through straight side flanges must be modified at different heights according to the

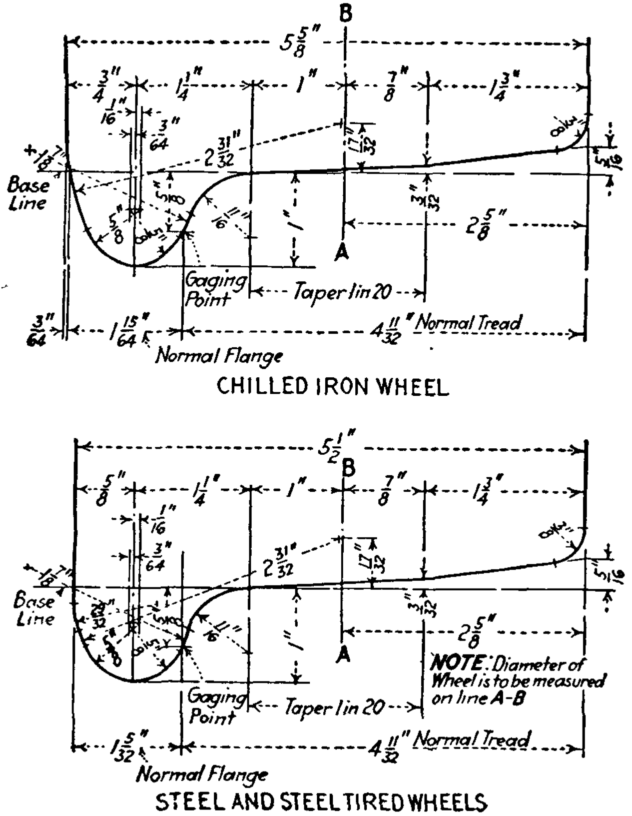


FIG. 26.—A.R.A. standard wheel contours.

curves to be run through, the aim being to give as good wearing surface as possible under the varying conditions, to reduce the wear, and to give points of contact on both the gage side and guard side as high up on the flange as practicable to reduce the leverage of frictional resistance. Average extreme conditions must be the principal guide. The graphical development under various conditions of wheel base and radius of curve, in combination with maximum diameter of wheel of 33 in., produces practically straight lines through the region of contact between wheel flange and rail. These lines for the standard girder guard rail have a practically uniform inclination from the vertical of approximately 27 deg. of angle on the gage side and 23 deg. of angle on the guard side. Theoretically, no rounding off of the nose of the flange below the points of contact is required, and the flange could be carried down on a continuation of the above angles to a sharp corner at its extreme depth. Practical experience in the operation of the cars on track, with a certain amount of unavoidable irregularities and possible obstructions in the flangeway, and the requirements of wheel manufacturers, make a maximum rounding of the nose desirable. Any flattening of the nose would increase the tendency of the flanges to chip and to strike frog points or other irregularities in the track. The rounding adopted for the four standard flanges is, therefore, the maximum permitted by the contour lines, but still leaves sufficient line or area of contact against the rail on curves. For the radius of the fillet joining the contour of flange to the tread, the average practice has been followed, and the radius made 3/8 in. for all flanges. Should a compound fillet, giving a partially curved tread, be decided upon or considered desirable in the future, the necessary modifications can be made easily, the effect on the horizontal position of the gage point being negligible.

On rolled steel wheels it is impractical in manufacture to produce a curved contour for the back of the flange above the tread line. The contours of the back of steel wheels above the tread line follow a straight vertical line, the exact contour above tread line not being essential. In chilled iron wheels it is desired from a manufacturing standpoint that the amount of material above the flange proper be increased and the standard wheel contours have such an increase by a continuation of the curved lines of the back of the flange above the tread line. Such continuation on a circle of the same radius as the back of the flange below the tread line gives *approximately 1/16 of an inch extra width at a height of 7/32 of an* inch above the tread line. The continuation of the back, however, may be made on any larger radius or on a straight line tangent to the contour at the point of intersection with the tread level, and the width above the tread line be increased further, and can be left optional with the manufacturer. This difference between the steel wheels and the chilled iron wheels does not in any way affect the action of wheels on the track and the wheel flange proper below the tread line is the same for both.

The A. R. A. standard flanges for steel wheels and cast iron chilled wheels follow the same principles. (Fig. 26.)

Standard Dimensions for Steel Wheels (A.E.R.E.A.)

Diameters, 21 in. to 36 in., inclusive

Flange contour	A	A	B	B	C	C	D	D
Tread	3"	3 1/2"	2 1/2"	3"	2 1/2"	3"	2 1/2"	3"
Flange height	7/8"	7/8"	3/4"	3/4"	3/4"	3/4"	5/8"	5/8"
Flange thickness	1 3/16"	1 3/16"	1 1/8"	1 1/8"	1"	1"	1"	1"
Rim width	4 3/16"	4 11/16"	3 5/8"	4 1/8"	3 1/2"	4"	3 1/2"	4"
Rim thickness*	2" (31 −) 2 1/2" (33 +)	2" (31 −) 2 1/2" (33 +)	2" (31 −) 2 1/2" (33 +)	2" (31 −) 2 1/2" (33 +)	2" (31 −) 2 1/2" (33 +)	2" (31 −) 2 1/2" (33 +)	2" (31 −) 2 1/2" (33 +)	2" (31 −) 2 1/2" (33 +)
Hub diameter*	7" (21) 7 1/2" (22, 24) 8 1/4" (26, 31) 9 1/4" (33 +)	7" (21) 7 1/2" (22, 24) 8 1/4" (26, 31) 9 1/4" (33 +)	7" (21) 7 1/2" (22, 24) 8 1/4" (26, 31) 9 1/4" (33 +)	7" (21) 7 1/2" (22, 24) 8 1/4" (26, 31) 9 1/4" (33 +)	7" (21) 7 1/2" (22, 24) 8 1/4" (26 +)	7" (21) 7 1/2" (22, 24) 8 1/4" (26 +)	7" (21) 7 1/2" (22, 24) 8 1/4" (26 +)	7" (21) 7 1/2" (22, 24) 8 1/4" (26 +)
Hub length*	4 1/2" or 5" (31 −) 6" (33 +)	4 1/2" or 5" (31 −) 6" (33 +)	4 1/2" or 5" (31 −) 6" (33 +)	4 1/2" or 5" (31 −) 6" (33 +)	4 1/2" or 5"	4 1/2" or 5"	4 1/2" or 5"	4 1/2" or 5"
Hub projection* 4' 8 1/2" gage	2 15/16" (31 −) 2 15/16" or 1 15/16" (33 +)	2 15/16" (31 −) 2 15/16" or 1 15/16" (33 +)	3" (31 −) 3" or 2" (33 +)	3" (31 −) 3" or 2" (33 +)	3 1/8"	3 1/8"	3 1/8"	3 1/8"
Hub projection* 5' 2 1/2" gage	3 15/16" (31 −) 2 15/16" or 1 15/16" (33 +)	3 15/16" (31 −) 2 15/16" or 1 15/16" (33 +)	4" (31 −) 3" or 2" (33 +)	4" (31 −) 3" or 2" (33 +)	4 1/8"	4 1/8"	4 1/8"	4 1/8"

* Figures in parentheses indicate wheel diameter, in inches. Minus sign indicates "or under." Plus sign indicates "or over." Two figures, include intervening sizes.

Advantages of Large Wheel. Following are some of the advantages of the large wheel: It affords greater mileage between turnings, gives lower journal surface speed with consequently less wear and axle steel fatiguing stresses per car mile, causes smaller stresses to be set up in wheel and rail surfaces, strikes lighter blows on the track and is less likely to give trouble at a defective frog, switch point or rail. A large wheel makes possible the use of a large gear and maximum gear reduction with a corresponding high motor speed which permits the use of a comparatively small light weight motor and necessitates the least energy consumption. Special installations may be such that the above advantages would be inferior to the advantages accompanying the use of small wheels. Thus, the satisfactory design of many low floor cars has demanded wheels of small diameter.

Truck Wheel Base. The use of 4 ft. wheel base trucks for purely city service is desirable only by reason of the ease with which they may be operated over short radius curves. The inside hung motor truck of an approximate 6 ft. wheel base is superior to outside hung motor trucks of short wheel base for the following reasons: the lesser displacement of tractive effort between the two pairs of wheels during acceleration and braking; greater factor of strength on account of the location of motor suspension in close proximity to the center of truck; less wheel flange pressures against rail heads required to swivel trucks on curves, and also occurring during "nosing" periods; and easier riding qualities. There is a less liability to derail, less flange wear, less tendency to cause rail corrugations, less wear on track, and less maintenance, with the long wheel base truck. Interurban service uses the long wheel base truck, which is best for high speed operation. In a paper before the Central Electric Railway Association, A. C. Vauclain of the Baldwin Locomotive Works stated that the final determination of the wheel base should, if curves permit, be based entirely upon the transom width being sufficient to allow a proper width of bolster and bolster springs, and the fixed length of the motors measured from their axle journals to their supporting noses.

Minimum Curve for Given Truck. Fig. 27 by Graham Bright, *Electric Journal*, 1911, shows the relation between wheel size, wheel base and minimum curve. These curves are plotted according to the expression

$$R = \frac{W}{2 \sin a} \quad \text{(published by the Baldwin Locomotive Works)}$$

in which R = radius of sharpest curve that can be passed, feet.
W = wheel base, feet.
a = angle which flanged wheels make with rail

With diameter of wheel	20 in. to	24 in	$\sin a = 0.117$
	25	30	$= 0.107$
	31	40	$= 0.090$
	41	50	$= 0.080$
	51	60	$= 0.075$

Ball Bearing Center Plates. The principal advantages of ball bearing center plates over smooth plate bearings are that their use is accompanied by a minimum of flange wear and energy consumption on curves. They bring about less liability to derailment and require less lubrication.

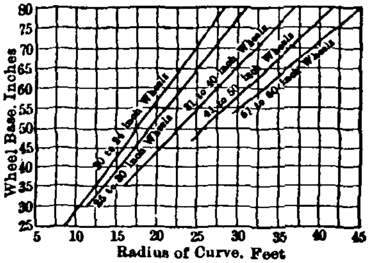


FIG. 27.—Relation between minimum curve, wheel diameter and truck wheel base.

DIMENSIONS OF A.E.R.E.A. STANDARD JOURNAL BOXES

(See Fig. 28)

Journal size	3″×6″ 3¼″×6″	3¾″×7″	4¼″×8″	5″×9″	5½″×10″	Journal size	3″×6″ 3¼″×6″	3¾″×7″	4¼″×8″	5″×9″	5½″×10″
Dimension	In.	In.	In.	In.	In.	Dimension	In.	In.	In.	In.	In.
1	4 1/4	4 3/4	4 3/4	4 3/4	4 3/4	23	6 3/16	6 1/16	6 1/16	7 3/8	7 3/8
2	5 1/2	6 13/16	6 13/16	8 13/16	8 13/16	24	5 7/16	5 9/16	5 9/16	6 5/16	6 5/16
3	5 1/4 D	6 5/8 D	6 5/8 D	7 5/8 D	7 5/8 D	25	5 1/16	7 1/4	7 1/4	7 1/4	7 3/4
4	3/4	1	1	1	1	26	3 7/16	3 15/16	4 5/8	5 1/8	5 11/16
5	3/4	3/4	7/8	1	1	27	4 1/8	4 3/8	4 7/8	5 5/8	5 7/8
6	1 1/8	1 1/4	1 1/2	1 7/8	2	28	....	3 1/4	3 7/8	3 7/8	4 1/4
7	3 7/8	4 21/32	5 1/32	5 25/32	6 9/32	29	2 7/8	3 1/4	3 3/4	4 3/8	4 1/2
8	3 7/8	4 9/16	4 15/16	5 11/16	6 3/16	30	2 3/8	2 3/8	2 3/8	2 3/4	2 3/4
9	1 1/16 D	1 1/16 D	1 1/4 D	1 1/4 D	1 3/8 D	31	2 3/4	3 3/8	3 3/8	3 13/16	4 1/16
10	7	8	8	9	9 1/2	32	4 5/16	4 11/16	5 5/16	5 5/8	5 13/16
11	1R	1R	1 1/8 R	1 1/8 R	1 3/16 R	33	1	1 1/4	1 1/4	1 3/8	1 3/8
12	6 1/2	7 3/8	7 3/8	8 3/4	8 3/4	34	7/8	1	1	1 1/8	1 1/8
13	4 1/4	4 3/4	4 3/4	4 3/4	4 3/4	35	1 7/8	2 1/8	2 1/8	2 5/8	2 5/8
14	3 1/4	4 3/8	4 3/8	5 5/8	5 5/8	36	3 1/2	4 1/8	4 5/8	5 1/4	5 3/4
15	5 1/4	6 3/8	6 3/8	7 5/8	7 5/8	37	3 7/16	3 13/16	4	4 1/2	5
16	4 1/2	5 5/8	5 3/4	6 5/8	7 1/8	38	2 7/16	2 7/8	3 1/8	3 11/16	4
17	3 3/4	4 3/8	4 3/8	5 3/8	5 3/8	39	1 1/16	3/4	3/4	3/4	3/4
18	5 1/2	6 13/16	6 15/16	8 15/16	8 15/16	40	4 1/2	4 3/4	5 1/4	5 7/8	6 1/8
19	4 1/8	5	5 1/2	6 3/4	7 1/4	41	3/8	3/8	7/16	7/16	1/2
20	6	7 1/4	7 1/2	8 3/8	9	42	6 1/8	8 5/8	8 5/8	9	9 1/2
21	1	1 1/4	1 1/4	1 1/4	1 1/2	43	4 7/8	5 5/16	5 1/2	6 1/16	6 5/8
22	3 3/8	3 3/4	3 3/4	3 3/4	4						

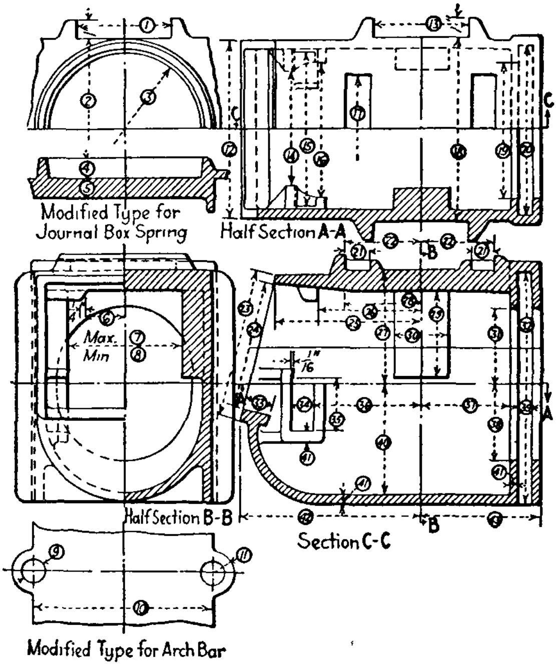


FIG. 28.—A.E.R.E.A. standard journal boxes.

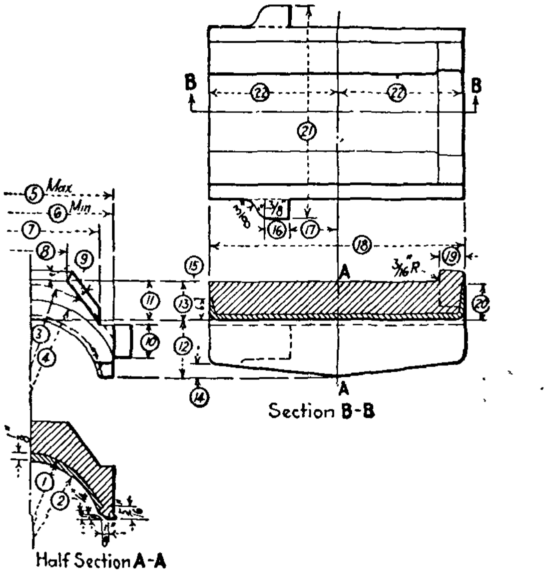


FIG. 29.—A.E.R.E.A. standard journal bearings.

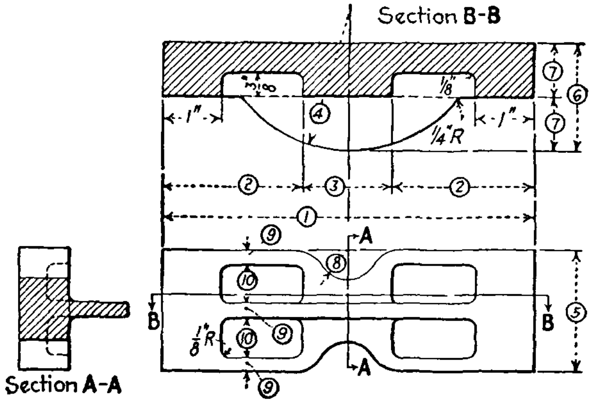


FIG. 30.—A.E.R.E.A. standard journal thrust plates.

DIMENSIONS OF A.E.R.E.A. STANDARD JOURNAL BEARINGS

(See Fig. 29)

Journal size	3″×6″	3¼″×6″	3¾″×7″	4¼″×8″	5″×9″	5½″×10″
Dimension	In.	In.	In.	In.	In.	In.
1 and 2	1 17/32 R	1 21/32 R	1 29/32 R	2 5/32 R	2 17/32 R	2 25/32 R
3 and 4	2R	2R	2 7/16 R	2 11/16 R	2 17/32 R	3 3/8 R
5	3 13/16	3 3/16	4 17/32	4 29/32	5 21/32	6 5/32
6	3 13/16	3 3/16	4 1/2	4 7/8	5 5/8	6 1/8
7	3 1/4	3 1/4	3 3/4	4 1/8	4 3/4	5 3/8
8	1 3/4	1 3/4	2	2 5/16	3 1/8	3 1/8
9	1/4	1/4	1/4	3/8	3/8	3/8
10	3/4	3/4	7/8	1 3/16	1 1/16	1 1/16
11	1	1	1 1/8	1 3/16	1 1/8	1 5/16
12	1 1/2	1 1/2	1 1/2	1 7/8	2 1/4	2 3/8
13	1	1	1	1	1 1/8	1 1/8
14	5/8	5/8	5/8	5/8	5/8	5/8
15	3/8	3/8	9/16	9/16	5/8	5/8
16	1/2	1/2	1/2	1/2	5/8	5/8
17	1 1/4	1 1/4	1 1/4	1 1/4	1 7/16	1 7/16
18	5 3/4	5 3/4	6 3/4	7 3/4	8 3/4	9 3/4
19	5/8	5/8	1 1/16	5/8	5/8	5/8
20	3/4	3/4	1 5/16	1 5/16	1	1
21	4 11/16	4 11/16	5 1/2	5 7/8	6 7/8	7 3/8
22	2 7/8	2 7/8	3 3/8	3 7/8	4 3/8	4 7/8

DIMENSIONS OF A.E.R.E.A. STANDARD JOURNAL THRUST PLATES

(See Fig. 30)

Journal size	3″×6″ 3¼″×6″	3¾″×7″ 4¼″×8″	5″×9″ 5½″×10″
Dimension	In.	In.	In.
1	5 1/8	6 1/4	7 1/2
2	1 5/16	2 3/8	3
3	1 1/4	1 1/2	1 1/2
4	1 3/4 R	2 1/4 R	3 1/4 R
5	1 3/4	2	2 1/2
6	1 1/2	1 3/4	2
7	3/4	7/8	1
8	1/2 R	1/2 R	5/8 R
9	1/4	1/4	5/16
10	1/2	5/8	2 5/32

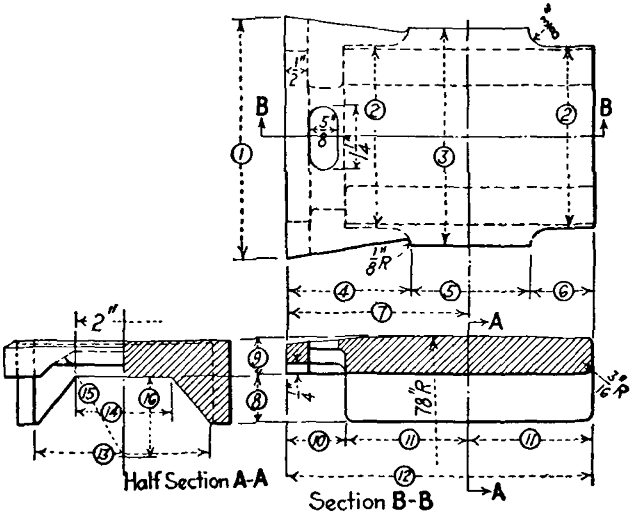


FIG. 31.—A.E.R.E.A. standard journal wedges.

DIMENSIONS OF A.E.R.E.A. STANDARD JOURNAL WEDGES
(See Figs. 31, 32, 33)

Journal size	3"×6" 3¼"×6" (Fig. 31)	3¾"×7" (Fig. 31)	4¼"×8" (Fig. 32)	5"×9" (Fig. 33)	5½"×10" (Fig. 33)
Dimension	In.	In.	In.	In.	In.
1	4½	5	5	6¾	6¾
2	3¼	3¾	4½	5¼	5¾
3	3¾	4½	4⅞	5⅝	6⅛
4	2 3/16	2⅝	3 5/16	3⅝	4 3/16
5	2⅜	2½	2½	2⅞	2⅞
6	1	1⅜	1 15/16	2¼	2¾
7	3⅜	3⅞	4 9/16	5 1/16	5⅝
8	⅞	1	1 1/16	1 1/16	1 3/16
9	¾	¾	⅞	1	1
10	1 3/16	1¼	1⅜	1⅜	1 7/16
11	2 3/16	2⅝	3 3/16	3 11/16	4 5/16
12	5 9/16	6½	7¾	8¾	9 13/16
13	3⅛	3⅝	4 1/16	4¾	5 7/32
14	1 13/16	2 1/16	2 11/32	....	....
15	1 15/16R	2⅜R	....	5⅝	6⅛
16	1 1/16	1 19/32	....	....	....

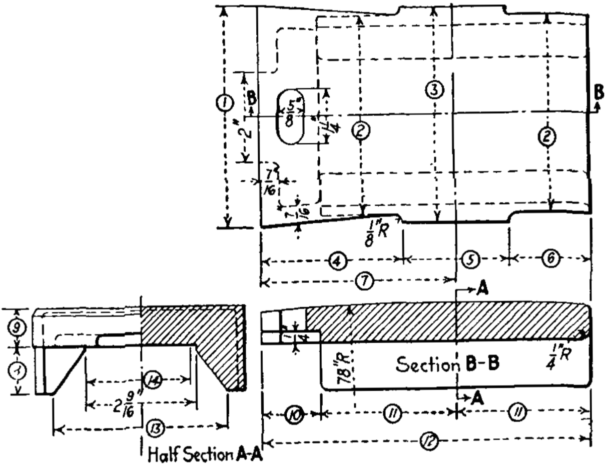


FIG. 32.—A.E.R.E.A. standard journal wedges.

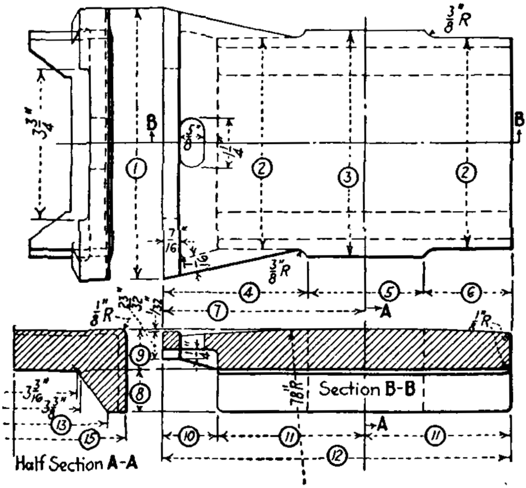


FIG. 33.—A.E.R.E.A. standard journal wedges.

A.E.R.E.A. Standard Journal Boxes, Bearings, Thrust Plates and Wedges. Figs. 28 to 33, inclusive, show the A.E.R.E.A. Standard Designs. They represent a development over a long period of time to overcome various difficulties. One of the chief difficulties was end wear on journals, which in the former types was greatest on the journal brass itself and next on the stop on the end of the axle. End wear was experienced chiefly where cars operated at relatively high schedule speeds and where there were a comparatively large number of short radius curves. In the present standards, end play has been overcome by casting guides integral with the journal box, so that a thrust plate can be dropped into the

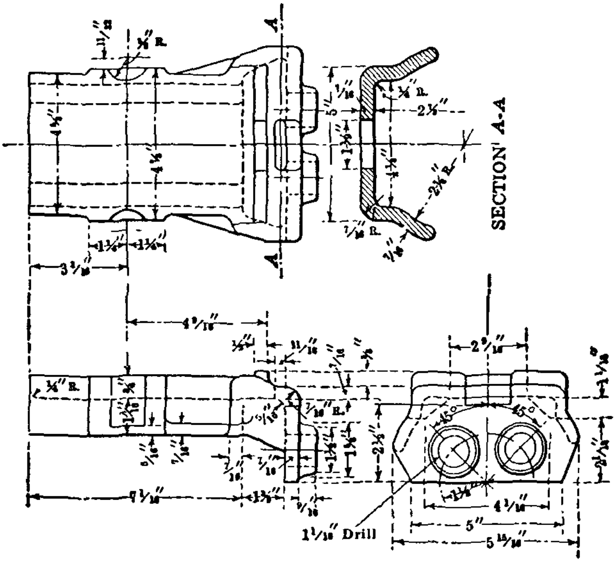


FIG. 34.—"Hooded" wedge.

guides and in close contact with the end of the axle, thereby imposing the lateral thrust of the axle upon the thrust plate rather than on the journal brass. The chief advantages of this method of overcoming end play are that the stress set up by the lateral thrust of the axle is more nearly in line with the axle itself, and that the thrust plate is easily renewed. It also serves to hold the packing in place in the lower portions of the journal and permits better inspection of the journal box. The journal bearing now has a more nearly full semi-circular cross-section, which gives ample bearing on each side of the journal to take the horizontal thurst from the brake shoes.

Hooded Wedge. Where the older types of journal box, bearing, etc., are still in use, the end play may be overcome by use of a hooded wedge, as shown in Fig. 34. The hooded wedge has a renewable wearing plate which is located so as to provide an end stop and wearing surface for the end thrust of the axle.

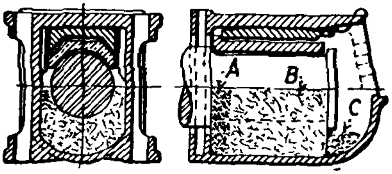
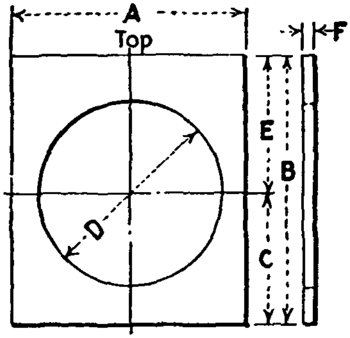


FIG. 35.—A.R.A. standard journal box, wedge and bearing, assembled.

A.R.A. Standard Journal Box, Bearing and Wedge. Figure 35 shows an assembly of the A.R.A. Standard Journal Box, Bearing and Wedge.

A.E.R.E.A. Standard Dust Guards. The A.E.R.E.A. standard dust guards, as shown in Fig. 36, fit the standard A.E.R.E.A. journal boxes. The guards are designed so that proper allowance is made for clearance and so that they cannot be reversed when being installed in the boxes.

Lubrication of Journal Bearings. (A.E.R.E.A. Miscellaneous Methods and Practices.) Once every 30 days, providing cars do



Size of Journal	A	B	C	D	E	F
3"x6" & 3¼"x6"	5¾"	7⅞"	4½"	3 15/16"	3⅜"	9/16"
3¾"x7"	6⅞"	8⅝"	4¾"	4 13/16"	3⅞"	⅝"
4¼"x8"	7⅞"	9¼"	5¼"	5 5/16"	4"	⅝"
5"x9"	8"	10⅜"	5⅛"	6 3/16"	4½"	⅝"
5½"x10"	8⅝"	10⅞"	6⅛"	6 11/16"	4¾"	⅝"

FIG. 36.—A.E.R.E.A. standard dust guards.

not make more than 150 miles per day, or if on a mileage basis, every 4500 miles. Journal bearings in high speed service should be lubricated weekly when cars are making from 300 to 500 miles per day, or if on a mileage basis, every 2500 miles. In both cases the journal box covers, springs and bolts should be examined to see that all parts are in good condition and that the cover is as nearly dust-proof as possible, and that the waste is stirred up in the boxes, and where it has worked forward, pressed back in place, and only a certain amount of oil should be applied. This can only be done by allowing a definite amount of oil for each car or by thoroughly educating certain men to do this part of the work.

A.R.A. Standard Method of Packing Journal Boxes. *Preparation of New Packing.* The waste should be loosened, placed in a saturating vat and kept completely submerged in car oil, at a temperature of not less than 70 deg. F., for a period of at least 48 hours to insure thorough saturation. It should then be drained for the purpose of removing the excess oil, until the packing is in a resilient or elastic condition. Prepared packing should be turned over at least once each 24 hours, or the oil which has accumulated in the bottom of the container should be drawn off and poured over the top of the prepared packing.

Preparation of Renovated Packing. All packing, when removed from journal boxes for the purpose of periodical repacking or renovating, should be piled into a container, avoiding contact with the ground or any other place where it may pick up dirt, and taken to the waste reclaiming plant. This packing must not be reused until renovated. In reclaiming packing it first should be picked over carefully, and dirt, metal, etc., shaken out, the knotted strands of waste pulled apart, and then placed in hot oil in renovating tank for a short time, working it with a fork for the purpose of thoroughly washing and loosening it. It then should be rinsed in clean oil, then drained for the purpose of removing excess oil.

Cleaning Boxes. Before packing a journal box the oil cellar should be thoroughly cleaned of all dirt, sand, scale and grit, and if water is present it must be removed. When new journal boxes are applied, or when reapplying journal boxes, the interior of the box, including the dust-guard well, should be so treated, and close-fitting dust guards and lids should be applied.

Cleaning and Applying Bearings. Before applying journal bearings, they should be thoroughly clean, have a smooth bearing surface, free from irregularities, and should have a proper bearing. Under no circumstances is it permissible to use sand paper, emery paper or emery cloth for the purpose of removing irregularities from the bearing surface; a half-round file or scraper should be used. Care must be taken that the wedge has a good contact on the crown of journal bearing. The surface of the journal should be smooth and thoroughly clean before the bearing is applied. When applying a journal bearing, a coat of lubricating oil should be applied to the bearing surface of same. Never wipe the bearing surface of the journal bearing with waste.

Application of Packing. (*a*) Inner. In packing a journal box, twist somewhat tightly a rope of packing and place it in the extreme back part of the box, as shown at *A* in Fig. 35. Make sure that it is well up against the journal so as to properly lubricate the fillet on the journal and keep out the dust. (*b*) Main. Apply sufficient packing (preferably in one piece) to fill the space shown at *B* in Fig. 35. Take care to have this packing bear evenly along full length of the lower half of the journal. The packing should not be too tight, but should be tight enough to overcome any tendency to settle away from the journal. The packing should extend to approximately the center line of the journal but not above at any point, and should be pressed down evenly at sides that no loose ends may work up under the journal bearings. (*c*) Outer. Apply a third piece of firmly twisted packing as shown at *C* in Fig 35, and pack tightly in order to prevent displacement of the main packing. There should be no loose ends hanging out of the box, as they would tend to draw out the oil.

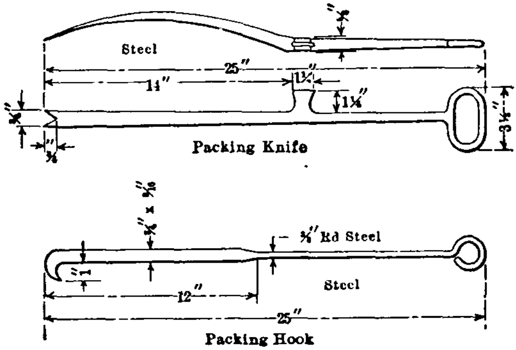


FIG. 37.—*Journal box packing tools.*

The committee which recommended the above practice also emphasized the importance of observing several other factors as follows, with the view of reducing hot boxes to the minimum: Journals after being turned should be cylindrical, free from taper, tool marks, ridges, corrugations and other defects, in other words the turned journal should reflect first class workmanship. The necessity of fully protecting journals against rust and corrosion during storage and guarding against damage to journals as a result of improper handling should be recognized. Care should be exercised in handling journal bearings to prevent tossing of journal bearings against each other, thus nicking and needlessly damaging the smooth bearing surface of the babbitt metal lining.

Figure 37 shows a representative set of journal box packing tools.

Journal Temperatures. The following comparing journal temperatures reached at different air temperatures are from tests by the use of the University of Illinois dynamometer car which is equipped with a recording thermometer, the bulb of which is inserted in a hole drilled in the body of one of the journal brasses. The car weighs 58,000 lb. and is equipped with 4¼ by 8 in. journals;

Average air temperature, deg. F.	63	9
Approximate average test speed, miles per hour	16	15
Maximum temperature attained by test car journal, deg. F.	116	98

Truck Overhauling. Complete overhauling means that the trucks must be entirely dismantled, the various parts repaired or excessively worn parts replaced, and the trucks again assembled. A thorough overhauling should place the trucks in the best of operating condition and as nearly as possible in their original condition. The basis for truck overhauling varies among different railway companies and conditions, some overhauling on the time basis in order to schedule their work in advance, while large systems employ the mileage basis, as it is really the mileage made by the equipment that determines the need for repairs.

The work of overhauling may be divided into the following: (1) measurements to determine necessary adjustments; (2) disconnecting brake rigging and motor leads; (3) lifting car body from trucks; (4) placing trucks in position for overhauling; (5) inspection to determine repairs needed; (6) dismantling equipment and parts; (7) repairing various parts; (8) reassembling; (9) running trucks under car body; (10) lowering car body; (11) connecting brake rigging and motor leads; (12) final measurements and adjustments. Some roads clean the trucks by some method or other before dismantling. The measurements taken for cars in city service may be step height, wheel guard height, side bearing height, or measurements at the corners of the car body to detect weak springs or defective parts. Cars in interurban or rapid transit service require measurements of platform and coupler height.

The following points were taken from the Westinghouse Railway Operating Data: (1) all parts of the truck and brake rigging should be carefully gone over and the badly worn hangers, pins, bolts, chafing plate, truck brake levers, and brake heads, should be repaired or replaced; (2) truck side and center bearings should be rigidly attached to the bolster; (3) height of bolster should be checked by a standard gage supplied for the purpose; (4) new brake shoes and turnbuckles should be supplied if these parts are badly worn; (5) adjust brake release spring; (6) wheels should be checked for diameter and flange wear, and either turned down or replaced by new ones, if necessary; (7) pairs of wheels should be kept to the same diameter to prevent crowding one rail or the other, with a consequent destruction of flanges and an increased train resistance; (8) diameters of pairs of wheels on the same car should not vary more than one-quarter inch, otherwise the motor on the larger diameter wheels will be overloaded; (9) gears should be checked, and renewed if they will not last until next overhauling period; (10) journal boxes, covers, bearings, check plates, etc., should be inspected, and removed if badly worn or broken; (11) axles should be checked in a lathe, and if bent they should be straightened or replaced by new ones; (12) if axles are badly worn, or have run their predetermined maximum mileage, replace by new ones; (13) axle collars should be tightened and checked for proper clearances.

SECTION VIII

BRAKING

Factors Controlling Length of Stop. The general factors which determine the distance traveled by a train while it is being brought to rest on a tangent level track are: (*a*) weight and speed of train at the beginning of the braking period; (*b*) coefficient of adhesion between wheel and rail (rail friction); (*c*) coefficient of brake shoe friction; (*d*) maximum brake cylinder force; (*e*) time required to attain this force; (*f*) efficiency of brake rigging in multiplying and transmitting this force to the brake shoe.

Available Adhesion between Rail and Wheel. This factor is subject to a wide variation on account of the varying condition of the rails and wheels with changing weather. For example, the adhesion between a dry rail and wheel may be twice that of a wet rail. The addition of sand to a slippery rail will increase the adhesion from 15 per cent to about 25 per cent of the weight on the rails. The maximum adhesion occurs when the wheels are rolling on the rail, and this adhesion rapidly decreases as soon as slipping occurs between wheel and rail. Thus the force of brake shoe friction which opposes the rotation of the wheels must never exceed the force of the adhesion between the wheels and rails which is keeping the wheels rotating. (See Coefficient of Adhesion, page 157.)

Brake Shoe Pressure. The maximum retardation which may be utilized in stopping a train by means of brakes is that which is realized by so applying brake shoes to all the wheels that the resulting brake shoe friction shall be uniform and just insufficient to overcome the static rail friction or adhesion. Thus the utilization of the entire retarding force available as rail friction or adhesion involves the application of a brake shoe pressure which shall (*a*) diminish as declining speed causes the coefficient of friction to increase; which shall (*b*) increase as increased distance of frictional contact causes the coefficient of friction to decline, and which shall (*c*), when diminishing or increasing for such purposes, further diminish or increase as reduction or increase of pressure itself causes the coefficient of friction correspondingly to increase or decline. It is impracticable to so regulate the brake shoe pressure that at any instant it will be equal to a maximum as the resultant of the above three, but since this resultant continues to diminish during the process of bringing the train to rest, the maximum possible retarding effect may be approximated by so diminishing the brake shoe pressure toward the completion of the stop that the wheels will not be caused to slide, nor the passengers and equipment be subjected to undue shock.

Emergency Braking. The above discussion shows that an emergency stop for high speed is less efficient than for a low speed, since the maximum pressure which will not slip the wheels near the end of the stop is applied at the beginning of the emergency application. A shorter stop would result if the pressure during the first part of the braking period was greater than that which would slip the wheels at low speed. This would require, however, some means to decrease the pressure near the end of the stop in order that the limits of rail friction are not exceeded and the efficiency of the stop thereby decreased. The additional apparatus is not desirable unless the conditions warrant the further complications. (See "Practical Application of Principles," page 436.)

Importance of High Rate of Retardation. In long runs, the braking rate is of little moment, but in short runs it becomes an important factor and, as in subway work, may be of maximum importance. In general, a high braking rate is advantageous because it allows more coasting in any run, which means an earlier point of cut-off, resulting in lower energy consumption and less heating of the motors than would accompany a low rate of braking. Fast braking thus tends to minimize the size of motor for a given service. The minimum length of a block and the consequent minimum headway is equal to the minimum distance in which a train can be brought to rest. Thus, increasing the rate of braking decreases the allowable headway and increases the capacity of the track.

High Speeds and Wheel Failures. In the use of high speeds care should be taken that the conditions of braking are such that they will not be dangerous to car wheels. The temperature attained at the wheel tread during braking depends upon the pressure at which the brake shoe is applied, the speed at which the wheel is turning, and the duration of the brake application. When the wheel is turning at high speed and the brake shoes are applied with great pressure, the metal at and near the surface of the tread suddenly becomes much hotter than that a short distance in. Great thermal stresses are thus set up in the wheel and these may cause fracture.

Coefficient of Friction at Various Speeds

CAST IRON BRAKE SHOES ON STEEL TIRES (GALTON-WESTINGHOUSE TESTS, PROCEEDINGS OF THE INSTITUTION OF MECHANICAL ENGINEERS, 1879)

Number of experiments from which the mean is taken	Speed		Coefficient of friction		
	Miles per hour	Feet per second	Extremes observed: Maximum	Extremes observed: Minimum	Mean
12	60	88	0.123	0.058	0.074
67	55	81	0.136	0.060	0.111
55	50	73	0.153	0.050	0.116
77	45	66	0.179	0.083	0.127
70	40	59	0.194	0.088	0.140
80	35	51	0.197	0.087	0.142
94	30	44	0.196	0.098	0.164
70	25	36½	0.205	0.108	0.166
69	20	29	0.240	0.133	0.192
78	15	22	0.280	0.131	0.223
54	10	14½	0.281	0.161	0.242
28	7½	11	0.325	0.123	0.244
20	Under 5	Under 7	0.340	0.156	0.273
	Just moving	Just moving			0.330
Fleeming Jenkin (steel on steel) 0.0002 to 0.0080			0.365	0.337	0.351

Rennie. Static friction under
pressure of 180 lb. per square inch........... 0.300
pressure of 336 lb. per square inch.......... 0.347

From the above values R. A. Parke has developed the following formulas to represent the law of variation of the coefficient of friction with speed:

From the mean values $$f = \frac{0.326}{1 + 0.03532S}$$

from the maximum values $$f = \frac{0.382}{1 + 0.02933S}$$

where f = coefficient of friction
S = speed in miles per hour.

The latter formula gives values corresponding more nearly to recent experiments.

The following table furnishes a comparison of the values of the coefficient of friction as calculated by the above formulas, with those secured by test (see above table):

Speed, miles per hour	Coefficient of friction			
	Mean		Maximum	
	Calculated	Observed	Calculated	Observed
0	0.326	0.330	0.382	....
5	0.277	0.273	0.333	0.340
10	0.241	0.242	0.295	0.281
15	0.213	0.223	0.265	0.280
20	0.191	0.192	0.241	0.240
25	0.173	0.166	0.220	0.205
30	0.158	0.164	0.203	0.196
35	0.146	0.142	0.188	0.197
40	0.135	0.140	0.176	0.194
45	0.126	0.127	0.165	0.179
50	0.118	0.116	0.155	0.153
55	0.111	0.111	0.146	0.136
60	0.105	0.074	0.138	0.123
65	0.099	....	0.131	....
70	0.094	....	0.125	....
80	0.085	....	0.114	....
90	0.078	....	0.106	....
100	0.072	....	0.097	....

Relation of Coefficient of Friction between Wheel and Rail to the Coefficient of Friction between Brake Shoe and Wheel and the Effect of Sliding. In the Galton-Westinghouse tests it was found that when the brake shoe friction overcame the rail friction and caused the wheels to slide upon the rails, the coefficient of rail friction immediately began to decline and then varied inversely as the speed, in much the same way as brake-shoe friction. It was found to have a value of less than one-third the coefficient of brake shoe friction when the brake shoe pressure was such as to allow the wheel to continue revolving. From this, the retarding effect of a wheel sliding upon a rail would be less than one-third the maximum available when the wheels are allowed to revolve without sliding.

Coefficient of Friction between Steel Tires and Steel Rail—Wheel Sliding. (Galton-Westinghouse Tests)

Speed, miles per hr.	Coefficient of friction
Just coming to rest	0.242
6.8	0.088
13.6	0.072
27.3	0.070
34.1	0.065
40.9	0.057
47.7	0.040
54.5	0.038
60.0	0.027*

* This is from the mean of three experiments only.

The rail friction is very materially affected by the condition of the rail, being greatest when the rail is perfectly dry or very wet (as when washed by a hard rain) and least when the rail is quite moist; but by the use of sand upon the rails, the effect of moisture is practically eliminated. (See Coefficient of Adhesion, page 157.)

Coefficient of Brake Shoe Friction as Affected by Distance of Application. Capt. Galton stated that a decrease in the coefficient of brake shoe friction results from the time during which the brakes have been kept applied, irrespective of any change in speed. He gave the following table and explained that the values given in the column headed "Commencement of experiment" are somewhat different from those that have been given in the table "Coefficient of Friction at Various Speeds . . .," page 433, because they resulted from the average of fewer experiments, but that the effect of time reducing the coefficient of friction may be accepted as correct.

COEFFICIENT OF BRAKE-SHOE FRICTION AS AFFECTED BY TIME (GALTON)

Speed, miles per hour	Coefficient of friction				
	Commencement of experiment	After 5 seconds	After 10 seconds	After 15 seconds	After 20 seconds
20	0.182	0.152	0.133	0.116	0.099
27	0.171	0.130	0.119	0.081	0.072
37	0.152	0.096	0.083	0.069	....
47	0.132	0.080	0.070	....	....
60	0.072	0.063	0.058	....	....

The following is from a paper by R. A. Parke, A.I.E.E., 1902: "It was assumed by Capt. Galton that the decline of the friction is as the increase of the time of contact, but a careful analysis of the results shows that it is a function of the product of the speed and the time, or of the distance, through which the shoe rubs upon the wheel." The following formula was developed by Mr Parke in an endeavor to show the relation of the coefficient of brake shoe friction at any point during the application of the brake shoe to the initial coefficient of brake shoe friction at the speed and pressure at which the brake shoe was first applied to the wheel and the distance traveled by the wheel during which the brake shoe has been applied to the wheel:

$$h = \frac{1 + 0.000472l}{1 + 0.002390l} f$$

in which h = coefficient of brake shoe friction at instant considered
l = distance wheel has traveled in frictional contact with the brake shoe, feet
f = coefficient of brake shoe friction at speed and pressure at which brake shoe was applied at the beginning of the distance l.

S. W. Dudley, in a paper before the A.S.M.E., 1914, on the Pennsylvania-Westinghouse Brake Tests, 1913, stated that factors such as speed, pressure and time of action, which are ordinarily considered to cause variations in cast iron brake shoe performance, are effective chiefly as they affect the temperature of the working metal of the brake shoe and wheel.

Practical Application of Above Principles. In emergency application of modern electric railway air brakes, a high cylinder pressure generally is held without reduction until the release is made. This at first appears contrary to the principles established by the Galton-Westinghouse brake trials to the effect that, as the speed of a train is diminished by continued brake action, the effectiveness of a given brake shoe pressure gradually increases, due to the coefficient of brake shoe friction increasing with decreasing speed. However, as pointed out by S. W. Dudley (*Electric Journal*, 1920) this principle remains as firmly established today as when first demonstrated. But it is found that with emergency braking of 125 to 150 per cent maximum in modern practice (as compared with the 200 to 300 per cent used in the Galton-Westinghouse trials) and the amount of work done per unit of brake shoe bearing area, the abrasion and heating of the brake shoe surface is such that the coefficient of friction remains substantially constant for the major portion of the stop. The increase in coefficient of friction and consequently in retarding force for a given brake shoe and brake cylinder pressure, which invariably occurs when the speed decreases below 25 miles per hour, although considerable, either does not become high enough to cause the wheels to slip at all or not until so near the stopping point that what slipping does occur is not detrimental. A significant fact in this connection is often not fully appreciated. It is better to take advantage of the maximum possible retarding force throughout the major portion of an emergency stop and risk slipping the wheels for the last ten or twenty feet, than to proportion the braking force so that the wheels will not slip at all, even on a bad rail, which may mean a collision when a safe stop could otherwise have been made. For these reasons, it has been found both desirable and feasible to hold the maximum braking force obtainable in emergency applications without reduction, until the train is stopped.

Distance Traveled by Train during Application of a Constant Retarding Force. If a constant retarding force is applied to a train

$$l = 0.733 \frac{S^2}{a}$$

or

$$= 66.8 \frac{WS^2}{k(F + F_1)}$$

in which l = distance traveled by train in coming to rest, feet
S = speed of train at beginning of period, miles per hour.
a = rate of retardation of train, miles per hour per second
k = ratio of linear inertia to total inertia of train (see page 151)
W = weight of train, tons

F = total retarding force applied to train, pounds
$= Ph$

or

$= 2000bW$

P = total braking pressure applied normally to wheel treads by brake shoes, pounds

h = coefficient of brake shoe friction

b = ratio of retarding force produced by brake shoes to total weight of train

F_1 = combined train, track grade and curve resistance
$= fW$

f = train resistance, pounds per ton weight of train.

Neglecting the energy of rotation of wheels and armatures, the formual is

$$l = 0.0333 \frac{S^2}{b}$$

$$= \frac{S^2}{30b}$$

Braking Distance, Single Car Tests. The following results were obtained from Bulletin No. 13, Engineering Experiment Station, Purdue University. This bulletin publishes the results of emergency stop tests on four different cars:

1. Interurban car, double truck, four motors.
2. City car, double truck, four motors.
3. City car, double truck, two motors.
4. Birney safety car.

As the result of a careful analysis of the test data, the following conclusions were reached: (1) There is no one best way to stop any car in an emergency. In general, the best way will depend somewhat on the type of brake and motor equipment. (2) As a broad rule, subject to occasional exceptions, it may be said that the best way to stop a car equipped with air brakes is to apply emergency air and sand, simultaneously throwing the controller to the off position. (3) The electrical methods of braking, reversed motors and bucking motors, subject the equipment to very severe strains. Because of the abruptness with which the braking force is applied, the value of these methods in emergency braking is more apparent than real, except in those cases in which the air brakes from some cause are inoperative. (4) Of the two electrical methods, reversed motors and bucking motors, the reversed motors method is the most effective. (5) If the motors are reversed in conjunction with emergency air and sand, skidding invariably results, and even under good rail conditions, the results are not superior to those obtained using air alone. (6) The electric braking methods are more likely to produce erratic results than the emergency air, and for this reason should not be resorted to as a general practice. (7) Because the rail conditions under which these tests were made were better than those ordinarily obtaining on city streets, the results should be regarded as minimums rather than averages or maximums. (8) The best way to stop a Birney safety car in an emergency is for the motorman to raise his hand from the controller handle. This is true only when the car is running with brakes released. With brakes applied the brake valve should be thrown into the emergency position

DESCRIPTION OF CARS USED IN PURDUE TESTS

Car No	61	1026	244	55
Type	Interurban	City	City	Birney
Weight loaded	82,139 lb.	42,758 lb.	38,507 lb.	20,446 lb.
Weight light load				16 945 lb.
No. trucks	2	2	2	1
No motors	4	2	4	2
Type brakes	Straight & Automatic	Straight	Straight	Safety
Wheel diameter	$35\frac{19}{32}$ in	$31\frac{1}{8}$ in	$29\frac{3}{4}$ in	$24\frac{5}{8}$ in.
Per cent braking power at 50 lb brake cylinder pressure (car empty)	102 1	58 0	83 5	167 0
Shoe area in contact, per cent	100	80	100	100
Sander	Air	Air	Hand	Air

BRAKING DISTANCES, PURDUE TESTS

(Distances in feet)

See table above for description of cars and equipment

Speed m p.h	Car No 61		Car No 244		Car No 1026		Car No. 55					
							Heavy load				Light Load	
	A	S & A	S HP	S LP	E HP	E LP	HL HP	HL LP	E HP	E LP	HL HP	HL LP
5			15					12				.
10			36	55	32	37	27	30	30	35	24	25
15		95	78	105	62	72	55	60	60	66	53	58
20	140	160	130	167	105	117	92	98	98	105	87	100
25	205	245	200	245	160	172	135	142	145	155	125	140
30	290	340	285		225	248						
35	380	440										
40	500	550										
45	650	660										

NOTE: A Automatic Air; S & A Straight and Automatic Air; S Straight Air.; E Emergency Air; H P High Pressure; L P Low Pressure; H L Hand Lifted from Safety Controller

Braking Distance Including Application Time. Neglecting the energy of rotation of wheels and armatures, neglecting train resistance, and considering the time required for the brakes to become effective, when the wheels do not slip

$$l = 1.467St + \frac{S^2}{30Peh}$$

in which l = length of stop, feet

S = speed of train at beginning of stop, miles per hour

t = time at beginning of the stop during which the brakes are to be considered as having no effect, seconds.

The value of t depends upon the equipment. In the Pennsylvania-Westinghouse 1913 tests on a twelve-car train it was found to range from 2 0 to 2 5 seconds for the quick action automatic (PM) brake and from 0 70 to 0 85 seconds for the electro-pneumatic (UC) brake

P = nominal per cent braking power (decimally expressed) corresponding to the average cylinder pressure existing for that portion of the stop after the brake is considered fully applied (The nominal per cent braking power is the ratio of the total shoe pressure, calculated from the full service brake cylinder pressure, to the weight of the car, expressed in per cent)

e = efficiency of brake rigging (decimally expressed)

h = coefficient of brake-shoe friction

In the Central Electric Railway Association-Purdue University tests of emergency braking on electric cars the following results were obtained

TIME LAG BETWEEN STOP INDICATION AND BEGINNING OF RETARDATION

(For description of cars and equipment, see page 438)	Time from the stop indication to the beginning of retardation, seconds			
Car No..	61	244	1026	55
Automatic air	1 08			1 08
Straight and automatic air	1 54			
Straight air	1 67	91	1 48	.
Reversed motors	44	1 00	1 54	88
Bucking motors	39	71	1 83	2 85
Reversed motors plus emergency air	1 02	43	1 39	
Hand lifted from safety controller				84

NOTE In the above tests the initial time indication was made automatically in the time circuit of a chronograph in the air brake tests, while in the electric braking tests the indication was made by the manipulation of a switch by the chronograph operator upon the report of a gun which was the signal at which the motorman applied the brakes.

Brake Shoe Pressure and Braking Power. *Nominal Brake Shoe Pressure* is that obtained by multiplying the brake cylinder (or hand brake lever) pressure by the lever ratio of the brake rigging On account of the friction in the brake rigging levers this nominal brake shoe pressure is never attained.

Actual Brake Shoe Pressure is the actual pressure exerted by the brake shoes against the wheels, and is equal to the nominal pressure divided by the efficiency of the brake rigging (the system of rods and levers between the cylinder and the brake shoes).

Braking Power, usually expressed as a per cent, is the ratio between the brake shoe pressure and the weight of the car on the wheels to which such pressure is applied. To obtain *nominal braking power*, nominal brake shoe pressure should be used, or for *actual braking power* actual brake shoe pressure as above described should be used. It should be noted that the term "braking power" is recognized to be a misnomer and should properly be replaced by "braking force," but as the former is the generally accepted term, it is thought best to conform to common usage herein.

The following values for braking power were given by E. H. Dewson in the *Electric Journal*, 1905, to show the approximate relation which the pressure applied to the brake shoes should bear to the total weight of the braked wheels to produce a brake friction equivalent to the adhesion of the wheels to the rails. Consideration is given to the fact that the coefficient of brake shoe friction is less at high speeds than at low.

Speed, miles per hour	Approximate ratio of total pressure on brake shoes to total weight on braked wheels, Coefficient of adhesion			
	0.30	0.25	0.20	0.15
7½	1.20	1.04	0.83	0.60
15	1.41	1.18	0.94	0.70
20	1.64	1.37	1.09	0.82
30	1.83	1.53	1.22	0.92
40	2.07	1.73	1.38	1.04
50	2.48	2.07	1.65	1.24
60	4.14	3.47	2.77	2.08

Brake Rigging Efficiency. A part of the brake cylinder force is lost in being transmitted to the brake shoe and as the leverage of the brake rigging is calculated from the piston force it is necessary to make proper allowance for this frictional loss. The efficiency of the rigging will depend upon the equipment, condition and pressure used. Tests to determine its value have been unsatisfactory. Such tests have yielded various values, but they indicate that in most cases the value for ordinary equipment in good condition probably lies between 80 and 85 per cent. Commenting on the Pennsylvania-Westinghouse 1913 tests, S. W. Dudley, A.S.M.E., 1914, states that they indicate that at least 85 per cent transmission efficiency could be obtained with either single shoe or clasp brake rigging. These tests indicate that the following features are important in securing the maximum overall brake rigging efficiency: (*a*) protection against accidents that may result from parts of rigging dropping on the track; (*b*) maximum efficiency of brake rigging at all times to insure the desired stopping with a minimum per cent of braking power; (*c*) uniform distribution of brake force, in relation to weight braked, on all wheels; (*d*) with a given nominal per cent braking power, the actual braking power to remain constant throughout the life of the brake-shoes and wheels; (*e*) piston travel to be as near constant as practicable under all conditions of cylinder pressure; (*f*) minimum expense of maintenance and running repairs of brake rigging between the stopping of cars.

The total retarding force acting on a car may be determined by multiplying the weight of the car in tons by the retardation in miles per hour per second and this product by 91.1. If from this force, the train resistance be subtracted, the remainder will be the retarding force produced by the brakes. The brake rigging efficiency then will be:

$$\text{Efficiency} = \frac{\text{Braking Force}}{\text{Braking Power} \times \text{Coefficient of Shoe Friction}}$$

Brake Efficacy. As it is difficult to determine the coefficient of shoe friction with accuracy, the product of this coefficient and the brake rigging efficiency are often used in comparing the efficacy of braking systems. S. W. Dudley, A.S.M.E. 1914, used the following values of *eh*, brake efficacy, as determined in the Pennsylvania-Westinghouse 1913 tests.

VALUES OF *eh* (BRAKE EFFICACY)

Kind of brake rigging		Clasp brake		Single shoe*	
Type of brake shoe		Plain	Flanged	Plain	Flanged
Speed, miles per hour	Nominal per cent braking power				
30	125	0.141	0.169	0.108	0.112
	150	0.129	0.154	0.099	0.103
	180	0.118	0.141	0.090	0.094
60	125	0.103	0.122	0.074	0.090
	150	0.094	0.112	0.068	0.082
	180	0.086	0.102	0.062	0.075
80	125	0.092	0.109	0.070	0.074
	150	0.084	0.100	0.064	0.068
	180	0.077	0.092	0.059	0.062

* Value of data uncertain, due to non-uniform brake shoe conditions.

In this test a single car (locomotive not attached) was stopped from an initial speed of 60 miles per hour in 725 ft. This was the shortest 60 miles per hour emergency stop made. The equipment was electropneumatic, clasp brake, flanged brake shoes, using 180 per cent braking power. Under the same conditions the shortest stop from 80 miles per hour was 1422 feet.

In the Central Electric Railway Association-Purdue University tests, the brake efficacies were calculated for a few cases and are given in the following table. (For description of cars and equipment, see page 438.)

Car	Speed m.p.h.	Brake efficacy	Car	Speed m.p.h.	Brake efficacy
61	8	0.155	244	3.6	0.138
61	36	0.082	244	16	0.195
1026	14.75	0.127	55	6.9	0.131
1026	25.5	0.192	55	22.5	0.104

Retardation Produced by a Given Retarding Force. The rate of retardation may be approximated by the following formula:

$$a = 0.01098 \frac{k(F + F_1)}{W}$$

in which

a = rate of retardation of train, miles per hour per second
k = ratio of linear inertia to total inertia of train (see page 151)
W = weight of train, tons
F = total retarding force applied to train, pounds
$= Ph$

or

$= 2000bW$
P = total braking pressure applied normally to wheel treads by brake shoes, pounds
h = coefficient of brake shoe friction
b = ratio of retarding force produced by brake shoes to total weight of train
F_1 = combined train, track grade and curve resistance
$= fW$
f = train resistance, pounds per ton weight of train

For a more rapid, very rough approximation in ordinary service, neglecting the energy due to the rotation of the wheels and armatures and the energy required to overcome train resistance, the formula is:

$$a = 0.011 \frac{F}{W}$$

or

$$= 22b$$

in which the symbols have the same significance as above.

Braking Distance Chart. The chart, Fig. 1, by Gaylord Thompson, A.E.R.E.A. 1914 Committee on Block Signals, gives a graphical approximation of the distance and time in which a train will be brought to rest when retarded at a constant rate. Example: Assume that such a braking force has been applied to a car as to produce a retardation of 3 miles per hour per second. It is desired to know how long a time will be required for the car to come to a stop and in what distance it will do so from a speed of, say, 50 miles per hour. The solid line for 3 miles per hour per second shows that a speed of 50 miles per hour corresponds to a braking distance of 600 ft. This is indicated by the point A. Directly below A, at the point where a vertical line dropped from A intersects the distance-time curve for 3 miles per hour per second and reading on the right-hand scale, we find that the time required for the stop is 16.5 seconds. That is, with this rate of retardation it will take

16.5 seconds to stop a car running at 50 miles per hour, and in stopping it, a distance of 600 ft. will be covered. A similar procedure shows that from 60 miles per hour the same braking rate will stop the car in 20 seconds and 880 ft.

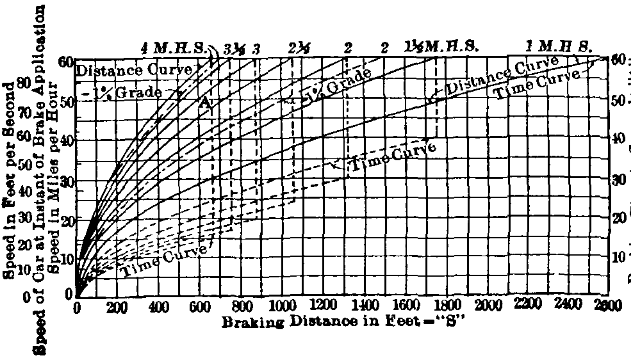


FIG. 1.—Distance and time for braking.

Rates of Retardation. *The effective rates of retardation* obtained in the Central Electric Railway-Purdue University tests are as tabu'ated below, in miles per hour per second (for description of cars and equipment, see page 438).

Speed m.p.h.	Car No. 61		Car No. 244		Car No. 1026		Car No. 55					
							Heavy load				Light load	
	A	S & A	S HP	S LP	E HP	E LP	HL HP	HL LP	E HP	E LP	HL HP	HL LP
5	....	....	1.22	..	....	....	..	1.53	.	...	....	.....
10	....	...	2.08	1.33	2.29	1.98	2.72	2.44	2.44	2.10	3.06	2 82
15	....	1.74	2.12	1.57	2.66	2.29	3 00	2.75	2 75	2.50	3.11	2 84
20	2.09	1.83	2.25	1.76	2.79	2 51	3.19	3.00	3 00	2 79	3.37	2 93
25	2.25	1 89	2.31	1.89	2 89	2 68	3 42	3.25	3 19	2.98	3 70	3 30
30	2.28	1 94	2.32		2 93	2 66	.	.		.		..
35	2 36	2.08	...	...	....	....	....	....	....	....	...	.....
40	2 35	2.13	....	....	....	....	....	....	....	....	...	.....
45	2 31	2 28	.	..	....	. .	....	....	....	....	...	.....

Note.—A Automatic Air. S & A Straight and Automatic Air.
S Straight Air. E Emergency Air.
H P High Pressure. L P Low Pressure.
H L Hand Lifted from Safety Controller.

The "effective" rates of retardation as shown above were calculated by using the distance from stop signal indication to full stop and the speed at the stop signal indication. Thus these values assume a uniform rate of retardation and include the lag due to personal element of motorman and due to the time it takes for the pressure to build up in the brake cylinder. In the tests, the actual braking rate varied throughout the braking period, in general becoming the maximum at the end of the braking period. For example, the following maximum values were noted in the report on the above cited tests:

Car 61, at 10.5 m.p.h., 4.9 m.p.h.p.s.
Car 244, at 7.5 m.p.h., 4.97 m.p.h.p.s.
Car 1026, at 8.0 m.p.h., 5.1 m.p.h.p.s.
Car 55, at 8.2 m.p.h., 7.28 m.p.h.p.s.

Weight Transfer Due to Braking. The following is based on the analysis by R. A. Parke, A.I.E.E., 1902:

Retardation caused by the application of brakes to the wheels involves a redistribution of the weight which, where the retardation corresponds to the maximum brake application, results in a very serious loss of braking efficiency, unless means be provided for varying the brake shoe pressure to correspond with the changed wheel pressures against the rails. Whatever the source of the retarding force, its operative effect in retarding the motion of the car is the same as that of an infinite number of small retarding forces each engaged in retarding the motion of an elementary portion of the mass of the car, and therefore, in order that a single retarding force, or a combination of retarding forces, shall so operate, without either changing the direction of the car's motion, or producing rotation of the structure as a whole, or calling into operation other forces to prevent such deviation or rotation, the force or the resultant of the combination of forces must be so applied that it shall pass through the center of gravity of the mass, in a direction opposite to that of the motion of the car. In utilizing the rail friction for the retarding force, while it has the proper direction, it is applied at the lowest points of the mass of the car and must therefore either cause rotation of the entire structure or the interposition of other forces which combine with the retarding force to preserve the simple motion of translation. The car may be considered as a single mass or as being composed of three separate masses, according as it is provided with one rigid or two swiveling trucks. Rotation of the car by the eccentric retarding force does not occur in either case; but, in the first case, this is because the reacting pressure of the rails upon the forward pair of wheels exceeds that upon the rear pair of wheels in such measure that the contrary rotative moment, thereby introduced, just balances that of the eccentric retarding force, also applied by the rails to the wheels. In the case of the car with two trucks, the retarding force is applied at the lower extremity of the two trucks, being equally divided between them (if constructed alike); that portion of the retarding force necessary to retard the mass of the trucks is absorbed in so doing, and the remainder is applied by the trucks to the car body at substantially its lowest extremity. In consequence, the center of gravity of the car body being above the points of application of the retarding force, rotation through the eccentrically applied retarding force is prevented only by the resisting rotative moment of a greater supporting pressure from the forward than from the rear truck.

Each truck is subject to the combined rotative moment of the eccentric retarding force at its lower extremity, and the eccentric reacting force from the car body at its upper extremity, and rotation is prevented only by the contrary rotative moment of a greater supporting pressure by the rails upon the forward than upon the rear pair of wheels. Thus the very act of applying the brakes to the wheels produces a new and very different system of wheel pressures upon the rails, and it is the wheel pressures under these conditions which determine the available retarding force. As the total pressure of all the wheels upon the rails cannot vary, the existence of a greater rail pressure for the forward than for the rear pair of wheels of the truck implies the virtual transfer of a portion of the normal pressure from one pair of wheels to the other. The brake shoe pressure upon the rear pair of wheels must be insufficient to cause the wheels to slide upon the rails and must therefore be cut down in proportion to the transfer of weight from the rear to the forward pair of wheels. But as the forward pair of wheels will become the rear pair when the car moves in the opposite direction, the brake shoe pressure upon that pair of wheels must also be limited in the same way. Thus, the braking pressure upon each pair of wheels must be restricted to correspond with the minimum pressure at the wheels upon the rails, which occurs when the wheels are the rear pair.

External Forces Acting on Car Body in Braking. (Fig. 2.) The following equations were obtained by taking the summation of the horizontal forces, summation of the vertical forces, and moments of forces about the point of application of P_1

$$H = \frac{W_1 a}{2g}$$

$$P_2 = \frac{W_1}{2} - \frac{W_1 aj}{gl}$$

$$P_1 = \frac{W_1}{2} + \frac{W_1 aj}{gl}$$

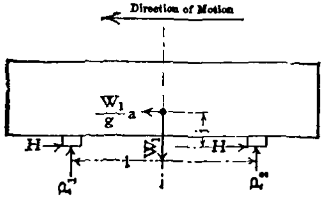


FIG. 2.—Weight transfer in braking. Car body.

in which H = horizontal retarding force on each truck center pin, pounds
(H is the same for each truck because the brake pressure is to be limited for each pair of wheels to *correspond with the condition of minimum pressure* of wheels on rail, and hence each truck during braking will exert an equal retarding force for equal brake pressure on all wheels of car)

P_2 = pressure between body and truck rear center plates, pounds

P_1 = pressure between body and truck forward center plates, pounds

W_1 = weight of car body, pounds. (Center of gravity being in a vertical axis midway between truck centers)

j = height of center of gravity of body above center plate surface, inches
l = distance between center pins, inches
g = acceleration due to gravity, feet per second per second = 32.2
a = rate of retardation, feet per second per second = (miles per hour per second) $\times$ 1.467
k = ratio of linear inertia to total inertia of cars. (See p. 151.)

External Forces Acting on Rear Truck in Braking. (Fig. 3.)

$$R_2 = \frac{W_2}{2} + \frac{P_2}{2} - H\frac{h}{b} - \frac{W_2ad}{gb}$$

$$R_1 = \frac{W_2}{2} + \frac{P_2}{2} + H\frac{h}{b} + \frac{W_2ad}{gb}$$

$$T_2 = qR_2$$

$$T_1 = qR_1$$

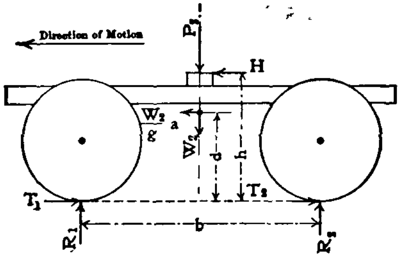


FIG. 3.—Weight transfer in braking. Truck.

in which R_2 = total pressure between rail and rear pair of wheels, pounds
R_1 = total pressure between rail and front pair of wheels of rear truck, pounds
T_2 = total maximum retarding rail friction available between rail and rear pair of wheels, pounds
T_1 = total maximum retarding rail friction available between rail and front pair of wheels of rear truck, pounds
W_2 = weight of each truck, pounds. (Center of gravity of truck being in a vertical axis midway between wheels)
W = total weight of car, pounds = $W_1 + 2W_2$
h = height of truck center plate surface above rail
b = wheel base of truck
d = height of center of gravity of truck above rail
q = coefficient of adhesion between wheel and rail

For significance of other symbols see above.

The above formulas do not take into consideration the weight transfer due to the inertia of rotating parts (wheels and motor armatures). This effect may vary, depending upon the relative weights and arrangements of such rotating parts; the effect is small, however, and in all practical cases, the formulas as shown will give results within the limits of error in the assumptions necessary with respect to locations of centers of gravity and coefficient of adhesion.

Brake Shoe Suspension. The application of the brake shoes at the outer face of the wheels results in an upward thrust of the brake hangers, proportional to the brake shoe friction, upon the rear end of the truck frame, and a corresponding downward drag upon the forward end. However, by hanging the brake beams between the wheels, instead of outside, and inclining the hanger links at a proper angle, the increased pressure and consequently the increased friction of the brake shoes upon the forward pair of wheels and the diminished pressure and friction of the brake shoes upon the rear wheels, due to the effect of the friction itself in causing the shoes to press more or less forcibly upon the wheels through the angularity of the hanger links, are made to correspond with and compensate for the transferred weight from the rear to the forward wheels. In the same manner that running in the opposite direction causes a reversal of the conditions for the transfer of weight, so, too, the rotation of the wheels in the opposite direction causes a reversal of the effect of the inclined hanger links, and the increased brake shoe pressure is always applied to the wheels carrying the increased weight. The application of this method of inclined hanger links is not without some difficulty. The chief trouble is that no constant angle of the links can be maintained, as the wearing away of the brake shoes, together with wearing and turning down or grinding of the wheel treads, causes constant and considerable variation. Thus, if the angle of inclination and the braking pressure be calculated for the conditions existing when the brake shoes and wheels are new, the increased angle when the shoes become much worn and the treads have been well turned off, would probably cause the forward wheels to slide upon the rails. On the other hand, if the calculations be made for turned or ground wheels and worn shoes, the rear wheels would probably slide when the wheels and brake shoes are new. It is therefore necessary to compromise between the extremes, in reference to the angle of inclination of the hanger links, by inclining the hanger links at the angle determined when the brake shoes and wheels are each half worn, and the brake beam force must then be so established that neither pair of wheels shall be caused to slide in the extreme positions of the hanger links.

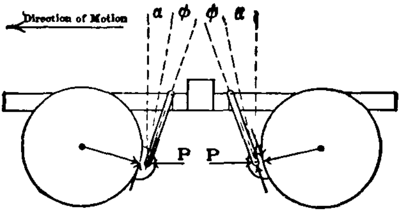


FIG. 4.—Brake-hanger suspension angles.

Angle of Suspension of Brake Hanger. (Fig. 4.) The following formula gives the angle between the brake hanger link and the tangent to the wheel at the center of pressure of brake shoe face necessary to secure the maximum retarding force with inside hung brake shoes and with motors driving all axles:

$$\tan\ \phi = 2\,\frac{qk(W_1h + W_2d)}{rWb}$$

in which ϕ = angle between brake-shoe hanger-link and the tangent to the wheel at center of brake-shoe surface

r = coefficient of brake-shoe friction (0.33 may be used for this).

For significance of other symbols, see pages 445 and 446.

Effect of Rotational Inertia. In addition to destroying the energy of translation existing in the moving car, the brake shoe friction must also absorb the rotational energy of the wheels, axles and motors. This inertia is entirely independent of that due to translation, and in destroying it the coefficient of adhesion between wheels and track does not enter. Practically, a greater braking force must be used to produce retardation when the wheels and other rotating parts are taken into consideration. This is taken care of in the above formula by the value k.

Brake-beam Pressure (Motors Driving All Axles). (Fig. 4.)

$$P = \frac{qW^2(1 - r^2 \tan^2 \phi) \cos \phi}{4rk\left(W + 2qW_1\dfrac{j}{l}\right) \cos\,(\alpha + \phi)}$$

in which P = horizontal braking force applied to brake beam in the direction of motion of the train, or, the sum of the horizontal braking forces applied to one pair of wheels in the direction of motion of the train, pounds

α = angle between radius of wheel to center of brake-shoe surface and the horizontal.

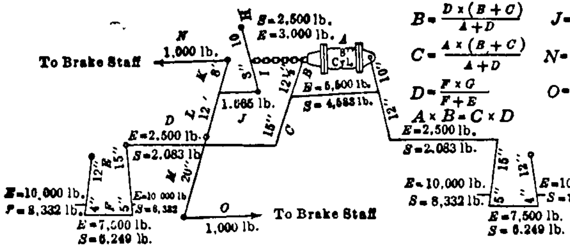
For significance of other symbols see preceding paragraph, also pages 445 and 446.

Wear on wheel and brake shoe will bring about a change in the values of the angles α and ϕ and the dimension j, consequently the value of the force P will vary during the life of wheel and brake shoe. The value of the force P should be determined for new wheels and brake shoes, and again for worn wheels and brake shoes. The lesser value thus obtained will be the maximum value of P that may be used safely.

Brake Rigging Calculation. Fig. 5 gives formulas for calculating truck pull rod force, brake shoe pressure and the dimensions of the lever carrying the brake shoe for the three general methods of suspension. Figs. 6 and 7 give formulas which show the relations between the brake shoe pressures on adjacent wheels and the pull rod force. By an adjustment of the lever dimensions these brake shoe pressures may be made to bear a desired relation to each other.

The following example of brake rigging calculation (see Fig. 9) is by E. H. Dewson, *Electric Journal*, 1905:

A car weighing 46,700 lb. without load has a weight of 32,200 lb. on the motor truck and 14,500 lb. on the trailer truck. The motor



$$B = \frac{D \times (B + C)}{A + D} \qquad J = \frac{A \times H}{H + I}$$

$$C = \frac{A \times (B + C)}{A + D} \qquad N = \frac{J \times L}{K + L}$$

$$D = \frac{F \times G}{F + E} \qquad O = \frac{J \times L}{M}$$

$$A \times B = C \times D$$

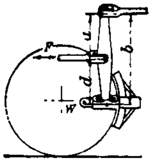
The following table gives the forces exerted upon the pistons of the different size cylinders with pressures of 50 lb., 60 lb. and 70 per sq. in.

Size of Cylinder	16″	14″	12″	10″	8″	7″
Area of Cylinders				78,54 sq. in.	50,27 sq. in.	38,48 sq. in.
50 lb. Pressure	10,050	7,700	5,650	4,000	2,500	1,850
60 lb. Pressure	12,050	9,200	6,700	4,700	3,000	2,300
70 lb. Pressure	14,100	11,800	7,900	5,500	3,500	2,700

Light Weight of Motor Car, 40,000 lb.
Braking Power, 40,000 lb. or 100 Per-cent
E indicates Forces in Emergency Applications and
S indicates Forces in Full Service Applications.

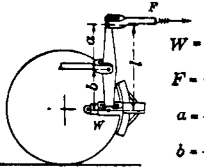
AUXILIARY RESERVOIRS USED WITH DIFFERENT SIZES OF BRAKE CYLINDERS.

10″ x 33″ Auxiliary Reservoirs with 8″ Brake Cylinder of all kinds
12″ x 33″ " " " 10″ " " " " "
14″ x 33″ " " " 12″ " " " " "
16″ x 33″ " " " 14″ " " " " "
16″ x 42″ " " " 16″ " " " " "



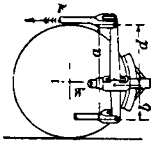
$$W = \frac{F \times a}{b}$$
$$F = \frac{W \times b}{a}$$
$$a = \frac{W \times b}{F} \text{ or } a = \frac{W \times d}{F - W}$$
$$b = \frac{F \times a}{W} \text{ or } b = \frac{F \times d}{F - W}$$

Applied Force between Fulcrum and Delivered Force.



$$W = \frac{F \times a}{b}$$
$$F = \frac{W \times b}{a}$$
$$a = \frac{W \times b}{F} \text{ or } a = \frac{W \times l}{F + W}$$
$$b = \frac{F \times a}{W} \text{ or } b = \frac{F \times l}{F + W}$$

Fulcrum between Applied and Delivered Forces.

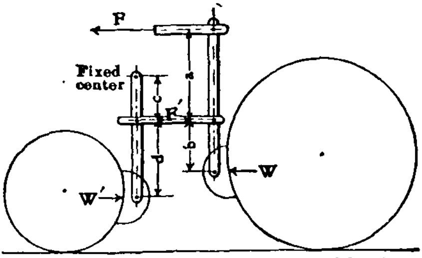


$$W = \frac{F \times a}{b}$$
$$F = \frac{W \times b}{a}$$
$$a = \frac{W \times b}{F} \text{ or } a = \frac{W \times d}{W - F}$$
$$b = \frac{F \times a}{W} \text{ or } b = \frac{F \times d}{W - F}$$

Delivered Forces between Fulcrum and Applied Force,

FIG. 5.—Formulas for calculating brake rigging (Hartford Shop Instruction Print).

truck is equipped with inside hung brake shoes connected directly to a double set of levers, no brake beams being used. The trailer truck has outside hung shoes mounted on brake beams, consequently a single set of levers is used. One hundred per cent brake power on the motor wheels requires a force of 8050 lb. at each brakeshoe. Ninety per cent brake power on the trailer truck requires a force of 6525 lb. at each brake beam. The total brake power will be 45,250 lb., and as a 10-in. piston under a pressure of 60 lb. per square inch exerts a force of 4700 lb., the total leverage ratio will be about 9.6 to 1, which is satisfactory. The levers supplied with the motor truck are 27 in. long and the shoes are hung $7\frac{1}{2}$ in. from the lower end. Considering the dead lever first with its fulcrum at the upper end we know the delivered force W and its distance b from the fulcrum, also the distance a from the fulcrum to the applied force F; substituting these values in the equation $F = \frac{W \times b}{a}$ we have $F = \frac{8050 \times 19.5}{27} = 5814$ lb. as the stress in the adjusting rod. Considering the lower end of live lever as the fulcrum we have $W = 8050$ lb., $b = 7.5$ in. and $a = 27$ in., consequently the pull at the upper end $F = \frac{8050 \times 7.5}{27} = 2236$. The proof is that $5814 + 2236 = 8050$.



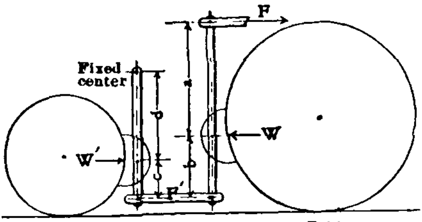
$$W = \frac{F \times a}{b} \qquad F = \frac{W \times b}{a}$$

$$a = \frac{w \times b}{F} \qquad b = \frac{F \times a}{w}$$

$$W' = \frac{F\,(a+b)\,c}{(c+d)\,b} \qquad W' = \frac{F' \times c}{(c+d)}$$

$$F' = \frac{F\,(a+b)}{b} \qquad W = W' \text{ when } \frac{a}{b} = \frac{c}{d}$$

FIG. 6.—Braking pressures on adjacent pairs of wheels. I.



$$W = \frac{F\,(a+b)}{b} \qquad F' = \frac{F \times a}{b}$$

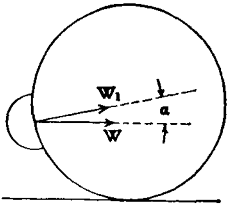
$$F = \frac{W \times b}{(a+b)} \qquad W' = \frac{F'\,(c+d)}{d}$$

$$a = \frac{(W-F)b}{F} \qquad W' = \frac{F\,(c+d)\,a}{b \times d}$$

$$b = \frac{F \times a}{W-F} \qquad W = W' \text{ when } \frac{a}{b} = \frac{d}{c}$$

FIG. 7.—Braking forces on adjacent pairs of wheels. II.

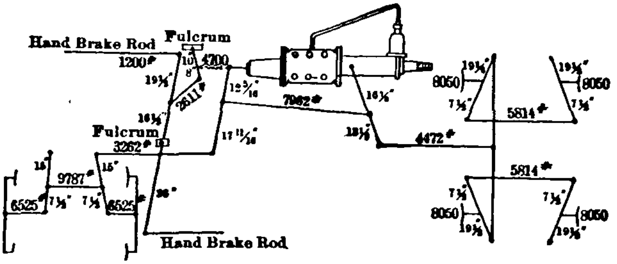
As a pull of 2236 lb. must be exerted on each side of the trucks, the stress in the truck pull rod will be 4472 lb. For the dead lever of the trailer truck we have $W = 6525$ lb., $b = 22.5$ in. and $a = 15$ in., consequently the stress in the adjusting rod is $F = \frac{6525 \times 22.5}{15} = 9787.5$ lb. With the intermediate point of the live lever taken for the fulcrum, $W = 6525$ lb., $b = 7.5$ in. and $a = 15$ in. and the stress in the pull rod $F = \frac{6525 \times 7.5}{15} = 3262.5$ lb. The proof is that $6525 + 3262.5 = 9787.5$.



$W_1 = w \cos \alpha$
in which W_1 is force normal to the wheel treads at center of brake-shoe surface.

Fig. 8.—Relation between horizontal and normal braking forces.

It is now necessary to so proportion the push rod lever that with a piston force of 4700 lb it will deliver 3262 lb. (neglecting the half) to the trailer truck pull rod and the cylinder lever that it will deliver 4472 lb. to the motor truck pull rod. For efficient operation of the slack adjuster these levers should not be less than 30 in. long; with short levers the action of the adjuster would cause excessive angular distortion. If the applied force at the cylinder end of the push rod lever is 4700 lb., and the delivered force at the pull rod is 3262 lb., that at the push rod will be 4700 +



Car weighs 46,700 lb.

32,200 lb. on motor truck	100 % brake power on motor truck
14,500 lb. on trailer truck	90 % brake power on trailer truck

Fig. 9.—Calculations for a specific brake equipment.

3262 = 7962 lb. Assuming that the pull rod pin is the fulcrum we have $F = 4700$, $a = 30$ in., $W = 7962$ and $b = \frac{F \times a}{W} = \frac{4700 \times 30}{7962} = 17^{11}\!/\!_{16}$ in. For the cylinder lever we have $F = 7962$ lb., $W = 4472$ lb., and $b = 30$ in., therefore $a = \frac{4472 \times 30}{7962} = 16\tfrac{7}{8}$ in.

With the ordinary brake handle, staff and chain supplied with electric cars, 1200 lb. is about as much force as the average man can exert on the hand brake pull rod. This must be multiplied to 4700 to give the same brake power by hand as by air pressure. Using a multiplying lever 18 in. long and connecting to the push rod pin by a chain fastened 10 in. from the fulcrum, the force required at the end of the lever will be $\frac{10 \times 4700}{18} = 2611$ lb. Assuming that the hand brake lever is 6 ft. long and pivoted at the center, we have $F = 1200$ lb, $a = 36$ in. $W = 2611$ lb and $b = \frac{1200 \times 36}{2611} = 16.54$ in., or practically $16\frac{1}{2}$ in.

Safety Stops for Brake Mechanism. The double truck cars on the Holyoke Street Railway are equipped with a special safety stop

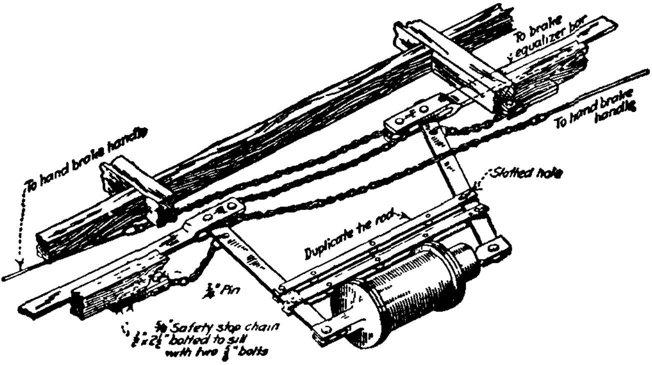


FIG. 10 —Safety stops for brake rigging. Holyoke.

for the brake rigging, originated by the engineers of that company. The mechanism, as shown in Fig. 10, prevents the brake rigging on one end of a car from becoming noperative through the breakage of a pin in the mechan sm on the opposite end. There are two provisions to hold the lever operated by the air cylinder in case of breakage of connecting pins. One of these is to protect against the breakage of the pin connecting this lever with the rod operating the brake equalizer bar on each truck. This protection consists of a short chain one end of which is attached to the lever and the other is bolted to a car sill. The other safety device is to protect against the breakage of the pins which connect the tie rod with the two levers operated by the air cylinder. This protection, as shown, consists of a duplicate tie rod close to the operating one and with a slotted hole at one end in which the brake lever pin can move, so that the duplicate tie rod does not interfere with the movements of the levers.

Variable Load Compensating Device. With light weight on the wheels a low brake cylinder pressure is essential in order that slipping of the wheels may not occur with no load. If the same brake cylinder pressure is used with a loaded car, stopping distances will be excessively long, with increased danger on severe grades. The use of the variable load compensating device limits the pressure passing from the operator's brake valve or from the emergency valve to the brake cylinder to a maximum amount which will not cause wheel sliding, whatever the live load on the car may be. It is thus possible to obtain a given braking ratio with the car empty or fully loaded or with some intermediate weight. When less than a full service application is desired, the motorman can make a partial application with his brake valve and then graduate the cylinder pressure to that desired. The variable load apparatus, however, prevents getting into the cylinder any more pressure than

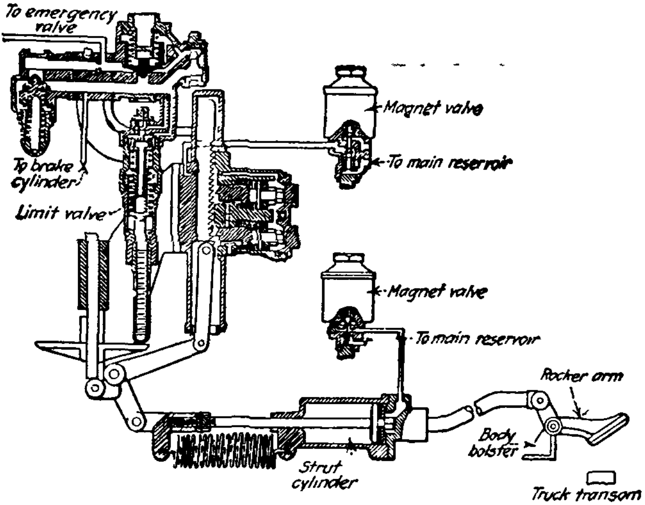


FIG. 11.—Variable load compensating device for air brake equipment

the load of the car will justify. Referring to Fig. 11, as the car comes to a stop, the opening of the doors energizes two magnet valves through a door contactor. The upper one of these magnet valves exhausts air so as to unlock the pressure limiting valve mechanism, while the lower magnet valve admits air to the strut cylinder. A rocker arm is mounted on the body bolster and the admitting of air to the strut cylinder extends the push rod of this cylinder so as to bring the foot plate of this rocker arm in contact with the truck transom. At the same time the other end of the push rod is moved out corresponding to the load on the car and adjusts the limit valve to its proper position. This limiting valve is in effect an adjustable feed valve and regulates the maximum pressure to the brake cylinder. With a certain light weight car, 85 per cent braking effort is obtained with approximately 34-lb. brake cylinder pressure, and the same car with a load of 21,000 lb., which corresponds to 150 passengers, requires 57-lb. brake cylinder pressure to give 85 per cent braking effort. The closing of the doors de-energizes the magnet, locks the limiting valve and exhausts air from the strut cylinder. The strut cylinder spring then pulls the cylinder so as to lift the foot plate from the truck transom, and thus the vibration of the car when it is moving is not transmitted to the variable load apparatus. All air to the brake cylinder passes through the limiting valve. As this is adjusted at each stop to the load on the car, the maximum pressure going to the brake cylinder is proportioned to the load on the car. In other words, it is possible to secure a cylinder pressure that will give a proper braking ratio with a loaded car and yet hold the cylinder pressure for an empty car down to a value that will not cause wheel sliding.

A full appreciation of the great value of this improvement is obtained when one considers the effect of a 40 per cent increase in the total weight, by means of the live load, upon the stopping distance of multiple unit trains as heretofore braked. The weight of the empty train determines the maximum braking power that can then be applied to it, and under the old conditions an increase of 40 per cent in weight meant lengthening all the stopping distances obtainable with the empty train by about 40 per cent when the same train was loaded. With the variable load compensating attachment, all trains can be stopped in the same minimum distances, irrespective of loading. As noted below, the acceleration also may be maintained at a uniform maximum for any given grade condition, irrespective of loading. These important factors, together with the shorter spacing of block signals involved, permit a remarkable increase in the train capacity on a rapid transit railway.

As used on the New York Municipal subway cars, the variable load compensating device is tied in with the selective acceleration feature of the multiple unit control, by means of an extra winding on the limit switch. The modification of current input to the motors is controlled from a switch operated in connection with the variable load compensating mechanism.

Importance of Proper Relation between Air Pressure, Piston Area and Leverage. The following is from a paper by Fred Heckler, C.E.R.A., 1907:

Pressure. If more than 2 per cent braking power per pound of cylinder pressure is attempted, a very high braking power for light cylinder pressures is obtained and, therefore, the cars cannot be handled without shocks at low speeds and either the range between maximum and minimum braking power obtainable must be very narrow or else wheel sliding will result when the maximum power is used.

Brake Piston Area. If the ratio of cylinder piston area to cylinder pressure is excessive, it means either a low leverage, with great shoe movement or high leverage with low pressure, which gives a very narrow range between maximum and minimum braking power.

Leverage. If the leverage is too low, it means excessive air consumption and too much shoe movement; if too high (that is, brake cylinder too small for weight of car, and it is here that the principles governing brake design are violated most frequently), smooth and accurate handling of the car or train becomes impossible and the shoes are constantly grinding on the wheels, consuming energy, wearing out the shoe and causing loss of time; or else piston travel must be lengthened out, thus greatly increasing the air consumption, lengthening the time of application and release, and reducing both service and emergency braking power. Besides, the high leverage makes necessary a frequent adjustment of piston travel or a constant and very rapid decrease of braking power will result. Furthermore, high leverage, if made at truck levers, necessitates low hung brake shoes, which, when suspended from a spring supported part of the truck, results in great increase of piston travel and resultant decrease of braking power with the loading of the car; this always occurs at a time when readjustment of piston travel is impracticable and when instead of a decrease of braking power an increase is greatly to be desired. In addition, the danger of the levers fouling is greatly increased, particularly where the truck leverage ratio is high, and very frequent and careful inspection is required or the total loss of the brake may result.

DIAMETER OF BRAKE CYLINDER, TOTAL LEVERAGE RATIO AND WEIGHT OF CAR

Diameter of cylinder in inches	Force of piston at 50 lb.	Total leverage ratio	Weight of car with brake power equal to		
			90 per cent	100 per cent	110 per cent
8	2,500	12	33,300	30,000	27,300
10	3,950	11	49,300	43,450	39,500
12	5,650	10¾	67,500	60,760	55,250
14	7,700	10	85,500	77,000	70,000
16	10,050	10	111,660	100,500	91,360

Position of Brake Shoe on Wheel. If the brake shoe is hung too far below the center of the wheel and the parts are badly worn, the application of the brakes may cause the brake shoe to form a toggle joint with the wheel, stopping the rotation of the wheel. This makes braking inefficient and brings about unnecessary brake shoe and wheel wear. In the above-mentioned paper by R. A. Parke it is stated that the center of the brake shoe should be about 3½ in. below the center of the wheel. For the standard location adopted by the Central Electric Railway Association see page 457 ("Center on brake shoe").

Brake Cylinder, Lever and Rigging Standards of the Central Electric Railway Association. The following recommendations of the Standardization Committee were adopted by the Association, 1911:

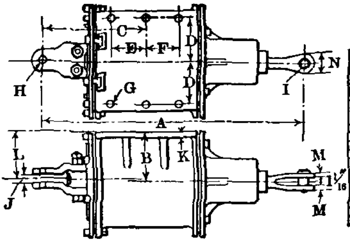
(*A*) *Revision of Standard Air Brake Cylinders, Levers and Brake Rigging.* Recommended: That the standard air brake practice should be according to revised print No. C-2 (see Fig. 12) as follows:

1. All braking power to be based on 50 lb. cylinder pressure. This in order to avoid confusion when stating percentages of brak-

ing power that may be figured on different brake cylinder pressures; *e.g.*, 100 per cent braking power on 60 lb. cylinder pressure may be taken to mean greater than 100 per cent on 50 lb. cylinder pressure, and if this is always referred to on a common base, confusion will be avoided.

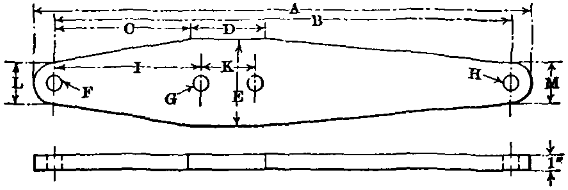
2. All interurban cars to be braked at 100 per cent of light weight on rails and motor axles, and 90 per cent on non-motor and trailer axles

Size of Cyl.	M	N
8"x12"	3/4"	2 1/2"
10"x12"	3/4"	2 1/2"
12"x12"	3/4"	2 3/4"
14"x12"	3/4"	3"
16"x12"	3/4"	3"



Size of Cyl.	L		
	Plain Head	Emer. Valve	Triple Valve
8"x12"	4 9/16"	4 1/16"	4 1/16"
10"x12"	5 11/16"	5 3/16"	5 3/16"
12"x12"	6 1/4"	5 3/4"	6"
14"x12"	7 15/16"	-	7 15/16"
16"x12"	9"	-	9"

Size of Cyl.	A			B	C			D	E	F	G	H	I	J	K
	Plain Head	Emer. Valve	Triple Valve		Plain Head	Emer. Valve	Triple Valve								
8"x12"	34 3/8"	37 5/16"	37 11/16"	4 9/16"	14 3/8"	16 15/16"	17 5/16"	4 1/16"	4 1/2"	4 3/8"	11/16"	1 1/8"	1 3/8"	1 3/8"	3/8"
10"x12"	34 3/8"	37 7/16"	36 3/4"	5 11/16"	13 7/8"	16 11/16"	15 5/8"	4 11/16"	4 1/4"	4 1/4"	11/16"	1 1/8"	1 3/8"	1 3/8"	11/16"
12"x12"	35 3/8"	37 5/8"	38 1/8"	6 3/8"	14"	16 3/8"	17"	6 3/8"	4 5/8"	4 7/8"	15/16"	1 1/8"	1 3/8"	1 3/8"	13/16"
14"x12"	35 3/8"	-	38 1/8"	7 15/16"	11 1/4"	-	17 1/4"	6 5/8"	4 5/8"	4 5/8"	15/16"	1 1/4"	1 1/4"	1 1/4"	1"
16"x12"	35 5/8"	-	38 7/16"	9"	14 3/8"	-	17 3/16"	8 5/8"	4 5/8"	4 5/8"	1 1/16"	1 1/4"	1 3/8"	1 1/4"	5/8"



CYLINDER LEVERS

Size of Cyl.	A	B	C	D	E	F	G	H	I	K	L	M
8"x 12"	29"	26"	6 3/4"	8 1/4"	3 1/2"	1 3/8"	1 3/8"	1 1/8"	7 3/4"	5 1/4"	3"	3"
10"x 12"	31"	28"	7 1/4"	8 1/4"	4 1/2"	1 3/8"	1 3/8"	1 3/8"	8 3/4"	5 1/4"	3"	3"
12"x 12"	34 1/8"	31 1/8"	8 9/16"	8 11/16"	6"	1 3/8"	1 3/8"	1 3/8"	10 1/16"	5 11/16"	3"	3"
14"x 12"	39 3/16"	35"	9 3/8"	9 1/2"	7"	1 1/4"	1 3/8"	1 3/8"	11"	6 1/2"	3 1/2"	3"
16"x 12"	39 3/16"	36"	10"	9 3/8"	8 1/2"	1 1/4"	1 3/8"	1 3/8"	11 3/8"	6 1/2"	3 1/2"	3"

ALL PINS TO BE 1/32" LESS IN DIAMETER THAN DRILLED HOLES.

FIG. 12.—Central Elec. Ry. Assn. standard brake cylinders and levers.

3. All city cars to be braked at 85 per cent on motor axles and 75 per cent on non-motor and trailer axles. Seventy-five per cent on 50 lb. is practically the same as 90 per cent on 60 lb.

4. Brake pipe pressure to be 70 lb. per square inch with automatic equipments.

5. Governor adjustment—85 and 100 lb. for automatic equipment. Governor adjustment—50 and 65 lb. for straight air equipment.

6. The standing piston travel adjustment—4 in.

7. Total truck leverage to be 6 to 1 for long wheel base trucks for inside hung motors, and 9 to 1 for short wheel base trucks with outside hung motors.

8. *A 12 to 1 maximum total leverage is permissible when brake* shoes are hung not more than 2 in. below the center of the wheel. If brake shoes are hung lower than this it will be necessary to reduce the maximum total leverage accordingly, and if brake shoes should be hung 5 or 6 in. below the center of the wheel a total leverage of 10 to 1 should be the limit.

9. The standard M.C.B. recommendations for maximum stress in levers, rods and pins to be adopted as follows:

(*a*) Maximum stress in levers—23,000 lb. per square inch. (*b*) Maximum stress in rods, except jaws—15,000 lb. per square inch, no rod to be less than ⅝ in. (*c*) Maximum stress in jaws to be 10,000 lb. per square inch. (*d*) Maximum shear on pins—10,000 lb. per square inch, single shear. (*e*) Diameter of pins to provide a bearing value not to exceed 23,000 lb. per square inch, projected area.

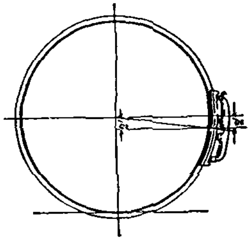


FIG. 13.—Standard brake shoe position, Cent. Elec. Ry. Assn.

10. Safety valve adjustments—10 lb. above maximum governor setting.

(*B*) *Center on Brake Shoe.* Recommended: That center of brake-shoe be set 2 in. below center of wheel. (See Fig. 13.)

Piston Travel and Shoe Clearance. "Total leverage" represents the total number of pounds of nominal brake shoe pressure that is obtained for each pound of force exerted on the brake cylinder push rod. To obtain any desired degree of braking force or percentage of braking power with a specified air pressure in the brake cylinder, the greater the value of total leverage employed, the smaller is the diameter of the brake cylinder required, but with a corresponding increase in piston travel. A very high total leverage is objectionable because a slight shoe wear produces a great variation in piston travel. The increase in piston travel due to the wear of brake shoes is equal to the average amount of wear of the shoes multiplied by the total leverage employed. Proper operation of the automatic brake depends largely on the relative volumes of the auxiliary reservoir and the brake cylinder and thus it is important to use a reasonably low total leverage and so maintain a more nearly constant piston travel with ordinary shoe wear. In steam railroad service, where cars receive attention at only long intervals and where the piston travel is not adjusted regularly, a lower total leverage is necessary than in street railway service, where systematic inspection and maintenance generally can be relied upon. Moreover, in street railway service the amount of air consumed would be excessive if the piston travel were long and the leverage low, as brake applications usually are very frequent in such service. It is customary to adhere as closely as possible to the piston travel standards which have been fixed by the best practice, and which must be carefully maintained in service in order to secure satisfactory results. Standard piston travel should always be the average of the maximum and minimum values actually obtained. A further reason for adhering to established standards of piston travel and braking power is that it is most essential to provide for clearance between brake shoes and wheels when the brake is released. There is always a certain amount of lost motion in a foundation brake rigging, and the use of either a very high total leverage or an excessively short piston travel might result in dragging the brake shoes. If there were no lost motion between the push rod and the shoes, the shoe clearance would be exactly equal to the piston travel divided by the total leverage. However, owing to the clearances in pin holes and to the spring and "give" of the various parts of the rigging, the shoe clearance actually is considerably less than this value. The piston travel obtained when a car is standing is usually an inch or two less than that obtained when a car is running, because vibration and motion of the various parts provide for their better adjustment. Consequently allowance should always be made for this inequality, as it is the correct "running" piston travel that must be maintained. If the brakes are applied while the car is in motion and held on until and after the stop is made, the piston travel thus measured will be the "running" value.

A very important consideration in the installation and adjustment of brake apparatus is the motion or travel of the various parts concerned. The location and dimensions of all moving parts must be so chosen that each lever and rod will always be unobstructed in its movements, even when the brake shoes are worn down completely and the brake cylinder piston has traveled to its maximum limit, or otherwise the brake easily might be rendered totally ineffective. An additional requirement of good practice is that all levers, except the truck levers, should stand at right angles to their connecting rods when the brake is fully applied and the piston travel has the standard running value. This applies particularly to the two cylinder levers The connecting rods will thus maintain their required approximately parallel directions much more nearly than would otherwise be the case, for obviously when a lever is moved through half of a specified angle on each side of a perpendicular position with respect to the connecting rods, the resulting deflection of these rods will be much less than when the lever is moved through the same angle but not equally on both sides of the perpendicular. In order to determine the travel or the position of any moving part of a brake rigging, it is merely necessary to remember that when a lever (or standard bellcrank) turns about one of its three points as a center, the two other points travel in circular paths, and the length of the path passed over by each of these points is in direct proportion to its distance from the center about which the lever turns. In cases where a lever has no positively fixed point, as for example a cylinder lever of live truck lever, the travel of a specified point can be determined by considering as being fixed, in turn, each of the two points the travel of which is known.

Clasp Brake. The following is based upon comments by S. W. Dudley, A.S.M.E, 1914, after the Pennsylvania-Westinghouse 1913 tests: The use of the clasp type of brake rigging eliminates unbalanced braking forces on the wheels and so avoids the undesirable and troublesome journal and truck reactions that come from the use of heavy braking pressures on but one side of the wheel. This not only has an important effect on freedom from journal troubles, but it also enables the wheel to follow freely vertical inequalities of the track. Heavy braking pressures on but one side of the wheel causes uneven wear on the journal bearings. Although the clasp brake rigging will produce better stops than a single shoe brake rigging equally well designed (other conditions being equal), its advantage in this direction is of less importance than in the improved truck, journal and shoe conditions. The use of two shoes per wheel permits a design of rigging which will allow flanged shoes to be used without danger of pinching flanges and causing excessive flange wear or non-uniform brake forces which result when flanged shoes are used with rigid beam connections. The use of two shoes instead of one per wheel will result in a higher coefficient of friction and less wear per unit of work done. A comparison of the values of mean coefficient of friction for standard and for clasp brake conditions indicates a decided advantage for the clasp brake throughout the entire range of braking powers. The gain in favor of the clasp brake with slotted shoes amounts to about 40 per cent at a braking power of 180 per cent, and 100 per cent at a braking power of 40 per cent, an average gain for the whole range of braking powers of about 70 per cent

From a brake shoe standpoint the advantage of using two shoes instead of one shoe per wheel may be summed up as follows: First, the clasp brake is associated with but one-half the wheel load and consequently has but one-half as much energy to absorb, second, the clasp brake shoe is working at only one-half the shoe pressure at which the standard shoe must work under the same braking power; third, the available work area for the same amount of energy to be absorbed is double.

A possible source of disadvantage when using two shoes per wheel is that a warped or poorly bearing shoe is subjected to less pressure tending to force it into a good contact with the wheel. For this reason, though the available shoe area is doubled when using clasp brakes, the actual amount of working metal throughout the stop may be less than with a single shoe, which is less capable of resisting the tendency of the heavier pressure to cause a better fit of shoe to wheel.

With plain solid shoes the durability will be increased 40 per cent under clasp brake conditions as compared with that under single shoe conditions. With plain slotted shoes the durability will be increased 33 per cent under clasp brake conditions as compared with that under single shoe conditions.

Automatic Slack Adjuster. The function of the automatic slack adjuster is to maintain the brake shoe travel automatically at a practical minimum while keeping the piston travel at a minimum and nearly uniform. By doing this, a maximum efficiency of brakes is approached, the necessity of inspection for the adjustment of brakes is eliminated, the brakes on all cars operate as nearly alike

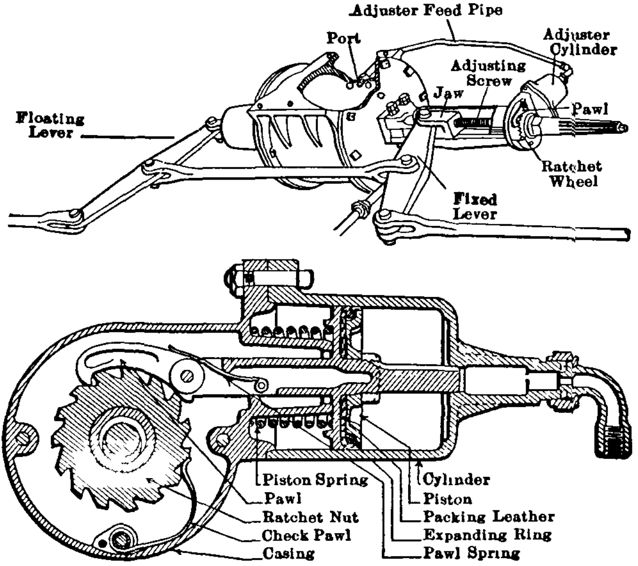


FIG. 14.—American automatic slack adjuster.

as may be and their operation is nearly the same throughout the life of the brake shoes, the energy consumption and brake shoe wear due to shoes dragging are eliminated and a minimum of air and the accompanying minimum amount of energy and compressor duty are demanded. Automatic slack adjusters are usually located either at the brake cylinder or on the truck. Figs. 14 and 15 show typical adjusters located at the brake cylinder and Fig. 16 shows a typical adjuster located on the truck where it takes the place of the ordinary turnbuckle brake rod connecting the bottoms of the live and dead levers.

American Automatic Slack Adjuster. (Fig. 14.) The adjuster cylinder is connected by the adjuster feed pipe to a port in the brake cylinder wall. When the brake piston uncovers this port, compressed air flows to the adjuster cylinder, thus operating its piston which in turn actuates a pawl engaging the ratchet wheel and nut on the screw. Release of brakes allows the spring in the

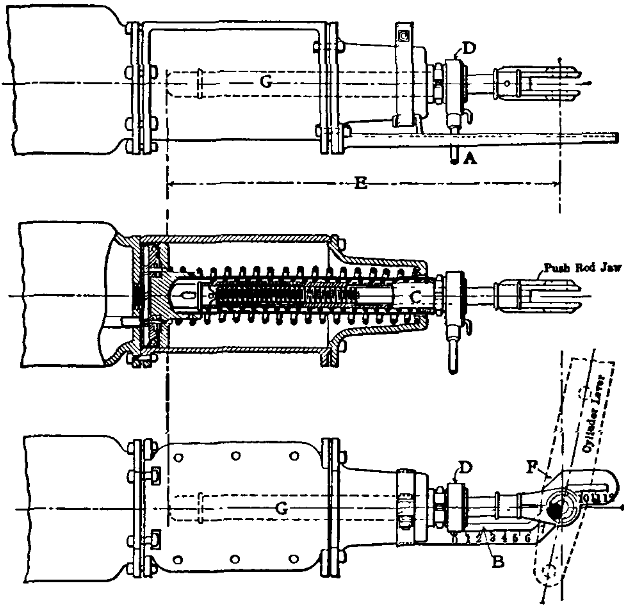


FIG. 15.—Creco automatic slack adjuster.

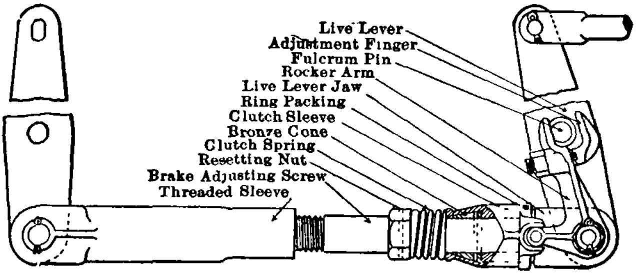


FIG. 16.—Anderson automatic slack adjuster.

adjuster cylinder to return the piston in the latter to its normal position, resulting in the rotation of the ratchet nut on the adjusting screw. The inner end of the screw carries a jaw which acts as a fulcrum for the fixed lever of the brake cylinder.

Creco Automatic Slack Adjuster. (Fig. 15.) Pin *A* traveling in slot *B* of bracket *F* passes beyond the straight line which is made the length of the desired piston travel and goes into that part of the braket slot which is at an angle of 45 deg. from the straight line or the line of normal piston travel. The action of the pin in traveling through the angled portion of the slot is to carry the ratchet pawl over the teeth of the ratchet *D*, and upon the return of the pin back to the angle with the release of the brakes to turn the barrel *C*, which is rigidly attached to the ratchet, upon the screw portion of the push rod proper, thereby lengthening the distance *E* from the lever pin hole in the jaw to the end of cylinder *G*.

Hand Brakes vs. Air Brakes. Hand brakes are satisfactory for ordinary operation on light cars and should be supplied for emergency on all cars. On one man cars pneumatic devices are often required in the interests of safety; in such cases air brakes are used even though the cars are light in weight. However, as the weight of a car increases the difficulty of control increases till it reaches such a value that power is necessary for ordinary braking. Just what weight should divide the hand brake class from the power brake class has been a matter of much investigation, many tests and disagreements. The difficulties of such tests are briefly summed up by J. N. Dodd of the Public Service Commission, First District, New York, as follows:

"In the course of this search it became apparent that tests do not always furnish a reliable criterion of the relative value of different brakes. They merely state the stopping distance of the car under the particular conditions that obtained at the time of the test. Most of the conditions that affect this distance vary widely. Among these varying conditions may be mentioned the brake-shoe adjustment, the weight of the car, the condition of the rail, and the human element. Most of these factors vary widely also during the course of the day, and each of them may change independently of any of the others. Thus, on account of the wear of the brake shoes, the brake adjustment may change materially even in the course of a single trip. The weight of the car is continually changing, owing to the varying number of passengers. The condition of the rail may alter entirely in the course of a few seconds. This change is often such that a visual inspection of the rail fails to reveal its quality. Thus, a wet rail may provide an ideal surface for stopping a car quickly, or it may offer the reverse. In the same way, though not to the same extent, the distance in which a car may be stopped on a dry rail varies according to whether the rail is clean or covered with dust or dirt. For this reason it is impossible to be sure that the rail conditions are the same in tests on two different brakes or that the rail conditions at the time of the test correctly represent average rail conditions under which the car must operate throughout the year. The human element also is extremely variable. In actual service there are many strong motormen and many who are physically weak, many who are intelligent and mentally alert and many the reverse, many in fresh physical and mental condition and others may be tired from a day's work. During most tests the motorman usually knows that he is soon to receive the stopping signal and, knowing what he is expected to do, is intent upon doing that thing in the most efficient manner. During such tests, also, the streets are usually bare of traffic and the motorman's attention is not distracted by other duties such as making up lost time, keeping a lookout to pick up passengers and obeying the conductor's signals. Usually a picked motorman is chosen, selected for his general intelligence and interest in his work. The motorman is generally in fresh physical condition and therefore in good mental condition and to that extent capable of responding to any demands made upon him. The results obtained in service at the close of a day's work might be entirely different from the results obtained in any series of tests. It is impossible to devise any series of tests under conditions which even approximate the average conditions existing in actual service because it is impossible to tell what the average of these varying factors may be."

After much consideration the Public Service Commission, First District, New York, required cars weighing more than 25,100 lb. to be equipped with air brakes.

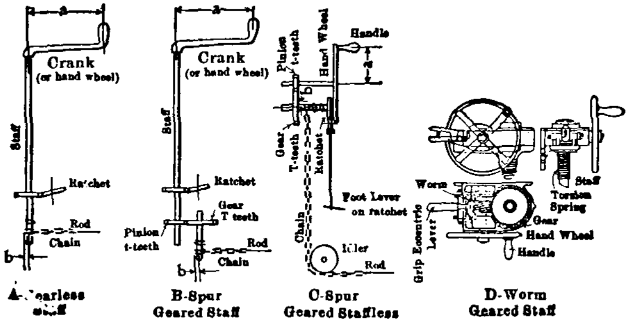


FIG. 17.—Typical hand brake schemes.

Typical Hand Brake Schemes. The more common hand brakes may be divided into four types, namely: gearless staff, spur geared staff, spur geared staffless and worm geared staff. The scheme of each of these is shown by Fig. 17. Although the drum on which the chain is wound may be of uniform diameter throughout its length, many of the drums now used are conical or eccentric so that the slack chain may be rapidly taken up over the portion of large radius and the final tension applied to the chain at small radius, thereby giving a maximum tension when it is needed. As the crank or handwheel is turned the tension is transmitted through this chain, thence through a rod, thence through a chain to the brake lever system (see Figs. 5, 9 and 10). In calculating the relation of tension in the hand brake rod to the pull applied to the crank, the distance b, Fig. 17, is the effective radius of the drum when the brakes are fully applied. This distance varies according to the shape of the chain, its method of winding and the travel of the brake shoes. Tests made by G. L. Fowler on the cars of the Brooklyn Heights Company, using dynamometers in the truck pull rods, showed the braking pressure at the wheels to vary as much as 40 per cent with a given pressure applied to the brake handle on different applications on the same car. This was found to be due entirely to the manner in which the chain rolled on the brake staff.

Gearless Staff Hand Brake (A, Fig. 17). The gearless staff is the most common type of hand brake used on light cars.

$$(tension\ in\ rod) = \frac{a}{b} \times (pull\ on\ handle).$$

Spur Geared Staff Hand Brake (B, Fig. 17). An increase in mechanical advantage over that of the gearless staff type is secured by the spur gear.

$$(\text{tension in rod}) = \frac{a}{b} \times \frac{T}{t} \times (\text{pull on handle}).$$

Spur Geared Staffless (C, Fig. 17). The force applied at the handwheel is transmitted through a spur gear to the winding drum. The ratchet is normally held at release by gravity and is set by a foot lever. Most satisfactory results have been secured with gear ratios of 14 : 23 and 12 : 36.

$$(\text{tension in rod}) = \frac{a}{b} \times \frac{T}{t} \times (\text{pull on handle}).$$

Worm Geared Staff (D, Fig. 17). A worm on the handwheel shaft drives a gear on the staff. Ordinarily a compression spring holds the worm in mesh with the gear. Release is obtained by means of a grip eccentric lever which disengages the worm from the gear. The slack chain is taken up by a torsion spring.

Hand Brake Maintenance. Brake staff defects are due principally to the staff binding and not releasing freely, and are often caused by the drawbar rest being displaced. Brake chain troubles are largely due to the hand brake binding and jamming between the brake staff and sill. One of the most important points in the transmission of braking power from the brake handle to the wheels is the winding of the brake chain on the staff. A close link chain should be used and care taken to have sufficient lead to the chain to allow it to roll on the staff without one turn binding or running upon another. There also should be sufficient release spring pressure to pull slack chain promptly from the staff, so that the chain will wind on the staff directly below the eye bolt. Great care should also be taken to see that the lead of the chain is such as to prevent its winding above the eye bolt and jamming against the platform and rendering the brakes inoperative. Another point to be guarded against is that of the chain at the rear end of the car catching on the snow scrapers and thus preventing an application of brakes. Inspection of the brake chain should guard against badly worn links or eye bolts, or the possibility of nuts working off the eye bolts.

Types of Air Brakes. Straight air brake system, recommended for single car operation only.

Emergency straight air system, suitable for two car operation, particularly when one is operated single most of the time and with a trailer added during rush hours.

Automatic air brake system, suitable for trains of three or more cars.

Combined straight and automatic air brake system, for locomotive service and for operation of single cars or trains of several cars.

Electropneumatic air brake system, for service similar to that of the automatic system.

Straight Air Brake System. (Fig. 18.) The straight air brake system consists essentially of a source of compressed air (either a tank filled at intervals from an air compressor, motor or axle driven, located upon the car, or rarely from a compressor at charging stations); a reservoir which receives the air from the compressor or charging tanks and in which the pressure is maintained practically constant by means of a governor which automatically controls the operation of the compressor; a brake cylinder, the piston of which is connected to a system of brake levers in such a manner that when the piston is forced outward by air pressure the brakes are applied; an operating valve mounted in each vestibule by means of which the compressed air is either admitted to or released from the brake cylinders; a pipe system connecting the above parts, including cut-out valves, hose, and angle fittings between cars. In order to prevent any possibility of accumulating an excessive pressure, a safety valve designed to open at 100 lb. per square inch is placed in the air supply system. A set of pressure gages is usually supplied with each complete equipment in order that the motorman may observe the pressure in the reservoir and remedy any defects in the governing apparatus.

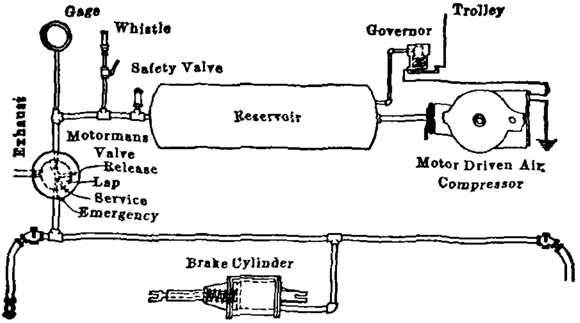


FIG. 18.—Straight air brake system.

To operate the motorman's valve the handle is inserted when the valve is in lap position where the slot in the body of the valve is enlarged for this purpose (and to prevent its removal in any other position). In this position the valve is set so that air can neither pass into nor out of the brake cylinder. Moving the handle to the left places the valve in full release, that is, connects the brake cylinder to the atmosphere and allows the air which holds the brakes applied to escape, when the spring which is opposed to the air pressure restores the piston and releases the brakes. To partially release the brakes, which is necessary in braking in order to prevent shocks as the car stops, the handle is moved to the left and returned to lap position. This reduces the pressure on the brake shoes, but does not entirely release them. To apply the brakes for a service stop the handle is moved to the service application position which is to the right of the lap position. This connects the reservoir with the brake cylinder through a small port in the valve. The handle should be left in this position until a sufficient amount of pressure is built up in the brake cylinder to give the retarding effect desired, when the handle should be moved back to the lap position. As the speed of the car is reduced, brake cylinder pressure may be reduced in a series of steps by moving the handle from the lap to the release position and then back to lap, repeating this movement until the stop is reached. At the point of stopping, there should be only sufficient air in the brake cylinder to prevent the car from rolling. Better braking results will be obtained by making one application as described than by admitting only a small amount of air to the brake cylinder at the beginning of the stop and increasing the pressure as the speed of the car is decreased. The latter method usually results in rough stops and a waste of air. The practice of applying and releasing the brakes several times during the stop should be avoided. Moving the handle further to the right connects the reservoir to the brake cylinder through a large opening, thus causing the cylinder to fill rapidly and quickly apply the brakes with maximum pressure. Sand usually is applied to the tracks as soon as the handle is turned to emergency to avoid skidding the wheels. In descending grades a light application of the brakes may be made and the handle returned to lap. A sufficient length of time should be allowed for car to feel the effect of the brakes before applying more pressure. If speed is higher than desired a second light application should be made and operation repeated as often as necessary until the desired speed is obtained, or until the car has left the grade.

The straight air system of air brakes, although only recommended for single car operation, may be used when operating with a trailer. The equipment for trail cars consists of a brake cylinder and system of levers similar to the ones on the motor car, a length of pipe running the entire length of the car and provided with hose couplings and cut-out cocks for connections to the forward and rear cars. In connecting up trail cars, all the hose couplings must be thoroughly united to insure that air will apply throughout the entire train. All the cut-out cocks must be opened except those on the rear of the last car, and the front of the first car, which must be closed.

So far as single car operation is concerned, the straight air brake system is very satisfactory, as the desired flexibility in the matter of graduations of applications and release of the brakes with due regard to the passengers standing can readily be secured, and this apparatus is usually so simple in construction that the motorman may become familiar with its operation to such an extent that accurate stops may be secured with a minimum amount of instruction. Installation and maintenance costs are low, on account of the small number of parts. In trains of considerable length, however, the response of the brakes on the rear cars is too slow, since all the air must pass from the main reservoir on the front car through the opening in the motorman's valve to the brake cylinders of each car. As the addition of each car adds to the volume of the brake system, the main reservoir on the first car must be considerably increased in order that the pressure will not be reduced to such an extent that the brake application will not be sufficient and result in overrunning the desired stopping place. These latter objections would not be sufficient to prevent the use of this type of air brakes on short trains of two or three cars were it not for the fact that a broken hose connection or leaky train pipe renders the brakes on the whole train inoperative.

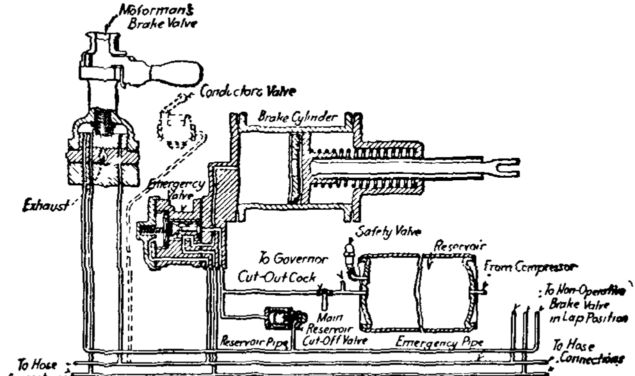


FIG. 19.—Emergency straight air brake system.

Emergency Straight Air Brake System. (Fig. 19.) The emergency straight air brake differs from the straight air brake in the details of the motorman's valve, the addition of an emergency valve, and the use of three pipe lines—emergency, reservoir and the straight air application and release pipes in place of the single pipe line. In the ordinary operation of single cars or short trains, the emergency valve is seldom brought into play. It is necessary, however, to provide a short direct passage from the reservoir to the brake cylinder in order to insure the quickest possible action in time of emergency and to provide some means of automatically braking the rear cars should a break occur in the train line. At other times when it is desired to make a service application or release, the air is admitted or exhausted through the motorman's valve the same as in the straight air brake. In Fig. 19 the motorman's valve is in the service position and air from the main reservoir passing through the main reservoir cut off valve (which is always open except in case the reservoir pipe should break, when it closes and protects against loss of the brake in emergency) to the reservoir pipe is admitted to the straight air application pipe at the motorman's valve and reaches the brake cylinder through the emergency valve. The emergency valve normally connects the straight air application pipe to the brake cylinder, and service applications and release are made by increasing or decreasing the pressure in the train pipe, as with the straight air system.

The emergency position of the motorman's valve should be used only when it is necessary to stop the car within the shortest possible distance to save life or avoid accident. In this position the straight air application and release pipe connection is blanked in the brake valve, while the air in the emergency pipe is exhausted to the atmosphere. This reduces the pressure on the outer face of the emergency valve piston so that main reservoir pressure acting on the other face forces it to the extreme outer position, carrying with it the slide valve. This movement uncovers a port in the slide valve bush so that the reservoir has direct communication with the brake cylinder, in consequence of which air flows quickly from the reservoir to the brake cylinder until both pressures become equalized. In the same way, should a hose burst or uncouple, or pipe break, the resulting drop in emergency pipe pressure will insure an emergency application of the brakes as described. When releasing after an emergency application, by placing the brake valve handle in release position, the pressure is restored in the emergency pipe. The equalized pressure on either side of the emergency valve piston permits the spring to return the piston to its normal position, releasing the brakes. The release is designed to take place slowly after an emergency application to secure additional protection and to discourage the unnecessary use of the emergency position of the brake valve handle.

The emergency straight air brake system for a trailer includes the brake cylinder, emergency valve, auxiliary reservoir, conductor's valve and the necessary piping. An auxiliary reservoir is used on a non-motor trailer to furnish an independent supply of air for applying brakes on that car when an emergency application is made. It is connected to the emergency valve in the same manner as is the main reservoir supply pipe on a motor car. The auxiliary reservoir is charged from the emergency pipe by way of the emergency valve. The emergency pipe is connected to the reservoir pipe on the motor car by means of the motorman's valve for all positions of this valve except the emergency position.

The conductor's valve, an additional safety device, is connected to the emergency pipe and may be located at any convenient point in the car, if desired, with a cord attached to its handle and running the length of the car. When this valve is opened, the air in the emergency pipe flows directly through it to the atmosphere, setting the brakes in emergency.

Automatic Air Brake System. (Fig. 20.) The automatic air brake system is constructed so that the brake will be applied automatically in case of any accident which permits air to escape from the system. It differs from the straight air brake in that it requires a decrease in the train line pressure to apply the brakes, and an increase in pressure to release them, whereas in the straight air system air is admitted to the train line to apply the brakes and exhausted to release them. To accomplish this, there is added to each car, in addition to the straight air system, an auxiliary reservoir, in which is stored a supply of compressed air sufficient to operate the brakes on that car; a triple valve to which the train line, auxiliary reservoir and brake cylinder are all connected and which serves to control the flow of air (1) from the train line to the auxiliary reservoir when charging, (2) from the auxiliary reservoir to

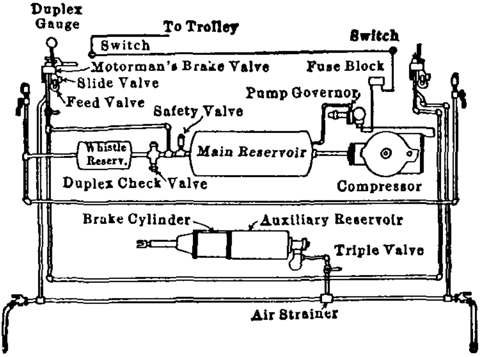


FIG. 20.—Automatic air brake system.

the brake cylinder when applying, and (3) from the brake cylinder to the atmosphere when releasing the brakes; and duplex air gages which indicate simultaneously the pressures in the main reservoir in the train line.

This system is capable of a great many refinements which may be added or omitted as requirements of a particular service may prescribe. The main points of difference between particular automatic air brake equipments will generally be found in the details of the triple valves, and the addition of pressure maintaining and reducing valves, which are essential in certain classes of grade work in order to prevent brakes leaking off. For the sake of simplicity these particulars have been omitted from this consideration.

Plain Triple Valve. Fig. 21 is a diagrammatic sketch of the plain triple valve which is used only on comparatively short trains or five cars or less. This figure represents the triple valve in the release position. The brake cylinder is in communication with the atmosphere by way of pipe B through ports 6 and 7, connected together by valve 3, and the brakes are released. Air from the train line enters the triple valve at *L*, and the pressure holds the piston 2 in the position shown; the by-pass at the top allows air to leak into chamber *R* and enter the auxiliary reservoir, charging it to train line pressure. This position of the triple valve prevents air from flowing to the brake cylinder from the auxiliary reservoir, as valve 3 closes port 8. When the pressure in the brake pipe is reduced below that in the auxiliary reservoir, either intentionally by manipulation of the motorman's valve or accidentally as in the case of a broken hose or pipe, the greater pressure from the auxiliary reservoir moves piston 2 of the triple valve to the left, which closes the by-pass just above the piston, moves valve 3 to the left, closes port 6 so that communication from the brake cylinder to the atmosphere is cut off, opens port 8 between the auxiliary reservoir and the brake cylinder, and applies the brakes. To release the brakes, pressure is restored in the train line from the main reservoir,

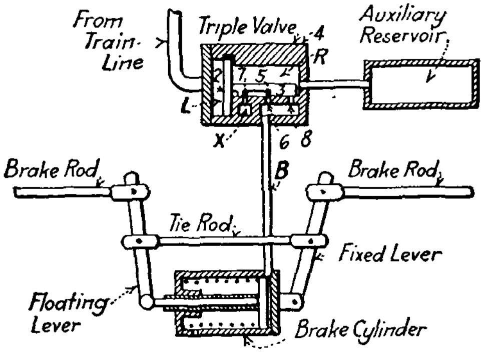


FIG. 21.—Diagrammatic sketch of triple valve and connections.

thus increasing the pressure at *L* above that remaining in the auxiliary reservoir and chamber *R* of the triple valve. Piston 2 of the triple valve then moves back to the right as shown in the figure, releasing the brakes and recharging the auxiliary reservoir as previously described.

A graduated release of the brakes may be obtained with this type of valve by piping the exhaust from the triple valve to the motorman's valve where a movement of the valve handle will release the air the same as in the straight air brake.

In practice, the service application requires only a slight reduction in train line pressure, and reference to Fig. 22 will show how the *graduating valve* functions to secure this end. In a similar manner to the previous simple explanation, a reduction in the train line pressure lowers the pressure in chamber *h* of the triple valve, and the higher auxiliary reservoir pressure moves piston 5 to the left, thereby opening communication between chamber *h* and the auxiliary reservoir through feed groove *i*. Attached to the piston stem is a pin valve 7, called the graduating valve, which when seated, closes communication between port *w* leading from chamber *m* to the graduating valve seat in the slide valve and the service port *z* leading from the graduating valve seat to the face of the slide valve. The first movement of the triple valve piston to the left unseats the graduating valve 7 so that air in chamber *m*, entering port *w*, flows to the service port *z*. There is a small amount of clearance between the slide valve 6 and the collar on the end of the triple valve piston stem, so that the first movement of the piston, which closes the feed groove *i* and opens the graduating valve 7, does not move the slide valve, but brings the collar on the stem against the end of the valve. Further movement of the piston then causes the slide valve to move until it has closed communication between brake cylinder port *r* and exhaust port *p*, and opened port *r* to the auxiliary reservoir through ports *z* and *w*. The piston then comes into contact with the graduating stem 8, and the resistance of the graduating spring combined with the reduction in auxiliary reservoir pressure then taking place prevents further movement of the parts. The valve is then in service position and air from the auxiliary reservoir flows through the service port to the brake cylinder, applying the brakes. While the pressure in the brake cylinder rises, that in

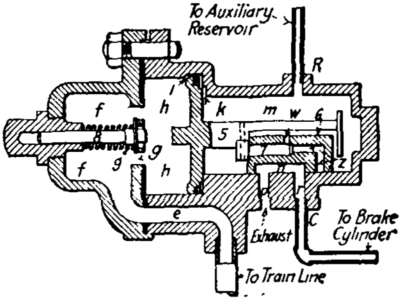


FIG. 22.—Plain triple valve.

the auxiliary reservoir falls and (the brake pipe reduction being 10 lb.) tends to become lower than that in the train line. As soon, however, as the pressure on the auxiliary reservoir side of the triple valve piston falls slightly below that on the train line side, the higher pressure causes the piston to move back toward its former (release) position, until the graduating valve is seated, closing communication betwen ports *w* and *z*. This prevents further flow of air from the auxiliary reservoir, the pressure in which is then practically equal to that in the train line, and at the same time prevents further movement of the triple valve piston toward release position, because the slightly higher pressure on the brake pipe side of the piston, which was able to move the piston and graduating valve alone, is not sufficient to move the slide valve also. Assuming that there is no leakage, the brake pipe and auxiliary reservoir pressures will remain balanced and the brake cylinder pressure held constant until the brake pipe pressure is further reduced, in order to apply the brakes harder; or increased, in order to release the brakes. A further reduction in pressure on the train line side of the triple valve piston below that on the auxiliary reservoir side causes the piston and its attached graduating valve to move as described for the first service application of the brakes. The slide valve, however, is already in service position, consequently as soon as the graduating valve is opened, air from the auxiliary reservoir flows to the brake cylinder and increases the pressure therein, thus increasing the pressure of the brake shoes against the wheels. If the brake pipe reduction is continued indefinitely, the auxiliary reservoir pressure will continue to fall and the brake cylinder pressure rise until they "equalize." This occurs at about 50 lb. cylinder pressure, when carrying 70 lb. train line pressure. After the pressures in the auxiliary reservoir and brake cylinder have "equalized" in this manner, air ceases to flow out of the reservoir and into the cylinder, because there is no longer any difference of pressure to cause a flow. Consequently, when the train line pressure is reduced below the "point of equalization," the brake cylinder pressure cannot rise above the "equalizing point," even though the train line may be reduced far below 50 lb. For this reason, therefore, nothing is gained by reducing the train line pressure below the "equalizing point" as explained above. Moreover, it is a needless waste of air and interferes with the proper release of the brakes.

An emergency application by means of the motorman's valve or a broken hose or pipe causes a sudden and rapid drop in the train line pressure. The pressure in chamber *h*, on the train line side of the triple valve piston, is reduced at a rapid rate, and the resulting difference of pressure is sufficient to move the piston and slide valve immediately to the extreme left and uncover the brake cylinder port *r* so that air from the auxiliary reservoir flows past the end of the slide valve directly through port *r* into the brake cylinder until the brake cylinder and auxiliary reservoir pressures are equalized. The pressure obtained in the brake cylinder is no higher than when a full service application is made, but the maximum pressure is obtained more quickly.

Quick-action Triple Valve. The quick-action triple valve shown in Fig. 23 is the same as the plain triple valve with additional parts and minor structural changes to permit the inclusion of these additional parts and retain the features of the plain triple valve. The preceding description of the plain triple valve, except for the emergency application, applies equally to the quick-action triple valve. When the piston and slide valve of the quick-action triple valve move to emergency position, as described for the plain triple valve, the port *s* in the slide valve registers with port *r* in the seat, allowing air to flow from the auxiliary reservoir to the brake cylinder. Port *s* is small, however, and in this position the slide valve also opens port *t* in its seat, allowing air to flow from chamber *m* through port *t* to the chamber above the emergency piston 8. The other side of the emergency piston 8 is connected to the brake cylinder, in which there is no air pressure, consequently the emergency valve is forced downward, pushing the emergency valve 10 from its seat and allowing the air in chamber *Y* above the check valve 15 to flow past the emergency valve 10 to chamber *X* and the brake cylinder. Train line air in *a*, below the check valve 15, then raises the check valve against the resistance of its spring 12 and also flows to the brake cylinder through the passages mentioned. During an emergency application, therefore, the quick-action triple valve supplies air to the brake cylinder from the train line as well as from the auxiliary reservoir. Port *s* is small, so as to restrict the flow of air from the auxiliary reservoir to the brake cylinder and thus allow as much air as possible to enter the brake cylinder from the train line. Approximately 60 lb brake cylinder pressure is, therefore, obtained on emergency applications, the air from the train line increasing the cylinder pressure about 20 per cent above the maximum obtainable with a full service application. Not only does the air vented from the train line give a higher brake cylinder pressure, but it causes a local drop in train line pressure at the triple valve which causes the next triple valve to apply in "quick-action," and it the next, thus transmitting the quick action from triple valve to triple valve serially throughout the train in a very short time, with the result that all the brakes in the train are applied in a

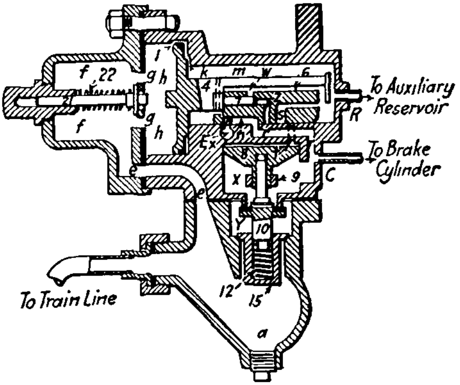


FIG. 23.—Quick action triple valve.

fraction of the time which would be required if all the valves were plain triple valves and all the train line reduction had to be made at the brake valve. The release after an emergency application is made the same as after a service application, except that it requires a longer time, the train line having to be recharged from zero to slightly above the pressure in the auxiliary reservoirs before the triple valve pistons can move to release position. The quick-action triple valve is designed to be used on freight trains of considerable length, its function being to apply the brakes on the rear cars in emergency so quickly that the taking up of slack is avoided.

Combined Straight and Automatic Air Brake. This system includes two sets of motorman's valves, operated by the same handle, for the control of each system. The straight air brake, operating with pressure between 55 and 70 lb. per sq. in., applies and releases the brakes on the front car independently of the brakes on the other cars. The automatic brakes operate with air pressure from 100 to 110 lb. per sq. in., and apply the brakes on the remainder of the train independently of the brakes on the front car. The chief advantage of such an arrangement is the possibility of holding the brakes on the locomotive applied while the train brakes are released for the purpose of recharging the auxiliary reservoirs.

Electropneumatic Brake System. In the electropneumatic brake, the ordinary features of the automatic air brake are retained, but the application and release of the air pressure is governed by electromagnetic control in a manner somewhat similar to that of the electropneumatic control of the motor circuits in multiple unit control operation. A special form of triple valve is used in which the admission of air to the brake cylinders is governed by electromagnetic valves. By proper combinations of electric circuits, the brake cylinder pressure may be built up to the maximum, may be held in the cylinders, or may be wholly or partially exhausted. The motorman's valve is composed of two distinct parts, the electric placed above the pneumatic part. The electric parts consist of a rotary drum with contacts attached to the same shaft as the rotary valve and with corresponding fingers on the body of the valve. The position of the handle is identical for electric and pneumatic release, service and emergency. With the electric features operative, the service magnet valves admit air from the auxiliary reservoir to the brake cylinder on each individual car at the desired rate, and at the same time the train line pressure is reduced by the motorman's valve so that if the electric control is inoperative, the pneumatic control will operate automatically to apply the brake. If any of the service application magnets in the train are inoperative for any reason, therefore, the brake will apply pneumatically on that particular car.

In comparative tests of service stops from an initial speed of 40 miles per hour, an 8-car rapid transit train with pneumatically controlled brake was stopped in 40 seconds in a distance of 1290 ft. In this operation a graduated action was obtained by increasing the train line pressure to release the brakes and then partially reapplying them. The electrically controlled brakes brought a 10-car train to rest after 20 seconds in a distance of 700 ft. In the operation of the electrically controlled brake, a much more effective graduation of the release was obtained. Due to this feature the maximum braking effort can be increased to a value which would be dangerous for the ordinary automatic brake. The final pressure at the end of the stop was less with electropneumatic control than with the other type of control. The emergency stops in the same tests showed the advantage of simultaneous action of the brakes on all the cars to a greater degree than the service application tests. The time was 22 seconds and distance about 625 ft. for automatic air braking of the 8-car train, and 11 seconds and 350 ft. for electropneumatic braking of the 10-car train.

The electropneumatic brake system adds to the pneumatically-operated brake certain advantageous features otherwise impossible of attainment, namely: simultaneous and uniform response of all the brakes in the train, which means the ability to obtain the desired results with the least skill and experience, regardless of the length of train; double protection against delays to traffic due to brake failure, since the pneumatic brake is always in reserve ready for use, if required; maximum efficiency and safety due to simultaneous operation of all the brakes in the train, in both service and emergency application, and the ideal flexibility of manipulation; economy in air consumption, and maintenance of brake cylinder pressure at will. The shorter service stop thereby made possible means that power may be shut off sooner and the train allowed to coast for a considerably longer time before applying the brakes and making the stop at the same point. Thus, for the same power consumption, the electropneumatic brake makes it possible to maintain higher average speeds, shorter schedules and an increased traffic capacity with the same number of cars; or the same traffic capacity with fewer cars; or enables the same average speeds, schedules and capacity of road to be maintained with the expenditure of less power. It should be borne in mind that the electropneumatic control of the brakes is a control only and that it must depend for its fundamental safety and protective features upon the pneumatic brake with which it is associated.

For single cars or short trains, electric control of brakes, although ideal and entirely practicable, has not appeared to be necessary or justified by the requirements of such service. It adds nothing to the effectiveness of the individual brake, its only function being to save time in applying the brakes by eliminating the serial time element inseparable from an impulse transmitted pneumatically from car to car. Where trains are relatively short, therefore, pneumatic transmission is so nearly instantaneous that the stop is not materially lengthened thereby, nor is there sufficient slack or time element to cause objectionable shocks from this cause. When the carrying capacity of a given trackage is taxed to its limits, every means available for increasing the passenger miles per car becomes of prime importance, and the value of electrical control of brakes becomes apparent. This does not add directly to the effectiveness of the brake on any one car, but eliminates the serial application effect on a train of several cars. In doing this, however, it accomplishes two results which are of consequence in long train service, namely, the elimination of slack action between cars and the saving of the time and distance through which the train would otherwise run while the pneumatic brake action was being serially transmitted from the head to the rear of the train. It is quite clear that these features become of greatest value where trains are long and run on close headway with frequent stops and the traffic capacity of the road severely taxed. For these reasons the electric control has been found especially desirable for such service as in the subway systems of Philadelphia and New York and the elevated of Boston.

Safety Devices of the Safety Car. Features that may be included on the safety car to make operation safe, proper and easy, are: (1) The car cannot be started with the doors open. (2) Opening the doors causes a brake application sufficient to stop car. (3) Unintentional removal of the operator's hand from controller handle causes emergency brake application. (4) Brakes cannot "leak off" while the car is standing with the doors open. (5) Automatic emergency application results if either of the main pipes is ruptured.

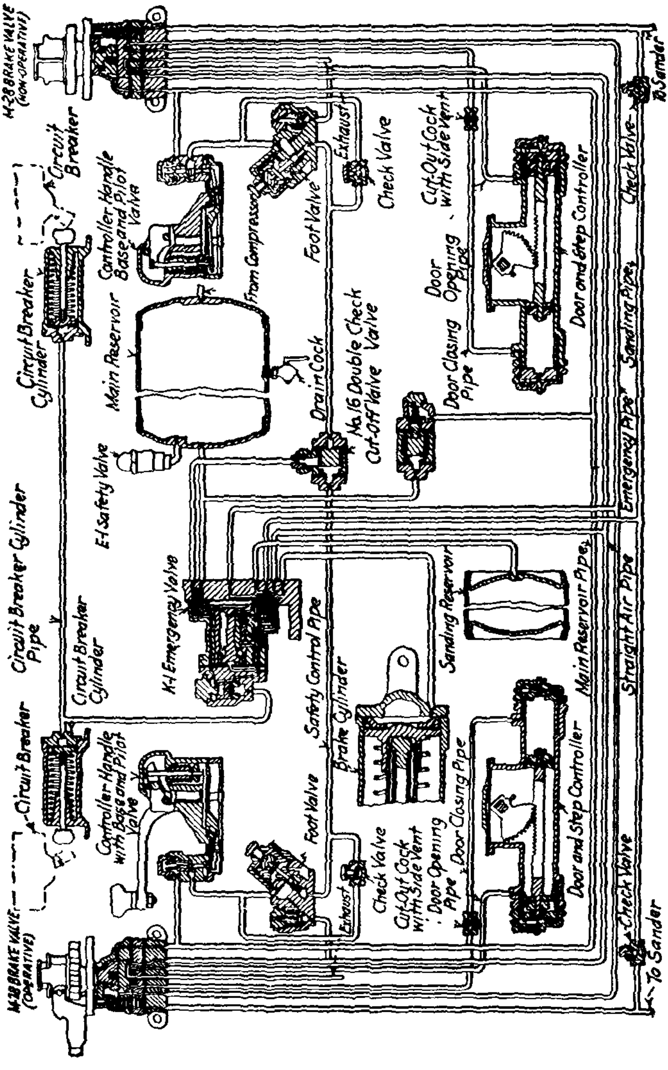


Fig. 24.—Typical layout of operating devices and control for safety car

(6) Power is cut off, sand is applied, and doors become hand operated whenever an *emergency application* is caused by releasing the controller handle. (7) If service application "leaks off" while changing ends, an emergency application results. (8) Rupture of platform piping causes an automatic emergency application, and the damaged piping is cut off so as to prevent exhaustion of the main reservoir. (9) To release brakes after a stop, the controller handle must be *pressed down*. (10) Ends cannot be changed without making a brake application or an automatic emergency application results. (11) Sand is automatically applied in an emergency. (12) No increase in the number of manipulative handles. (13) Doors are air operated. (14) Sanding is accomplished by air without the use of a special operating handle. (15) Intentional and temporary release of the control handle may be made without emergency application resulting. Fig. 24 shows a typical layout of the various air devices and their connections to produce the results enumerated above. Many cars are in operation with safety devices which do not include all those here shown, or which include some such devices in different form. Reference may be had to *Electric Railway Journal*, Vol. 55, pp. 788 et seq., for detailed description of the various devices and their operation.

Brake Inspection (A.E.R.E.A. Approved Practice). Start the air pump and allow it to pump to its maximum capacity; see that brake valve handle is in release position, and where automatic air is used see that gage hands show a difference between train line and auxiliary of 20 lb. If they do not, the governors need to be reset. Apply brake to show reduction of 40 lb.; place brake valve handle to lap position; see that air gage operates properly and that no leaks are in or around the brake valves or pipes leading thereto; examine all pipes, reservoirs, triple valves, cylinders, etc , while brake is set, and see that none are leaking and that brake does not release while the brake valve handle is in lap position. If the cylinder piston has a travel of more than 5 in. an adjustment of brakes is necessary. Inspect all shoes and see that they are in alinement with the wheel and that none are broken, and renew those that will not give sufficient wear until the next inspection. In renewing brake shoes put shoes of the same thickness on opposite wheels, be they either old or new. See that all brake shoe keys are in place and that none are lost or broken; examine shoe heads and see that none are lost or broken, and that all pins, bolts, etc., that hold heads to the beam or truck levers are not unduly worn; that all bolts, cotter pins, nuts, etc , are in good condition. Examine brake beams and see that none are cracked, broken or bent, and that all bolts, pins and holes are not unduly worn, and that all cotter pins and nuts are in place. Examine all hangers and pins connecting brake shoe head and beam to truck and see that all are in good condition, and that none of the pins and hangers are unduly worn so as to cause brakes to grab or chatter; special attention must be paid to all cotter pins in all parts of the brake rigging. Examine all turn-buckles and see that none of the threads are stripped and that all adjusting and jam nuts are tight and in their proper places. Adjust brakes so that cylinder piston will not travel more than 4 or 5 in. When brake is in release, see that none of the shoes bind the wheels, that release springs operate properly so that brakes will be free when released. Examine all pull rods for cracks or flaws, lubricate all pins in pull rods, levers and slides; set hand brake and see that it is in good condition; see that brake staff and chain are not unduly worn; that rod, pins, etc., are in good condition. Where slack adjuster is used, see that it is placed to its minimum of travel before any adjustment of brakes; see that it is operating properly and that it has not traveled to a maximum position, leaving the correct piston travel. Drain all reservoirs daily.

Maintenance of Motorman's Valve. The seats of the rotary or motorman's brake valve are subject to the collection of dirt largely because of the passage across them of air which is more or less laden with dust. When this collection causes the valve to work hard or the valve seat becomes badly scored, an attempt to grind the surfaces with emery should not be made. Grinding with emery generally causes leaks between the ports and an attempt to grind out cuts or scores will usually make the valve worse. When the condition of the valve becomes so bad that the valve cannot be kept in service it should be machined and scraped by hand to a perfectly flat surface, using a face plate to locate the high spots.

Brake valves should be *lubricated* at regular car inspection periods. To oil a brake valve, it is necessary first to exhaust the air from the valve; the oil should then be applied through the oil plugs, the valve stem pushed down a few times, and the valve operated to work the oil onto the various surfaces. Lost motion or play between the handle and stem prevents the proper registration of parts and should be eliminated.

Emergency valves, feed valves and **triple valves** should be completely disassembled and thoroughly cleaned at regular overhauling periods. The only part of the feed valve requiring lubricant is the slide valve which should be lubricated with dry graphite. In the emergency and the triple valve, lubricate the slide valve with dry graphite and the piston bushing with a drop or two of oil.

Inspection of Air Compressors (A.E.R.E.A. Approved Practice). The frequency of inspection necessary for an air compressor depends upon the service required of the compressor. The duties of air compressors on city and interurban cars vary greatly. For instance, a car may be equipped with air doors, electropneumatic control and air brakes. The average number of stops of a car of this character in city service may be every 1500 ft. A compressor under these conditions would necessarily have to be overhauled more often than one which makes stops two or three miles apart, has no air doors or electropneumatic control. The inspection period for the former class of service should be 600 miles and that for the latter class should be 1200 miles. Air compressors should be inspected as follows: Oil plug should be removed and oil added to replace what has been lost in service each inspection day. Carbon brushes removed and inspected each inspection day. Brush holder tension inspected each inspection day. Brush holder wiped off each inspection day. End of commutator wiped off each inspection day. Hair should be taken out of hair strainer and strainer washed in gasoline every thirtieth inspection day. Valves should be taken out and cleaned in gasoline each thirtieth inspection day. Exterior of pump should be wiped off each inspection day. Compressors should be thoroughly blown out with compressed air each tenth inspection day.

Air Compressor Tests (A.E.R.E.A. Approved Practice). With all air reservoirs empty and brake valve on release position, start air compressor and note length of time taken in pumping up, and pressure at cutting-out point. A test for leakage should then be made, leaving apparatus in same condition as above, and noting number of pounds drop in one minute.

The Connecticut Company overhauls its rolling stock on a 100,000 mile basis. The overhauling of car equipment includes air compressors, which are tested on the cars before and after the overhauling. For convenience in testing, a tank of exactly 5 cu. ft. capacity has been mounted on a truck. This is provided with a gage and a valve for bleeding off the air pressure. Compressors are tested for the time necessary to pump up to a given pressure and also for the number of revolutions. After connecting the tank to the compressor, the test consists of pumping up the pressure in the stand tank to 90 lb. This is then reduced to 40 lb. and again pumped up. When the pressure reaches 60 lb., a revolution counter is applied to the armature shaft and the number of revolutions required to pump from 60 lb. to 90 lb. is counted. A table has been prepared from the manufacturer's rating of the compressor, to show the number of revolutions under proper conditions. A glance then will show whether there are abnormal conditions that require attention. After overhauling, compressors must conform to the requirements within certain definite limitations in order to be passed for service.

Lubrication of Air Compressor (A.E.R.E.A. Approved Practice). Air compressors should be lubricated weekly. Before removing plugs, wipe all dirt and surplus oil from around the openings, then remove plugs and do not fill to an extent to overflow, thereby allowing the oil to collect with the dust and dirt around the pump frame and case. See that the plug threads are in good condition and oil-tight when closed.

Overhauling Air Compressor (A.E.R.E.A. Approved Practice). With a compressor designed for sufficient capacity for the service it is to perform, an overhauling for the former class of service outlined above (Inspection of Air Compressors) should be made every 60,000 miles, and for the latter class of service every 120,000 miles. In overhauling, the compressor should be taken from the car to a bench where the armature should be removed, oil drained from crank case and from all bearings outside of crank case, and bearings thoroughly washed out with gasoline. Crank shaft and connecting rods should have slack taken up on them or bearings re-babbitted or relined with bronze bearings, as the case may be, if wear is excessive. Head should be taken from the compressor, ports scraped out, hair removed from hair strainer and washed in gasoline, and valve and valve seat should be re-ground. Armature should be blown out, cleaned up and breakdown test applied. Also commutator trued up. Mica retaining rings should be painted with insulating paint. Armature painted with a coat of oil-proof insulating paint. Brush holders should be cleaned in gasoline and overhauled, replacing worn tips and shunts that have broken strands. Field coils should be taken out and insulation carefully looked over, and replaced if necessary. Inside of the motor shell should be carefully cleaned with gasoline to get oil and dirt out of shell The impregnation of pump field coils is recommended as tending to eliminate field troubles and greatly prolong the life of the fields The piston should be removed from pump, and rings removed from groove and carefully cleaned; also, groove should be carefully scraped out Springs from piston rings should be tested to make sure that they have not lost their tension. If, when pump is re-assembled and started to work under pressure for 5 minutes, it be then disconnected and allowed to run free and oil is discharged from the outlet, it is an indication that the rings are not tight enough in the cylinder, and springs of greater tension should be put in the rings The insulators placed between compressors and air piping on car should be taken from the car and cleaned and given a breakdown test of 1000 volts. The oil which was removed from the compressor when it was removed for inspection should be run through a filter and returned to compressor, enough oil being added to take the place of the dirt, etc., which it contained when removed In this connection care should be taken to use an oil which does not contain asphalt or which carbonizes when the pump is given hard usage

Storage Air Brake System. In the storage air brake system the service reservoir is supplied with air from storage reservoirs (generally two) carried on the car. These storage reservoirs are charged with air at high pressure from compressor stations along the line. The air is passed through a pressure-reducing valve between the car storage reservoir and the service reservoir, and thus its pressure is reduced to a value proper for service in the braking system.

Whether or not the storage system should be used rather than the individual car compressor for a given service must be decided from local conditions. The storage system may in some cases be cheaper than the individual car compressor system, but the many advantages in having the car self contained, as by the use of the individual car compressor, make the use of the latter system almost universal. Moreover, air operated auxiliaries demand air in addition to that for the brakes and such demands will, in most cases, make the employment of the storage system impracticable.

Size of Air Compressor Required for a Given Service (A.E.R. E.A Approved Practice). An air compressor should be of a size such that it will not be required to operate more than one-third the time. In connection with the above, due consideration should be given the air requirements of air operated doors, sanders and other auxiliary equipment.

Air Used and Electrical Energy Required to Drive Compressor. The data in the following table were taken from the Report of the Electric Railway Test Commission, 1906, and were based on tests made by the Commission on double truck city cars in operation in St. Louis

	Dry track	Wet track
Average number of stops per mile	4 1	4 5
Schedule speed of car, miles per hour	9 5	9 3
Maximum speed of car (approximate), miles per hour	16 0	16 0
Average volume of free air used, cubic feet per car per stop	1 68	1 51
Average volume of free air used, cubic feet per ton per stop	0 076	0 067
Electrical energy for compressing air, watt-hour per car per stop	6 74	5 86
Electrical energy for compressing air, watt-hour per ton per stop	0 306	0 261

The amount of air required per stop for interurban car operation, while greater, is not in proportion to the speed from which the stop is made. As a matter of fact, it should not be greater merely on account of the higher speed, as after the brakes are applied, only enough air is needed to make up for leakage. As the duration of the braking period may be greater for the interurban car, the possibility of leakage is correspondingly increased The air actually required by the brake in making the stop often is a small percentage of the actual amount used, due to improper brake valve manipulation, leakage of piping and brake cylinders, air operated devices such as bell ringers, door engines, whistles, etc.

Capacity of Air Reservoirs. The air reservoirs should have a capacity sufficient to supply air for three or four applications without reducing the pressure more than 12 or 15 lb. Otherwise every ordinary application of the brake will throw the compressor into action, thus keeping the latter in a constant state of starting and stopping, and causing unnecessary wear to both compressor and governor The Westinghouse Traction Brake Company recommends the following sizes of reservoirs:

for 8 in. brake cylinders, 16 in. × 48 in. reservoirs;
for 10 in. brake cylinders, 16 in. × 60 in. reservoirs;
for 12 in. brake cylinders, 16 in. × 72 in. reservoirs.

The lengths given above are overall.

General Characteristics of a Good Air Compressor. An air compressor should be reliable, of light weight and compact construction, should be protected from dirt and water, should be so constructed as to be easily inspected, lubricated, overhauled and repaired, should run with the least possible amount of noise and vibration, should have high efficiency of operation and should have a low cost of maintenance.

General Types of Air Compressors. The independent motor-driven air compressor most commonly used for electric railway service may be considered to be of two general parts, namely, the motor and the compressor. If the motor and compressor are geared together, the machine is known as the geared type compressor. If the motor shaft is direct connected to the compressor connecting rod, the machine is known as the gearless type compressor. Because of the absence of reduction gearing a gearless type air compressor differs mainly from the gear type of the same capacity in that the piston speed (revolutions per unit time) of the former is greater than that of the latter, consequently the piston displacement of the former is the less. Compared with a gear type air compressor of the same capacity, a gearless type compressor weighs about one-half, occupies about one-half the space and has a very low overall height. The stresses in the gearless type are much lower, the maximum transverse thrust on the crank being only about one-fifth as much as that in the geared type.

Installation of Air Piping to Prevent Freezing. Great precaution must be taken in the installation of air piping between the compressor and the main reservoirs to prevent freezing of the condensed moisture. This is especially true with pneumatically operated apparatus which employs valves with small openings. Freezing at very low temperatures is not so troublesome as at a little below 32 deg. F., because at lower temperatures most of the moisture has been frozen out of the atmosphere. The installation of the air piping should be such that the maximum amount of moisture is retained in the main reservoir. No pockets should exist where moisture is likely to be retained, as freezing will occur whenever a small body of water collects in the piping. The pipe between the compressor and the main reservoir, as well as the pipe between the two main reservoirs, should be at least 25 ft long, and when the length of car does not permit a straight run, the pipe should be made up in a series of return bends, or with several pipes as multiple paths connected into common headers. For a given length of piping of the same diameter in the two systems, the single path will require a greater velocity and hence a greater loss in pressure for the delivery of a given quantity of air. This means that, for a given main reservoir, the compressor must pump against a greater pressure with the single path system, because of the greater friction. This higher pressure is accompanied by a higher temperature and consequently, even though the radiating surface of both arrangements of piping are the same, the air from the single path of pipe, in view of its higher initial temperature, must necessarily be warmer when reaching the main reservoir than with the manifold arrangement. The extent to which this difference exists depends upon the size of pipe and upon the quantity of air delivered in a given unit of time.

The Cleveland Railway is using air coolers made from a 36 in. section of standard 5 in. pipe, as shown by Fig. 25. It is inserted in the air line between the compressor and the main reservoir to cool the air sufficiently in passage so that all excess moisture will be dropped. The warm air, on being received from the compressor, enters the cooler and is deflected by a baffle so that it follows along in the space between the wall of the 5 in. pipe and another internal 1½ in. pipe concentric with it. In this way the air is kept in contact with the larger pipe where the cooling effect is obtained. The air leaves through twelve ¼ in. holes in the further end of the inner pipe. Accumulated moisture is drawn out through a drain cock in the bottom of the cooler. The only special castings required are the internal baffle and the cap on the reservoir end of the cooler. Both of these parts are threaded to receive the inner 1¼ in. pipe.

In connecting the feed pipe to the reservoir, particular attention should be given to make sure that the connections do not give a reduction of pipe area at any point, as a change in area increases the possibility of freezing. All piping from the reservoir to the various pieces of apparatus should be arranged to drain back into the reservoir as far as possible, and when this is impossible it should drain away from the apparatus.

The best place to mount the intake is on the roof of the car, as it is then possible to obtain cool, clear air. With the intake mounted inside of the car, a greater percentage of moisture is obtained, which is always to be avoided.

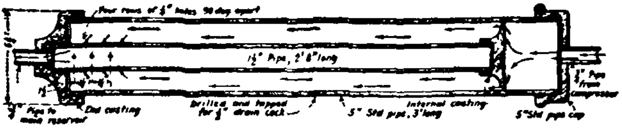


FIG. 25 —Cleveland air cooler.

Air Brake Hose. The bursting test of air brake hose is made by subjecting it to a hydraulic pressure of 200 lb. per sq in, and under this pressure the hose should not show any signs of leakage nor develop any defects, and the maximum expansion on the circumference should not exceed the following:

Nominal size of hose	¾ in.	1⅛ in.	1⅜ in.
Minimum expansion	½ in.	⅝ in.	¾ in

The hose must then withstand the following hydraulic pressures for a period of ten minutes:

Nominal size of hose	¾ in.	1⅛ in.	1⅜ in.
Pressure, lb per sq. in	600	600	500

The above specifications were taken from the Manual of the Am. El. Ry Eng. Assn. Other specifications such as refer to manufacture, porosity test, friction test, stretching test, tension test, are also included

Magnetic Brake. A form of magnetic brake in which are embodied both a track and wheel brake is shown by Fig. 26. A track brake-shoe is placed between the two pairs of wheels and is drawn to the rails by an electromagnet which is suspended from the car, thereby not merely adding its friction to the friction of the wheel brake, but also actually increasing the rail pressure of the wheels to the extent that the supporting springs for the track shoes and magnets are in tension through the descent of the track shoes to the rails. The electromagnet *a*, dividing the track brake-shoe *b* into two parts, is secured by pins to the two push rods *c*, and suspended at a proper distance above the rails by the adjustable springs *h*. The push rods are secured by pins to the lower ends of the brake levers *d*, which are connected at their upper ends by the adjustable rod *g* and are pivoted at an intermediate point to the brake shoe heads *e*, carrying the wheel brake shoes, and the hanger-links *f*, suspended from the truck frame. The push rods *c* are telescopic, so that a movement of the track shoe toward the right, relative to the truck frame, causes the wheel brake shoe at the right to be applied to the wheel and the connection *g* to be moved to the left, thereby applying the wheel brake shoe at the left, the stop *i* preventing the lower end of the brake lever at the left from following the track brake shoe. A relative movement of the track brake shoe to the left is accompanied by application of the wheel brake shoes through corresponding movement of the parts in the reverse order. The brake controlling device may be incorporated in the running controller or may be a separate device, placed by its side and operatively interlocked with it, so that neither can be caused to inter-

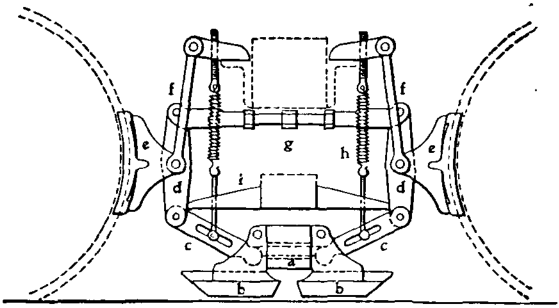


FIG. 26.—Magnetic brake

fere with the operation of the other. In the operation of the apparatus, the current is supplied by the motors running in multiple as generators with the fields reversed (the trolley current being cut off) and is divided between the electromagnets and controlled by a rheostat so as to cause the track brake shoes to be drawn upon the rails with a force proportionate to the braking requirements. In order to limit the amount of current through the track brake coils to prevent slipping of the wheels, it is common practice to provide a regulating resistance which, in connection with a regulating relay, shunts a portion of the field current when the braking current reaches an excessive value. Resistance is also inserted in the connection between the two motors in order to decrease any tendency toward unbalancing. In some cases it has also been found necessary to use a demagnetizing shunt to release the track brakes when power is again applied in the forward direction. The frictional resistance of the rails to the motion of the track shoes causes the wheel brakes to be applied with corresponding force. Thus, to the ordinary retardation of the wheel brakes is added that of the track brake and also the back torque of the motors, which latter, however, is practically limited to compensation for the rotative energy of the motor and car wheels. The force of application depends primarily upon the current and upon the electromagnets operating the brake shoes. The attractive force of the rails upon the magnets is under the control of the motorman up to a limit of about 150 lb. per square inch of brake shoe surface in contact with the rails. The strength of the magnet is limited by the sectional area of the rail acting as armature, and where the weight of the car makes a magnet of greater strength desirable, the track shoe is divided into three parts, instead of two, and wound to form a three-pole magnet, or two combined two-pole electromagnets with one common pole. The friction of the track brake shoe may also be adjusted to some extent through the angular inclination of the push rods *c*, by which some of the weight of the car may be thrown upon the track shoes, the levers *d* being correspondingly adjusted to reduce the wheel brake shoe pressure in proportion as the weight is transferred to the track shoe. The current declines with the speed during a stop, thereby offsetting the increased coefficient of friction at the lower speeds. In bad weather, when the condition of the rails is likely to be accompanied by wheel sliding, the braking force operating the wheel brake is correspondingly reduced, so that the force of application of the wheel brake is automatically proportioned to the rail friction which rotates the wheels. But, in addition to this valuable feature, if by chance the wheels should slide upon the rails, the interruption of wheel rotation is accompanied by the cessation of the track magnet current, through which the pressure of the brake shoes upon the wheel is instantly relaxed and rotation of the wheels is resumed, without injury or serious loss of time.

A large amount of special track work is a handicap to the operation of magnetic brakes, increasing maintenance on both the magnetic brake shoe and the track. To obtain an effective magnetic brake it is often necessary to design the truck especially for this feature, which involves additional cost and often increases the length of wheel base. The application of magnetic brakes on cars has resulted in the development of many detail designs caused by the differences in schedule, in car, truck and rail construction, in size and characteristics of motors, and in desires of the local operator. The magnetic brake is most useful on cars used in heavy grade operation.

Braking by Regeneration. The process of saving energy of a moving train that would otherwise be consumed in heating the brake-shoes and car wheels is discussed under "Regeneration" (see page 203). In furnishing the energy of regeneration the speed of the train is reduced without wear and tear and excessive heating of brake rigging, brake shoes and car wheels. The possibility of accident from these sources is thus reduced, and on long mountain grades the safety of operation due to the braking feature of regeneration may be of more importance than the saving of energy. The application of air brakes during the process of regeneration may reduce or destroy the regeneration and its braking action and might bring an excessive load on the brakes by causing the motors to take current from the line, but this may be avoided by employing an automatic interlock between the two systems so that the "service" air brake cannot be applied while the regenerative brake is in use.

Braking by Reversing Motors. A retarding force may be applied to the train by reversing the motors and applying current to them through a portion of the starting resistance. This method should be used only in an emergency, as it strains the car equipment, and if the direction of rotation of the wheels is reversed the braking action will be of low efficiency because of the low value of coefficient of friction between the slipping wheels and rails.

Braking by Bucking Motors. If two series motors whose armatures are revolving but to which no current is supplied from an external source have their connections reversed and then placed in parallel, they will tend to operate as series generators; the one of higher potential will continue to act as a generator and drive current through the other which will consequently act as a motor. This action will retard the motion of the armatures. The steps necessary to make use of this action in retarding the motion of a car are: Open circuit breaker, throw reverse lever to the position corresponding to motion opposite to that of the car and move controller handle to a parallel notch. On a four motor car, where pairs of motors are permanently connected in parallel, the last step noted is unnecessary. It should be noted that if the current supply to a car ceases as the car is ascending a grade and it is desired to use this method of braking to keep the car from backing down the grade, the reverse lever should be left in the position corresponding to the forward motion of the car up the grade. This method of braking should be used only in emergency.

Brake Shoe Wear. Concerning brake shoe wear, the 1910 Committee on Brake Shoe Tests, M.C.B.A., recommended that on the cast iron wheel the shoe wear be determined by making 100 applications of the shoe to the wheel, under a pressure of 2808 lb., and at a constant wheel speed of 20 miles per hour, at each application the shoe to be in contact with the wheel during 190 revolutions and out of contact during the succeeding 610 revolutions. That, under these conditions, the shoe shall lose in weight not more than 0.8 lb. for each 100,000,000 ft-lb. of work done. That, on the steel tired wheel, the shoe wear be determined by making 10 stops from an initial speed of 65 miles per hour and under a shoe pressure of 12,000 lb. Ten minutes shall intervene between successsive applications of the shoe. That, under these conditions, the shoe shall lose in weight not more than 4 lb. for each 100,000,000 ft.-lb. of work done.

That committee recommended the adoption of the following suggestions in place of the then existing specifications for brake shoes:

a. Shoes shall be tested for coefficient of friction and for wear upon the Master Car Builders' Association testing machine, or upon a machine with equivalent characteristics.

b. Shoes shall develop upon the cast iron wheel, in effecting stops from an initial speed of 40 m.p.h., a mean coefficient of friction of not less than 22 per cent when the brake shoe pressure is 2808 lb., 16 per cent when the brake shoe pressure is 6840 lb.

c. Shoes shall develop upon the steel tired wheel, in effecting stops from an initial speed of 65 m.p.h., a mean coefficient of friction of not less than 12½ per cent when the brake shoe pressure is 6840 lb., 11 per cent when the brake shoe pressure is 12,000 lb.

d. No limitation is placed upon the rise in coefficient of friction at the end of the stop.

General Performance of Brake Shoes. The following deductions are based on laboratory tests made in connection with the Pennsylvania-Westinghouse tests, 1913:

(*a*) The generation of the retarding forces and consequent absorption of the energy of the moving train is dependent upon but a very small quantity of brake shoe metal. (*b*) The actual bearing area rather than the total face area of the shoe is the important factor in brake shoe performance. (*c*) The magnitude of the bearing area changes throughout the stop and is greatest near the end of the stop. (*d*) The bearing area shifts continuously from one portion of the surface to another during the stop. (*e*) The principal factor in producing high friction for any given braking condition is the frequent shifting of the bearing area from the heated to the cooler spots over the face of the shoe. (*f*) Slotted shoes or shoes that are cracked are more flexible than solid shoes and the bearing area shifts more readily than in the case of solid shoes. (*g*) With shoes of the same type and approximately the same hardness, the wear per unit of work done is less with the slotted shoe than with the solid shoe. The stops with slotted shoes were always shorter and the mean coefficient of friction higher than with solid shoes. (*h*) The shifting of the bearing area will tend to be more rapid if the size provides more available area for shoe bearing. (*i*) The greater the pressure per square inch of bearing area, the lower will be the mean coefficient of friction. (*j*) Flanged shoes provide more available area for bearing than unflanged shoes.

"Squealing" During Braking. Opinions based upon tests and given in testimonies at hearings on street car noises agree that "squealing" during braking is due to rubbing of wheel flange against rail head as the freedom of the wheel to seek the path of least resistance is restricted by the brake shoe pressure. The greatest amount of noise is given off by steel wheels. Reduction of the noise has been sought in lubricating the flange by the use of a brake shoe of special composition or having inserts of a lubricating material. Among these materials are lead and a composition of graphite and asphalt. The ideal lubricant is the one which may be used at the least cost and which will not reduce the friction between brake shoe and wheel.

Wear, Relative Efficiency and Cost of Various Types of Brake Shoes. The following conclusions were drawn from the results of stand tests and a test by a year's service with 800 brake-shoes on the Brooklyn Rapid Transit System by George L. Fowler. Four general types of shoes were used in these tests; they are designated as "A," "B," "C," and "D," respectively. "A" was a plain, hard cast iron shoe containing 1.67 per cent combined carbon and 1.36 per cent graphitic carbon. The microscopic structure of the "A" shoe seemed to be between malleable iron and gray cast iron in that the graphite appeared both as nodules and very small flakes. "B" was of cast iron, with ends chilled, containing 1.20 per cent combined carbon and 1.85 per cent graphitic carbon. The microscopic structure of the "B" shoe would probably be classified as mottled iron. It differed, however, from the typical mottled iron in the more even balance of the pearlite and graphitic carbon by which the structure more nearly resemb'ed that of hard iron. "C" was of cast iron with an expanded metal filling. "D" was of cast iron with chilled iron inserts in the part bearing on the tread. The coefficient of friction and wear for each of the general types of shoes given in the following table were secured from tests in a brake shoe testing machine by applying the brake shoe under a pressure of 6840 lb. and a speed of 20 miles per hour:

Coefficient of Friction and Wear

Type of shoe	Coefficient of friction, per cent	Inches, wear per 100,000,000 ft.-lb.
A	39.8	1.79
B	31.2	0.61
C	41.9	1.83
D	29.1	0.47

Conclusions: (*a*) In order to avoid excessive costs for the application of brake shoes the weight should be limited so that no individual shoe should weigh more than 24 lb. To avoid an excessive loss of weight in the percentage of scrap the minimum weight of individual brake shoes should be 20 lb. (*b*) There appears to be a close relationship between the microscopical structure of cast iron and its wearing qualities. Hence the foundry practice should be such as to secure a structure that is closely granular, of uniform texture and with the combined and graphitic carbon pretty evenly balanced. The graphitic carbon should be in the form of nodules rather than flakes. (*c*) There is a general relationship between the coefficient of friction as determined on the testing machine and the stopping qualities of a brake shoe in service. (*d*) As far as the safety of a car is concerned, owing to the ability to stop from speeds common in surface car service, there is no appreciable difference between any of the shoes tested. The extreme variation at 15 miles per hour averages but 5 ft., or about 7.7 per cent. At 20 miles per hour it is but 17½ ft., or 16.26 per cent. At lower speeds the difference is correspondingly less. (*e*) Brake shoe wear averages from 3.75 to 6.5 lb. per 1000 wheel miles. The scrap weights of brake shoes on the surface lines should not be allowed to drop below 6¼ lb. per shoe. When the scrap weight has reached 5½ lb. there is danger of cutting the head. The ideal scrap weight is 30 per cent of the original weight. (*f*) Shoes containing chilled cast iron either integrally or in the form of inserts wear steel wheels more rapidly than those of gray cast iron only. The order of wear of the wheels with the shoes submitted in the total averages of mileage per 1/16 in. of wear is: "A," 100; "C," 88.23; "D," 83.23; "B," 80.32. (*g*) The wear of shoes is largely dependent on the character of the foundation brake rigging and the way in which it is maintained. (*h*) There is a wide range of mileage obtained with

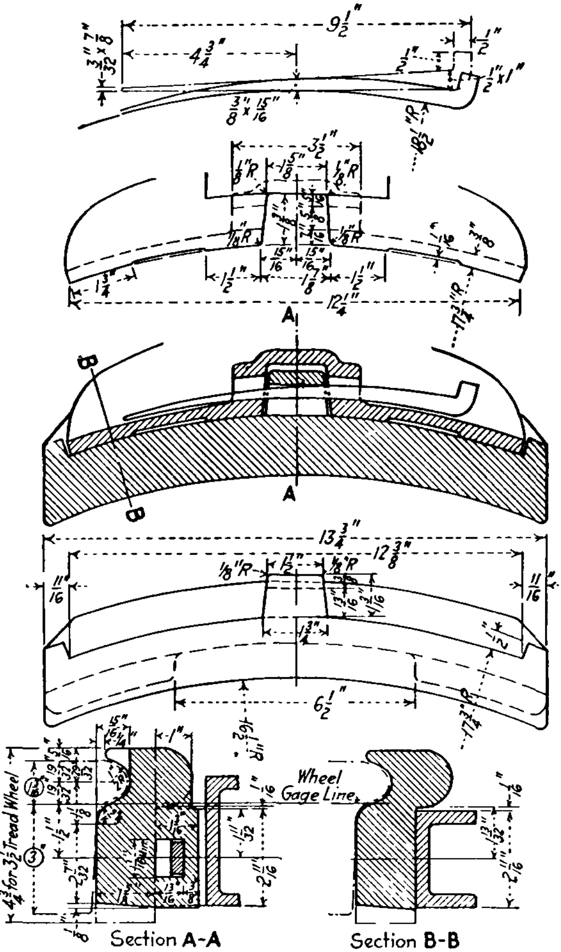


FIG. 27.—A.E.R.E.A. standard brake head, shoe and key for 3 in. and 3½ in. tread, wheels 33 in. to 36 in. diameter.

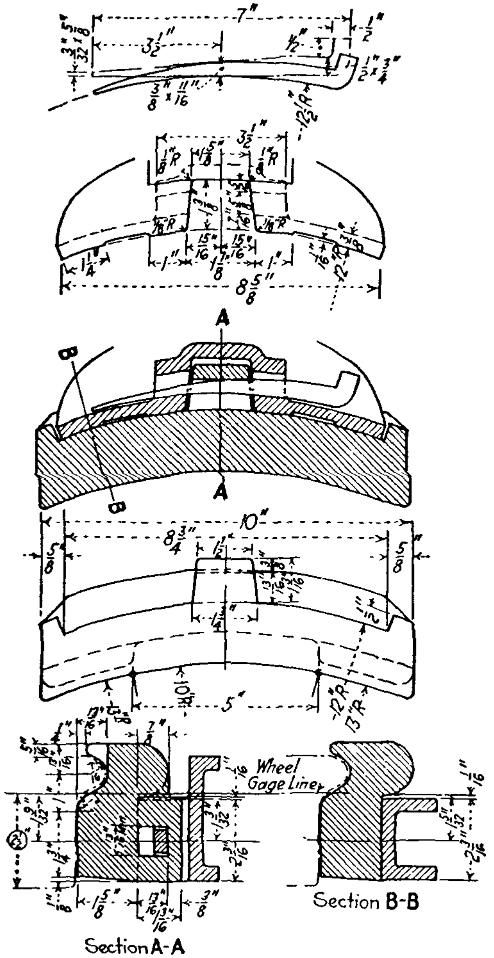
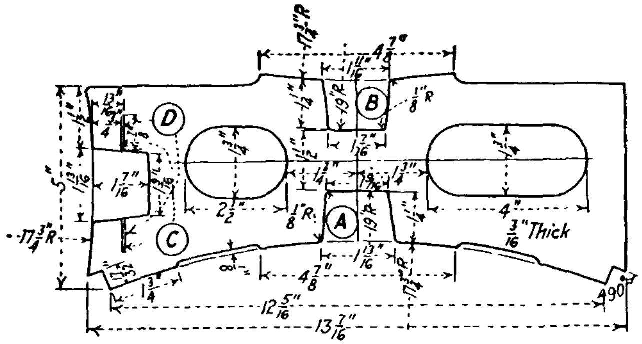


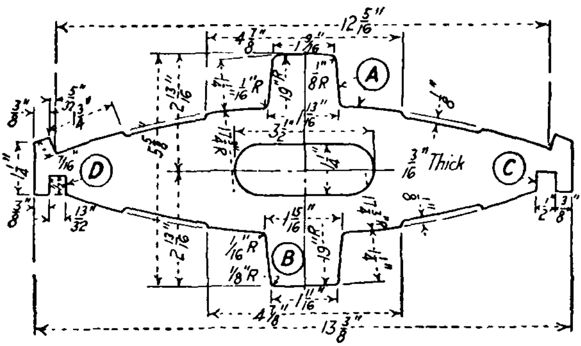
FIG. 28.—A.E.R.E.A. standard brake head, shoe and key for 2½ in. tread. wheels 21 in. to 26 in. diameter.



Brake Shoe Gage

Side of Gage Marked ***A*** *- Must Fit Back of Shoe*

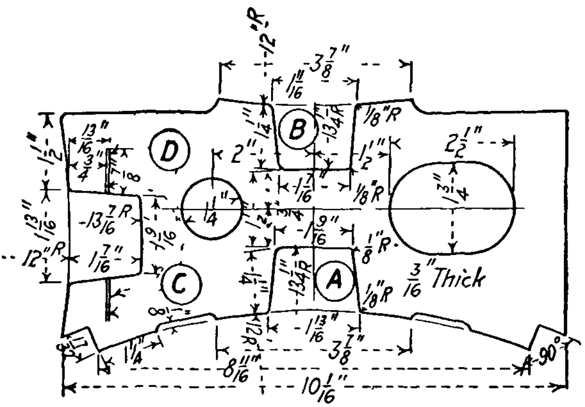
A *- Is Max. Width for Center Lug*
B *- Is Min. Width for Center Lug*
Line ***C*** *- Is Min Height of Slot in Center of Lug Measured from Back of Shoe*
Line ***D*** *- Is Max Height of Slot in Center of Lug Measured from Back of Shoe*



Brake Head Gage

A *- Head Must Admit Side of Gage to Full Depth and Must Fit Radius*
B *- Is Max Distance Between Lugs of Head*
C *- Is Max Thickness of Metal Between Face of Center Lugs and Key Slot*
D *- Is Min Thickness of Metal Between Face of Center Lugs and Key Slot*

FIG. 29.—Limit gages for brake heads and brake shoes for 3 in. tread wheels 33 in. diameter and over, and for $2\frac{1}{2}$ in. tread wheels 28 in. diameter and over (A.E.R.E.A. standard).



Brake Shoe Gage

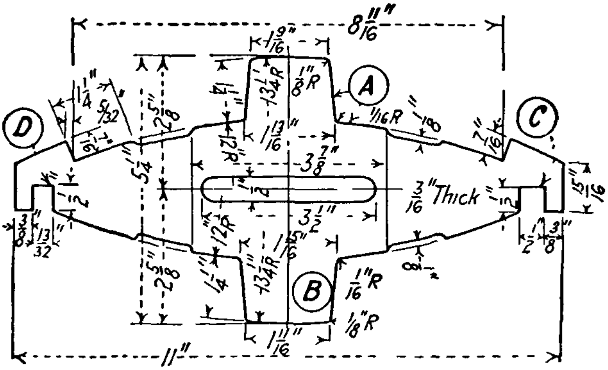
Side of Gage Marked A - Must Fit Back of Shoe

A - Is Max Width for Center Lug

B - Is Min Width for Center Lug

Line C - Is Min Height of Slot in Center of Lug Measured from Back of Shoe

Line D - Is Max Height of Slot in Center of Lug Measured from Back of Shoe



Brake Head Gage

A - Head Must Admit Side of Gage to Full Depth and Must Fit Radius

B - Is Max Distance Between Center Lugs of Head

C - Is Max Thickness of Metal Between Face of Center Lugs and Key Slot

D - Is Min Thickness of Metal Between Face of Center Lugs and Key Slot

FIG 30 —Limit gages for brake heads and brake shoes for $2\frac{1}{2}$ in and 3 in tread wheels 26 in diameter and under (A E R.E A standard).

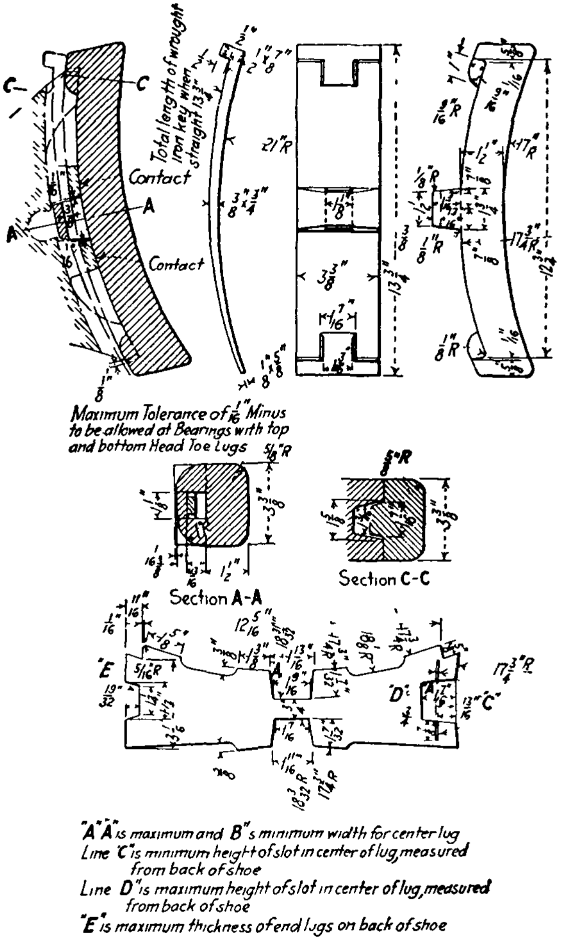


FIG 31 —A R A standard brake shoe key and gage individual shoes of the same type. (*i*) A steel wheel will run about 8000 miles per 1/16 in. wear in surface work. (*j*) The cost of brake shoes only per 1000 car miles is least for the cast iron "B" brake shoes with chilled ends. (*k*) The combination of brake shoe and wheel costs per 1000 car miles places the "A" and "B" brake shoes about on a par. (*l*) If the price is based on the cost of brake shoes per 1000 car miles, a higher price can be paid for the plain cast iron "A" brake shoe than for the one with chilled ends, because of the decreased cost of wheel wear. (*m*) Where steel wheels are used, both the wear of the wheel and the wear of the shoe should be considered, for though a shoe may be very economical in itself, it may have such a severe effect upon the wheel that when both wheel and shoe are taken together the result may not be economical.

A.E.R.E.A. Standard Brake Shoes, Brake Shoe Heads and Keys. Figures 27 and 28 show the principal dimensions of two of the A.E.R.E.A. standard brake shoes, brake shoe heads and keys, as revised in 1921. Standard designs for wheels of other dimensions are shown in the A.E.R.E.A. Manual. The thickness of the shoe is 1⅝ in. If a thicker shoe is desired, the increased thickness should be specified, but the other essential dimensions should not be changed.

A.E.R.E.A. Standard Limit Gages. Figs. 29 and 30 show the A.E.R.E.A. standard limit gages for brake shoe heads and brake shoes, as approved in 1921.

A.R.A. Standard Brake Shoe, Key, and Limit Gage. Fig. 31 shows the principal dimensions of the A.R.A. standard brake shoe, key and limit gage.

SECTION IX

CARS

Present Tendencies in Car Design. Of recent years there has been an increasing tendency to apply sound engineering principles to the structural design of electric railway cars and to give consideration to the great influence of car weight upon operating costs. With a proper consideration of the stresses involved, and an increased use of steel shapes and plates, the weights of car bodies have been reduced very materially, with no sacrifice of strength or life, and accompanied by considerably reduced maintenance costs.

The Birney "safety" car, a small single truck car of special design for one-man operation, was the first long step toward light weight. The first cars of this type, appearing in 1915, weighed approximately 10,000 lb., completely equipped with special trucks and motors. In response to a general feeling that the original construction of these cars was too light to afford proper life with reasonable maintenance costs, there was a reaction to the present generally used one-man safety car which weighs approximately 16,000 lb. Corresponding reductions have been made in the weight of double truck cars, both for city and interurban service. Both trucks and motors have also felt this tendency to lighter weight, as has been noted elsewhere. Figure 14, Section VII (page 393), for instance, illustrates an application of hollow axles and band brakes to trucks for light weight cars, while in Section IV may be noted the reduction in motor weights by the proper application of ventilation and other refinements of design.

There has been a further design tendency toward an elimination of the truck as such, by combining the spring, wheel and motor assembly with the car underframing, somewhat along the lines of an automobile chassis. In some cases the automobile design is followed still further in the adoption of the two-part axle, differential, and longitudinal shaft drive, and in general making full use of ball or roller bearings. These advanced designs are at present still in the design, or at best the experimental stage, and appear to be more attractive in Europe than in this country. (See *Electric Railway Journal*, Vol. 60, pages 825 and 828, and *General Electric Review*, Vol. 19, page 881.)

The great importance of time lost in stops, especially in city service, has brought attention to the matter of accelerating the movement of passengers in boarding and alighting, and has had considerable influence in the arrangement of entrances and exits in city cars.

One-man operation, where traffic conditions are such that it is feasible, affords an opportunity for great savings in labor costs per car mile. This also has occasioned numerous changes in details of car design.

Car Weights as Affecting Operating Costs. No general formulas can be given for the relations between car weights and the various items of operating expense Some expenses will vary almost directly with car weight, while others have no relation to it. Some expenses will be affected little unless the change in weight applies to all cars, while others will be affected proportionately by a partial change. Only a careful engineering study can solve the problem for any individual case; in general such a study will be along the following lines:

Maintenance of Way and Structures. Wear of rails and special work will vary almost directly with car weight, as will that of ties and ballast, but depreciation will not be affected. Track labor in proportion to the above Paving and other accounts, including distribution system and buildings, not usually affected

Maintenance of Equipment. There is no saving in these accounts purely on account of reduction in weight of cars, except possibly in wear on shop equipment, and in some cases minor savings due to reduced weight and cost of repair parts. As a change to lighter weight equipment most frequently includes a change to more economically maintained equipment, the savings due to the latter properly should be credited here.

Power If schedules remain the same, the amount of power in kilowatt hours will vary almost directly as the weight of cars (see Section III, pages 193 to 200). The variation in cost of power with respect to kilowatt hours, however, will depend upon conditions of power supply If power is manufactured, no saving will be effected beyond that in fuel and the cost of handling it, which, within the general limits of the change in output due to changing car weights, generally will be in proportion to the output. If power is purchased, the change in total cost due to change in kilowatt hours purchased may be estimated readily.

Conducting Transportation. There is no saving in these accounts purely on account of a reduction in car weight, except the possibility of minor savings in car service expenses, such as lubrication. If the change in car weight is accompanied by a change from two-man to one-man operation, there may be a large saving in wages of conductors and motormen, depending upon the differential in wages between two- and one-man operators.

General and Miscellaneous Expense. The cost of accidents is the only one of this group of expenses that may be affected in a change in car weights. The increased efficiency of modern safety devices which usually are included in new car equipment should reduce the number and cost of accidents. Such a reduction, if it is effected, is not due purely to the change in car weights, and estimates of such savings should be made with great care, if included at all

Fixed Charges With a complete change to light weight cars, longer tie spacing, lighter rail and lighter general track construction may be used, with correspondingly less first cost. Whether or not credit may be taken for a consequent reduction in fixed charges will depend upon the circumstances of the individual case. In a comparison of costs accompanying two plans for original construction, one with light, the other with heavy cars, full credit would be given this item. Similar consideration applies to distribution system copper and capacities of power and substations, as all these are dependent upon power peaks, which are directly affected by car weights.

Decreased Weight of Single-end Cars. The A.E.R.E.A. 1910 Committee on Equipment called attention to the large saving in weight which can be made by the use of single-end cars, as follows, as applying to double-truck car of the semi-convertible type.

Weight eliminated by making car single-end:

	Lb.
Step risers, hangers and brackets	306
Reversing mechanism for seats	108
Brake rigging	332
Fender and hangers	125
Doors, door pockets and operating mechanism	548
Sand box and mechanism	40
Brake valve	21
Air-brake piping	100
Trolley equipment	140
Electric equipment, wiring, etc.	292
Headlight	35
Switch cabinet	75
Life guards	130
Snow scrapers	225
Total saving in weight	2477

While the equipment advantages of single-end cars are admitted, many operators hesitate to adopt a design which cannot readily be turned back in case of blockades, which calls for either Y's or loops at the end of routes, and which is harder to handle quickly in case of car house fires.

Light Weight Single Truck Cars. This design is commonly known as the Birney "safety" car, as it was first developed by C. O. Birney of the Stone & Webster organization. While minor changes in design have been made by some roads, the car is of the general design as shown by Fig. 1. It seats 32 passengers when arranged for double-end operation, this capacity being increased to 35 by utilizing the controller and brake space when equipped only for single-end operation. The car is for one-man operation and is equipped with safety features which generally provide (1) that the brakes cannot be released or the motor current applied until doors are closed, (2) the doors cannot be opened until the controller is at the off position and the brakes applied, and (3) the controller is arranged to require a conscious effort on the part of the operator to operate it. Doors are operated and rails are sanded pneumatically, the air valve having operating positions so that one handle operates brakes, sand and doors. When the air valve handle is thrown to the emergency stop position, power is cut off, sand is applied, and doors are opened, automatically. (See p. 476.)

The small capacity of the car avoids the necessity for a large operating platform, and leaving passengers usually alight before

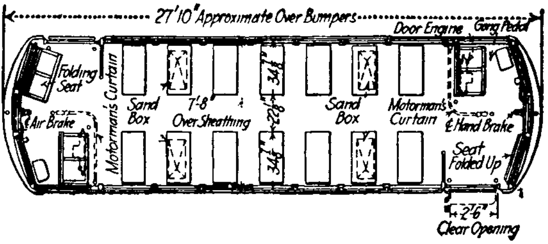


FIG 1 —Typical single truck light weight car

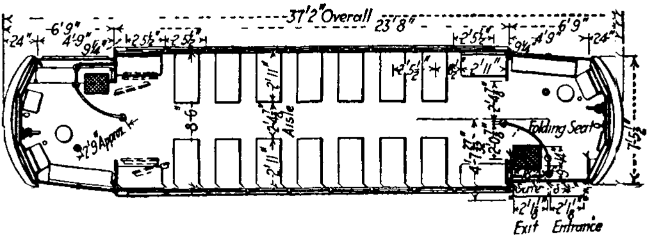


FIG. 2 —Double truck light weight car, with automatic exit gate for one-man operation, Chicago Surface Lines.

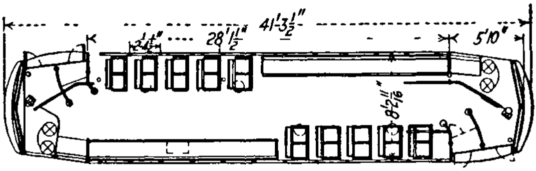


FIG 3 —Double truck light weight car Eastern Massachusetts St. Ry. city and suburban service

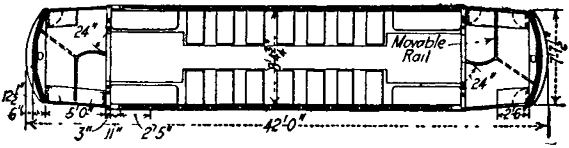


FIG. 4 —Ft Wayne city car.

others are allowed to board; this when boarding and alighting are through one door as shown by Fig. 1, the usual design. The managers of some roads, however, have felt that the single door caused undue delays at passenger stops, and have provided a separate exit door alongside the one shown in Fig 1, in such cases, the two doors usually are operated by separate controls so that the operator can handle them singly In Chicago, the exit door has been arranged to operate automatically, being opened by the leaving passenger's weight on a floor plate just inside the door, provided a full brake application has been made and the operating handle is in the door-opening position, the door closes automatically when the passenger has cleared the step, preventing boarding passengers entering via the exit door (see Fig 2).

The single-truck double-end safety car as built in Chicago weighs as follows:

	Lb	Lb.
Car body		5,367
Steel frame	1,930	
Wood framing, floors, roof, lining and finish	2,320	
Doors and operating mechanism	1,117	
Car body apparatus and fittings		1,662
Seats	709	
Auxiliary electrical apparatus, wiring and conduit	246	
Railings, signs, curtains ventilators, registers, draw-bars, etc	707	
Motor equipment		2,802
Motors, pinions and gear cases	1,915	
Motor wiring and conduit	105	
Controllers and resistance	477	
Trolley, arrester, fuse, choke coils, etc	305	
Brake equipment		1,583
Compressor, reservoirs, governor, etc	665	
Piping, valves and fittings	574	
Brake cylinder and foundation rigging	191	
Hand brake and rigging	94	
Sand boxes and traps	59	
Trucks		4,693
Framing springs, etc	2,780	
Axles wheels and gears	1,913	
Miscellaneous		268
Paint and varnish	110	
Miscellaneous bolts, screws, etc	158	
Complete car		16 375

The single-truck safety car may be used wherever a small car unit is desirable, and as compared with the former generally used city car, makes possible large reductions in costs per car mile, due to one-man operation, as well as on account of small weight, as outlined on page 496. The field of application comprises the following general classes of service· (1) lightly traveled lines on which the substitution is made on a car for car basis, (2) lightly traveled lines where materially increased service is furnished, (3) heavy service lines radiating from rapid transit or main truck surface lines; (4) lines paralleling rapid transit lines; and (5) main surface lines.

It usually is undesirable to substitute safety cars on a car for car basis especially where large double-truck cars were previously operated. There are, however, many lines now operating with two-man cars where a substitution of safety cars for an equal number of

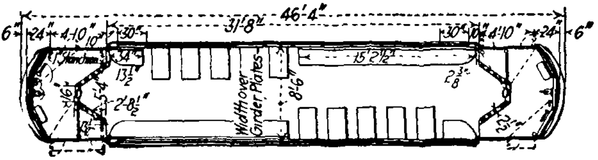


FIG. 5.—East St. Louis city car.

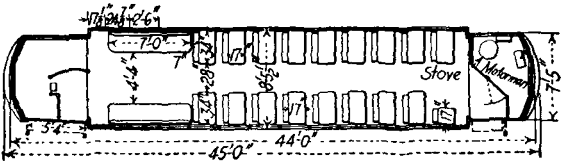


FIG. 6.—Omaha city car.

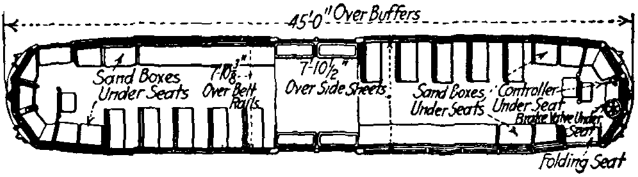


FIG. 7.—Pittsburgh Railways Co. city car.

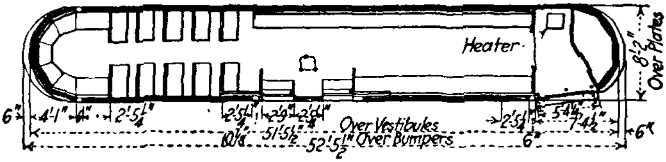


FIG. 8.—Cleveland city car; front-entrance, center-exit type.

the old cars is the only proper course. Many cases can be cited where railways began safety car operation by giving shorter headways on one or more of the busiest lines, and at the same time or shortly thereafter successfully substituted safety cars for old single-truck cars on a car for car basis on less important lines.

One of the most gratifying features of safety car operation is the increased revenue which often has resulted from short headways on lines previously regarded as unprofitable. The possibility of increase in receipts resulting from more frequent service always has been recognized. but the desirability of handling the greatest num-

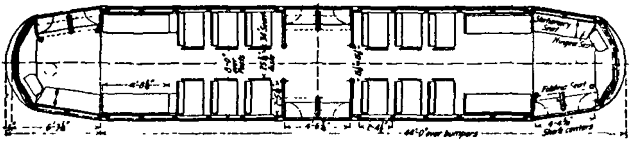


Fig. 9.—Brooklyn city car, double-end front-entrance center-exit type.

ber of passengers per platform man has heretofore required the use of large units running at infrequent intervals, because when shorter headways are operated with large cars, the increased revenue is counterbalanced by the increased operating expense. When the safety car was first used to replace large double truck cars, it was necessary in almost every instance to operate at least 50 per cent more mileage in order to handle the traffic properly. While some increase in car riding generally was expected, the decrease in opera-

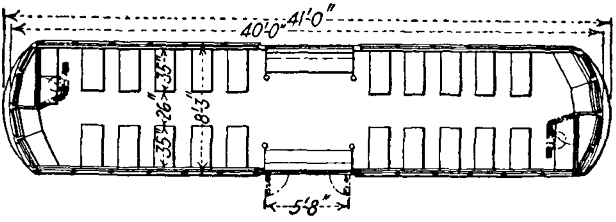


Fig. 10.—Charleston city car for multiple unit operation.

ting expense was the chief reason for the adoption of safety cars. Actual results of operation in many cases showed that the increase in revenue was greater than the saving in operating expense.

The application of safety cars to lines radiating from rapid transit lines or main trunk surface lines has been made in several of the larger cities. Here the morning loads are picked up gradually and discharged at one point, and in the evening the majority of passengers board the car at one point and leave in small groups. The collection of fares can be simplified under these conditions by having prepayment and post-payment areas at the rapid transit terminals;

all passengers boarding the car at other points can pay as they leave on inbound trips and as they enter on outbound trips.

In the application of safety cars to the main surface lines of large cities the methods vary in different sections of the country, and there is an even greater variation in opinions regarding their practicability in this service. The most common use of safety cars on main surface lines is where the car mileage is increased from 50 to 100 per cent. The rapid acceleration and retardation of the safety car, the smaller number of passengers per car and hence fewer stops, the low floor design, the elimination of signals between conductor and motorman, and the better view traffic officers have of the door, combine to speed up the movement of individual cars; on the other hand the safety provisions of the door and step control and the collection of fares by the car operator may tend to increase the duration of stops. As previously mentioned, the latter tendency is counteracted in some cases by the use of street collectors and by prepayment areas. The congestion arising from the greater number of units also tends to slow up traffic. This is not proportionate to the number of units because of the features which tend to accelerate the movement of the safety car, and because of their smaller size. It is certain that a number of safety cars can pass over a congested section of city street more rapidly than an equal number of large double-truck cars, and still more readily than an equal number of two-car trains. Observations made of congested city service when safety cars and double-truck cars operate on the same track show that the safety cars are frequently waiting for a double-truck car to get out of the way, and that in the neighborhood of 50 per cent more safety cars than double-truck cars can be operated over a given section of track in a given time. In general, a headway of 30 to 40 seconds for distances up to one mile on surface lines is about the limiting frequency even for safety car operation. Bearing in mind that no material increase in riding can be anticipated by decreasing headways under three or four minutes, the use of safety cars for the entire day generally does not seem advisable on the heaviest lines even where their operation is physically possible. On many heavy surface lines, safety cars are used in combination with double-truck cars, short-routing the former during rush hours.

Light Weight Double-truck Cars. The successful operation of the Birney single-truck one-man car directed attention to the larger double-truck car with respect to reductions in weight and possibilities of one-man operation. The table on page 508 shows several instances where double-truck cars seating about 48 passengers have been brought down to a total weight of approximately 30,000 lb., or about 625 lb. per seat. These cars often are equipped with all of the control and safety devices for one-man operation and are so operated. When built with proper door arrangement, they frequently are operated one-man in the normal hours and two-man in the rush hours.

Figs. 2 and 3 show the arrangement of light weight double-truck one-man cars of the Chicago and Eastern Massachusetts companies, respectively. Weights of the Chicago car are as follows:

	Lb.	Lb.
Car body		9,361
Steel frame	4,600	
Wood framing, floors, roof, lining and finish	3,309	
Doors and operating mechanism	1,452	
Car body apparatus and fittings		2,202
Seats	900	
Auxiliary electrical apparatus, wiring and conduit	346	
Railings, signs, curtains, ventilators, registers, draw-bars, etc	956	
Motor equipment		5,195
Motors, pinions and gear cases	3,864	
Motor wiring and conduit	175	
Controllers and resistance	892	
Trolley, arrester, fuse, choke coils, etc	264	
Brake equipment		1,831
Compressor, reservoirs, governor, etc	953	
Piping, valves and fittings	324	
Brake cylinder and foundation rigging	356	
Hand brake and rigging	139	
Sand boxes and traps	59	
Trucks		8,956
Framing, bolsters, transoms, springs, etc	4,658	
Axles, wheels and gears	4,148	
Truck guards	150	
Miscellaneous		505
Paint and varnish	225	
Miscellaneous bolts, screws, etc	280	
Complete car		28,050

Arrangement of Seats. The transverse or cross seat is more comfortable than the longitudinal seat, especially with rapid rates of acceleration and braking. Approximately the same seating capacity may be had with either type, but the longitudinal seat leaves more room for standing passengers. Where large standing loads are to be handled in rush hour service, the entire seating capacity often is in longitudinal seats, as in Fig. 12. In interurban cars where the rides are long, all seats usually are transverse, as shown in Figs. 15, 16, 17 and 18. The usual arrangement of the single-truck safety car also is with all cross seats, as in Fig. 1. The general practice in city and suburban service is to provide part cross and part longitudinal seats, thus partly catering to the passenger's comfort, and partly taking advantage of the larger car capacity afforded by the longitudinal seat. When both types are used, the cross seats generally are grouped at the ends of the car, as in Figs. 2, 4, 6, 13 and 14, thus providing space for movement of boarding and alighting passengers when the car is full. In some cases an arrangement is used wherein the cross seats are grouped on one side of the car, with longitudinal seats opposite, as in Figs. 3, 5 and 7.

Seat Space per Passenger. In determining the seating capacity of cars with longitudinal seats, it is customary to allow 17 to 18 in. per passenger. The Equipment Committee of the Am. El. Ry. Eng. Assn. (1911) recommended that 17 in. be used in determining the capacity of longitudinal seats; 17 in. is used by the Massachusetts Dept. of Public Utilities and by the District of Columbia Public Utilities Commission. The standard of the Public Service Commission, First District, New York, is 17.78 in., this being the average of a number of observations by Commission inspectors. Kidder's

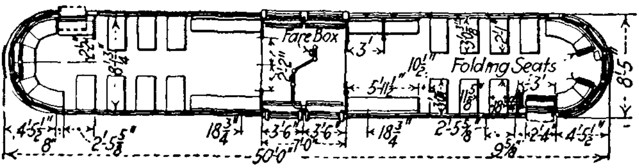


FIG. 11.—Milwaukee city car for multiple unit operation.

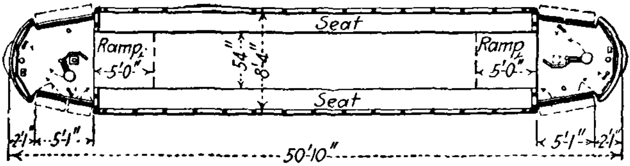


FIG. 12.—New Jersey city and suburban car.

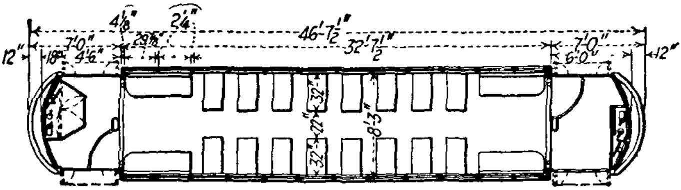


FIG. 13.—Richmond city and suburban car.

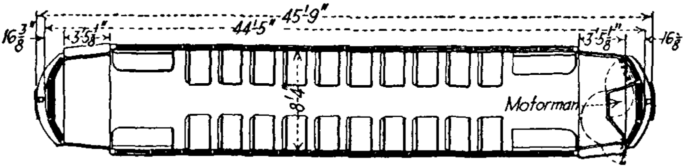


FIG. 14.—Connecticut city and suburban car.

Architect's and Builder's Pocket Book gives 18 in. as the length of church pew considered as a "sitting."

Passenger Capacity. Seating capacity is determined by the number of transverse seats, plus the capacity of longitudinal seats figured as above.

Standing capacity is customarily determined by dividing the floor area, measured in square feet, by 2, 3 or 4. In a longitudinal seat car, 6 to 8 in. is deducted in front of the seats to allow for knee room, and the floor area devoted to fare boxes, control apparatus, etc., is also deducted. Standards adopted by the U. S. army are based on the experience that an area of 3.35 square feet per person will permit free maneuvering. A report of Ford, Bacon and Davis to the Pennsylvania State Commission in the matter of service on the lines of the Philadelphia Rapid Transit Company, states that 4 square feet per standing passenger provides sufficient room for comfort and for free movement through the car, and that the total seated and standing capacity of the interior of the car will equal about 3.5 square feet per total seated and standing passenger. In a report of B. J. Arnold on transportation facilities in the city of San Francisco, it was recommended that reasonable standards to be applied to all types of cars are as follows: (1) comfortable standing, 50 per cent in excess of cross seats, and 100 per cent in excess of longitudinal seats, plus platforms; (2) normal maximum capacity 3 square feet per standing passenger; (3) emergency maximum capacity, 2 square feet per standing passenger.

In connection with the determination of car capacity, it should be noted that there are a certain number of passengers who will stand by preference, irrespective of whether or not seats are available. The number of such preferential standers varies with the conditions existing in various localities. For instance, investigation has shown that 21 per cent of any load will stand by preference in Madison, Wisconsin; in La Crosse, 15.5 per cent; in Milwaukee, 19 per cent; in Lincoln, Nebraska, 14 per cent; in Cincinnati, 15.5 per cent; and in Springfield, Mass., about 10 per cent.

Drop Platform Type. The greater number of city cars in use today are of this type. Platform sills are carried by cantilever suspension from the main sills and extend out to carry the end platforms and vestibules. The floor of the end platform is thus several inches below the floor of the car. In practically all cases the platforms are enclosed by a vestibule with folding doors over the entrance and exit steps. In later types of both city and suburban cars the practice is to omit the bulkhead at the end of the car body proper to facilitate passenger interchange. In two-man operation, such cars may be operated with the entrance at the rear and exit at the front, with prepayment or pay-as-you-enter operation, or front-entrance, rear-exit, with pay-leave operation. Typical cars of this type are shown in Figs. 4, 5, 6, 12, 13 and 14.

Center Entrance and Exit Type. In this type, as illustrated by Figs. 10 and 15, the combined entrance and exit doors are located in the middle of the car. This type was advocated for city use principally because by means of ramped floors or a depressed well at the middle of the car, the number and height of steps to the street

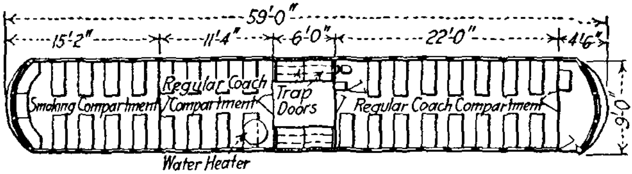


FIG. 15.—Kansas City interburban car.

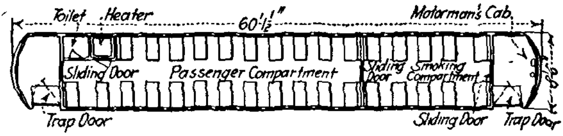


FIG. 16.—Lake Shore El. Ry. Co. (Ohio) interurban car.

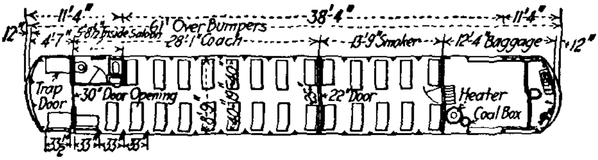


FIG. 17.—Union Traction Co. of Ind. interurban car.

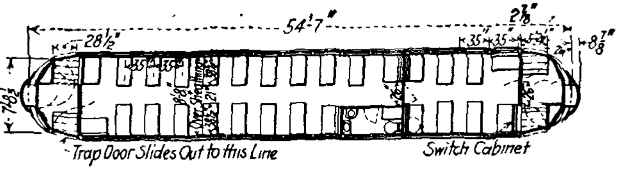


FIG. 18.—Chicago, North Shore & Milwaukee interurban car.

surface could be reduced very materially as compared with the drop platform type. Experience with this type is congested city operation, however, demonstrated that there was likely to be conflict between boarding and alighting passengers, with a resulting loss of passenger interchange time. Its use at present, consequently, is practically confined to a few cases in interurban service where passenger interchange time is not so important, to motor cars which are frequently used in multiple unit service, and to trail cars. In the two latter named cases, one conductor per car can handle passengers most rapidly with the entrance and exit combined or located side by side in the middle of the car. In the Toronto trailers, there are three doors at the middle of the car. with control and railings so arranged that two may be used for entrance and one for exit when the heavier movement of passengers is boarding, and the reverse when the heavier movement of passengers is alighting.

Front Entrance, Center Exit Type. This type of car is also known as the "pay-as-you-pass" type or the "Peter Witt" car, as it was originated in Cleveland during the time that Mr. Witt was City Commissioner of Street Railways. For single-end operation the car is illustrated by Fig. 8. Entrance is at the front end where the doors are controlled by the motorman, and exit is in the middle of the car through doors operated by the conductor who with his fare box is stationed at that point. Passengers board the car with no delay incident to fare collection up to the point where the entire front half of the car is filled. Fares are paid as passengers pass the conductor's position. This may be done immediately on boarding the car, if it is then convenient, or at any time during the trip while the car is in motion, in either of which cases the passenger takes his seat somewhere in the rear half of the car. Passengers who remain in the front half of the car pay fares as they pass the conductor on their way to the exit door. When cars are operated single-end exclusively, the arrangement of seats is usually as shown in Fig. 8, with longitudinal seats in the front half of the car and transverse seats in the rear half. This accomplishes the purpose of providing a large standing area in the front half of the car, so that fare collection need not lengthen the stop time where a large number of passengers are boarding, as might occur with pay-enter fare collection. As the transverse seats in the rear of the car are the more desirable, this arrangement also tends to induce passengers to pay fares and occupy positions in the rear of the car so long as space is there available, and this reduces the stop time for alighting passengers as compared with pay-leave operation.

When necessary to operate cars double-end, it is not possible to concentrate the longitudinal and transverse seats as above noted, but the arrangement is usually somewhat as shown by Fig. 9. The relative arrangement of entrance, exit, and conductor's position, used with this type of car, very materially reduces the time required at passenger stops. This is shown very definitely by Fig. 5, Section III, page 124, the data for which was obtained through a large number of observations in several cities.

CHARACTERISTICS OF REPRESENTATIVE CARS

Fig. no.	City or company	Service	Single or double end	Entrance and exit	Operation	Overall length		Overall width		Seating capacity	Total weight excl. pass., lb.	Weight excl. trucks and motors, lb.	No. motors	H.p. per motor	H.p. per ton	Lb. per seat	Diam. wheels, in.
						Ft.	In.	Ft.	In.								
1	Connecticut	City*	D	E	1	27	10	7	8	32	16,500	10,430	2	25	6.1	516	26
	Dayton	City	S	E	1	42	6	8	2	51	25,900	14,200	4	25	7.7	508	26
	Chicago	City	D	C	Tr	47	6	8	5	62	26,000	17,950	0	....	0	419	22
	Toronto	City	S	C	Tr	49	2	8	4	60	26,300	....	0	....	0	438	22
	Houston	City	D	E	1	40	1	8	5	48	28,000	15,460	4	25	7.1	583	26
2	Chicago	City	D	E	1	37	2	8	6	45	28,050	15,230	4	25	7.1	624	26
	Connecticut	City	D	E	1, 2	44	0	8	4½	44	30,000	15,800	4	25	6.7	682	26
	Boston	City	D	E	1, 2	45	0	8	6	48	30,860	18,360	4	25	6.5	643	26
3	E. Mass	Sub.	D	E	1	41	3½	8	4½	48	31,880	18,300	4	25	6.3	664	26
7	Pittsburgh	City	D	C, E	2	45	0	7	10½	59	33,200	17,920	4	40	9.6	563	24
	Memphis	City	D	E	1, 2	44	10	8	3	48	33,400	20,900	4	25	6.0	696	26
	Birmingham	City	S	C, E	2	49	5	8	5	62	34,000	18,884	4	35	8.2	548	26
6	Omaha	City	S	E	2	45	0	8	5½	45	34,300	20,280	4	25	5.8	762	30
4	Fort Wayne	City	D	E	1, 2	42	0	8	5	48	34,440	20,840	4	25	5.8	718	30
5	E. St. Louis	City	D	E	2	46	4	8	6	48	34,900	20,220	2	50	5.7	727	33, 21
	Philadelphia	City	S	C, E	1, 2	45	6	8	1½	54	35,820	19,940	2	60	6.7	663	28, 22
	Oakland	City	D	E	1, 2	44	10	9	1½	48	36,000	19,700	4	40	8.9	750	24
10	Charleston	City	D	C	Tn	41	0	8	3	56	37,180	22,000	4	40	8.6	664	24
	Brooklyn	City	D	C, E	1, 2	44	2	8	1½	49	37,500	20,860	4	35	7.5	765	26
	San Diego	City	D	E	1, 2	47	5	8	6½	56	38,000	22,000	4	40	8.4	670	26

CHARACTERISTICS OF REPRESENTATIVE CARS.—(*Continued*)

Fig. no.	City or company	Service	Single or double end	Entrance and exit	Operation	Overall length Ft.	Overall length In.	Overall width Ft.	Overall width In.	Seating capacity	Total weight excel. pass., lb.	Weight excel. trucks and motors, lb.	No. motors	H.p. per motor	H.p. per ton	Lb. per seat	Diam. wheels, in.
13	Richmond	Sub.	D	E	Tn	46	7½	8	3	48	41,580	22,400	4	40	7.7	866	30
12	New Jersey	Sub.	D	E	2	50	10	8	4	50	42,000	22,420	4	40	7.6	840	30
11	Milwaukee	City	D	C, E	Tn	50	0	8	5	56	43,000	25,280	2	65	6.0	768	34, 22
8	Cleveland	City	S	C, E	2	52	5½	8	2	59	44,360	23,750	4	50	9.0	752	26
	Boston	City	D	C	Tn	48	0½	8	5¼	56	44,800	26,520	4	40	7.1	800	24
14	Connecticut	Sub.	S	E	2	45	9	8	4	52	46,000	21,000	4	50	8.7	885	33
	Baltimore	Sub.	D	E	Tn	46	0	8	2	55	46,660	25,100	4	50	8.6	848	33
	Chicago	City	D	E	2	48	11	8	6	48	46,900	22,760	4	50	8.5	977	28
	Chicago Elevated	Elev.	D	E	Tn	48	0	8	10½	52	75,000	42,000	2	170	9.1	1,442	30
	Ind. & Cincinnati	Int.	S	E	2	60	0	8	11	54	75,000	38,000	4	100	10.7	1,389	37
15	Kansas City	Int.	D	C	2	59	0	9	0	68	78,150	42,300	4	115	11.8	1,149	36
16	Lake Shore	Int.	S	E	2	60	1½	8	6	64	84,500	45,300	4	140	13.2	1,320	38
17	U.T.C. of I.	Int.	S	E	2	61	0	8	9	65	85,600	45,200	4	125	11.7	1,317	38
18	Ch. N.S. & M.	Int.	D	E	Tn	54	7	8	8	58	90,000	49,000	4	140	12.4	1,552	36

* = Single truck, Birney type
C = Center
E = End

S = Single
D = Double
1 = One-man

2 = Two-man
Tn = Train
Tr = Trailer

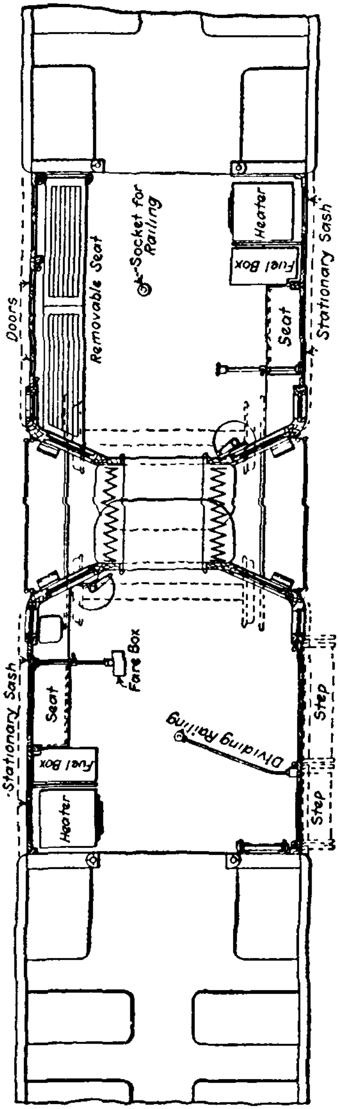


FIG. 19.—Milwaukee three-truck two-car train unit.

Articulated Cars. The Milwaukee Electric Railway & Light Company has designed and put into operation a three-truck, two-man train unit, in which two separate car bodies are carried by three trucks, two of which are located in the usual positions at the outer ends of the cars, while the third is located between the two cars and carries the weight of one end of each car. The arrangement of the ends of the two cars where joined together, with the vestibuling arrangement so as to provide for an interchange of passengers between the cars, and the location of the middle truck, are shown in Fig. 19. As constructed in Milwaukee, old cars being used, it was necessary to support the old car bodies in substantially the same manner as they had been previously carried upon individual trucks, namely, at the car body bolsters. This was accomplished by placing under each car an auxiliary steel underframe beginning at the outer truck bolster and continuing through to the point of joining the two car bodies together. By suitable design, this steel underframe carries the weight of the car at the old car body bolster, provides adequate support for a greatly enlarged platform, and transmits the load to the middle truck, where a kind of ball and socket joint arrangement on the ends of the two main auxiliary girders permits both to rest and pivot on the center-truck kingpin and bolster. Enlarged platforms were desired in order to provide space for the larger number of passengers to be handled and to accommodate the heating equipment. The platforms were connected by overlapping steel plates attached to the platforms, and arranged to permit the necessary joint movement on curves. The vestibules were connected by diaphragms. The motors of the cars were grouped on the outer trucks. There were recovered from each pair of old cars two trolleys, two sets of air and electric control, one air compressor and two standard trucks, besides various minor equipment, such as destination signs and headlights. Notwithstanding the addition of the third truck, new steel underframes and enlarged platforms, the total weight of the finished train is only slightly more than the combined weight of the individual cars. On account of the exposure of the middle truck on curves, it was found desirable to reduce the lateral dimensions of the truck and necessary clearance to a minimum, which was accomplished by building a special truck with inside journal boxes. A similar plan proposed for use in Detroit contemplates the use of three car body units mounted on four sets of trucks. In this case each of the car body units is made center-entrance and exit, so that the middle trucks may be of standard design, and all four trucks may carry motors. Experience in Milwaukee has shown the three-truck, two-car units to have advantages of (1) utilization of old equipment so as to make it entirely acceptable for rush hour and special service, as well as for regular service on heavy traffic lines; (2) economy in platform labor, as a motorman and one conductor can handle the load of two full-sized double truck cars; and (3) greatly reduced overhang of coupled ends of cars on track curves. The disadvantages include (1) slowness in passenger interchange as compared with single car units, and (2) some difficulty in locating the trolley stand for proper operation and still accessible to the conductor.

Double Deck Cars have been quite popular for city service in Great Britain, but in this country the use of this type has been confined to a few trial installations. Under congested traffic conditions the double deck car occupies only about half the street space per passenger, and it is capable of demonstration that its overall cost of operation per seat mile is less than that of any combination of motors or motors and trailers of similar capacity. Nevertheless its operation in American cities has been without the marked success which such figures might indicate, due to the apparent unwillingness of the street car rider to climb to the upper deck. Such results are difficult to understand in view of the popularity of the upper deck seats in motor bus service in New York and Chicago.

Rapid Transit Cars. In the design of a car for elevated and subway service the conditions to be met are considerably different from those encountered by a car on a street railway, both as to the limitations of size and the difficulties of traffic to be handled. The design also will differ from that of the rolling stock of a steam or electrified suburban line, in which service a higher ratio of seat-

TABLE OF COMPARATIVE DATA FOR VARIOUS ELEVATED AND SUBWAY CARS

Road and class of service	Number of cars in train			Total length over couplers (Note 1)		Total capacity				Horse-power (Note 2)						
						Seated				Per ton		Per passenger seat				
	Motor	Trailer	Total	Ft.	In.	Min. (rush hour)	Med. (med. traffic)	Max. (light traffic)	Max. (seated and standing)	Light	Loaded max. capacity (seated and standing)	Rush hour service (min. seating)	Med. traffic (med. seating)	Light traffic (max. seating)	Per max. pass. capacity (seated and standing)	Total per train
Interborough Rapid Transit Subway—steel cars.	1	0	1	52	0¾	44	46	48	169	10.27	7.88	9.09	8.70	8.33	2.37	400
	.	1	1	52	0¾	44	46	48	169	.....	......	......	......	......	......	....
	2	0	2	104	1½	88	92	96	338	10.27	7.88	9.09	8.70	8.33	2.37	800
	2	1	3	156	2¼	132	138	144	507	7.73	5.75	6.06	5.80	5.55	1.58	800
	3	2	5	260	3¾	220	230	240	845	7.13	5.28	5.45	5.22	5.00	1.42	1200
	4	2	6	312	4½	264	276	288	1014	7.73	5.75	6.06	5.80	5.55	1.58	1600
	5	3	8	416	6	352	368	384	1352	7.35	5.46	5.68	5.43	5.21	1.48	2000
	7	3	10	520	7½	440	460	480	1690	8.02	5.98	6.36	6.08	5.83	1.66	2800
New York Municipal Railway Corporation—Subway.	1	0	1	67	3	78	......	90	270							280
	2	0	2	134	6	156	......	180	540							560
	3	0	3	201	9	234	......	270	810							840
	4	0	4	269	0	312	......	360	1080							1120
	5	0	5	336	3	390	......	450	1350	6.6	4.5	3.3	......	3.1	1.04	1400
	6	0	6	403	6	468	......	540	1620							1680
	7	0	7	470	9	546	......	630	1890							1960
	8	0	8	538	0	624	......	720	2160							2240
Brooklyn Rapid Transit—Elevated, series motor car and standard trailer.	1	0	1	49	3		50		154	11.15	8.58		8.0		2.59	400
	.	1	1	48	0		48		148	.....	......		......		......	....
	2	0	2	98	6		100		308	11.15	8.58		8.0		2.59	800
	2	1	3	146	6		148		456	9.01	6.62		5.41		1.75	800
	3	1	4	195	9		198		610	9.62	7.16		6.07		1.97	1200
	3	2	5	243	9		246		758	8.46	6.15		4.88		1.58	1200
	4	2	6	293	0		296		912	9.0	6.62		5.41		1.75	1600

For notes, see p. 514.

TABLE OF COMPARATIVE DATA FOR VARIOUS ELEVATED AND SUBWAY CARS.—(*Continued*)

Road and class of service	Number of cars in train: Motor	Trailer	Total	Total length over couplers (Note 1): Ft.	In.	Total capacity: Seated: Min. (rush hour)	Med. (med. traffic)	Max. (light traffic)	Max. (seated and standing)	Horse-power (Note 2): Per ton: Light	Loaded max. capacity (seated and standing)	Per passenger seat: Rush hour service (min. seating)	Med. traffic (med. seating)	Light traffic (max. seating)	Per max. pass. capacity (seated and standing)	Total per train
Boston—Cambridge subway.	1	0	1	69	6½		72		266	9.3	6.50		5.55		1.50	400
	2	0	2	139	1		144		532							800
	3	0	3	208	7½		216		798							1200
	4	0	4	278	2		288		1064							1600
	5	0	5	347	8½		360		1330							2000
	6	0	6	417	3		432		1596							2400
	7	0	7	486	9½		504		1862							2800
	8	0	8	556	4		576		2128							3200
Hudson & Manhattan—Original type. Tunnel service.	1	0	1	48	4		44		158	8.64	6.65		7.27		2.02	320
	2	0	2	96	8		88		316							640
	3	0	3	145	0		132		474							960
	4	0	4	193	4		176		632							1280
	5	0	5	241	8		220		790							1600
	6	0	6	290	0		264		948							1920
	7	0	7	338	4		308		1106							2240
				386	4		352		1264							2560
Hudson & Manhattan—Pennsylvania Newark service.	1	0	1	48	4		44		158	12.33	9.46		10.2		2.85	450
	2	0	2	96	8		88		316							900
	3	0	3	145	0		132		474							1350
	4	0	4	193	4		176		632							1800
	5	0	5	241	8		220		790							2250
	6	0	6	290	0		264		948							2700
	7	0	7	338	4		308		1106							3150
	8	0	8	386	8		352		1264							3600

For notes, see p. 514.

TABLE OF COMPARATIVE DATA FOR VARIOUS ELEVATED AND SUBWAY CARS.—(*Continued*)

Road and class of service	Number of cars in train: Motor	Trailer	Total	Total length over couplers (Note 1): Ft.	In.	Total capacity: Seated: Min. (rush hour)	Med. (med. traffic)	Max. (light traffic)	Max. (seated and standing)	Horse-power (Note 2): Per ton: Light	Loaded max. capacity (seated and standing)	Per passenger seat: Rush hour service (min. seating)	Med. traffic (med. seating)	Light traffic (max. seating)	Per max. pass. capacity (seated and standing)	Total per train
Long Island Railroad Subway—Suburban Flatbush avenue division.	1	0	1	51	4		52		154	10.05	7.91		7.7		2.59	400
	.	1	1	46	6		56		136	.....	......		....		.....	.....
	2	0	2	102	8		104		308	10.05	7.91		7.7		2.59	800
	2	1	3	149	2		160		444	8.04	6.12		5.0		1.80	800
	3	1	4	200	6		212		598	8.62	6.64		5.66		2.01	1200
	3	2	5	247	0		268		734	7.54	5.70		4.48		1.63	1200
Long Island Railroad Subway—Suburban Pennsylvania station.	1	0	1	64	5¼		71		186							470
	2	0	2	128	10½		142		372							940
	3	0	3	193	3¾		213		558							1410
	4	0	4	257	9		284		744							1880
	5	0	5	322	2¼		355		930	9.0	7.21		6.62		2.52	2350
	6	0	6	386	7½		426		1116							2820
	7	0	7	451	0¾		497		1302							3290
	8	0	8	515	6		568		1488							3760

NOTE 1: Projection of couplers beyond buffers in case of Brooklyn Rapid Transit, Boston, and Hudson & Manhattan, assumed to be 2 in., and for New York Municipal Railway 1½ in. NOTE 2: The horse-power per car of New York Municipal Railway Corporation proposed subway cars, 280 h.p. NOTE 3: The weight of each passenger assumed to be 140 lb. NOTE 4: Standing capacity figured at 1½ sq. ft. per passenger, allowing 9 in. at longitudinal seats for knee room. NOTE 5: No deduction made for motorman's cab in calculating passenger capacity.

TABLE OF COMPARATIVE DATA FOR VARIOUS ELEVATED AND SUBWAY CARS.—(*Continued*)

Road and class of service	Weight in pounds							Per cent weight on drivers	
	Total		Per foot of train (light)	Per passenger seat light			Per max. pass. capacity (seated and standing)		
	Light	With max. pass. load (seated and standing)		Rush hour service (min. seating)	Med. traffic (med. seating)	Light traffic (max. seating)		Light	Loaded (max. capacity)
	77,870	101,530	1496	1769	1693	1622	460	59.5	57.3
	51,410	75,070	987	1168	1118	1071	304	......	......
Interborough Rapid Transit Subway—steel cars.	155,740	203,060	1496	1769	1693	1622	460	59.5	57.3
	207,150	278,130	1324	1568	1500	1439	408	44.8	41.8
	336,430	454,730	1292	1529	1462	1403	398	41.4	38.3
	414,300	556,260	1324	1568	1500	1439	408	44.8	41.8
	543,580	732,860	1305	1543	1476	1416	402	42.6	39.7
	699,320	935,920	1343	1589	1520	1457	414	46 4	43.5
	85,000	122,800							
	170,000	245,600							
	255,000	368,400							
New York Municipal Railway Corporation—Subway.	340,000	491,200	1268	1089	......	944	315	50.1	49.9
	425,000	614,000							
	510,000	736,800							
	595,000	859,600							
	680,000	982,400							
	71,660	93,220	1455		1433		465	60.4	57.9
	34,500	55,220	719		719		233	......	......
Brooklyn Rapid Transit—Elevated, series motor car and standard trailer.	143,320	186,440	1455		1433		465	60.4	57.9
	177,820	241,660	1213		1200		390	48.6	44.7
	249,480	334,880	1274		1260		409	51.8	48.4
	283,980	390,100	1164		1153		375	45.6	41.6
	355,640	483,320	1213		1200		390	48.6	44.7
	85,900	123,140							
	171,800	246,280							
	257,700	369,420							
	343,600	492,560							
Boston—Cambridge Subway.....	429,500	615,700	1235		1193		323	58.0	55.6
	515,400	738,840							
	601,300	861,980							
	687,200	985,120							

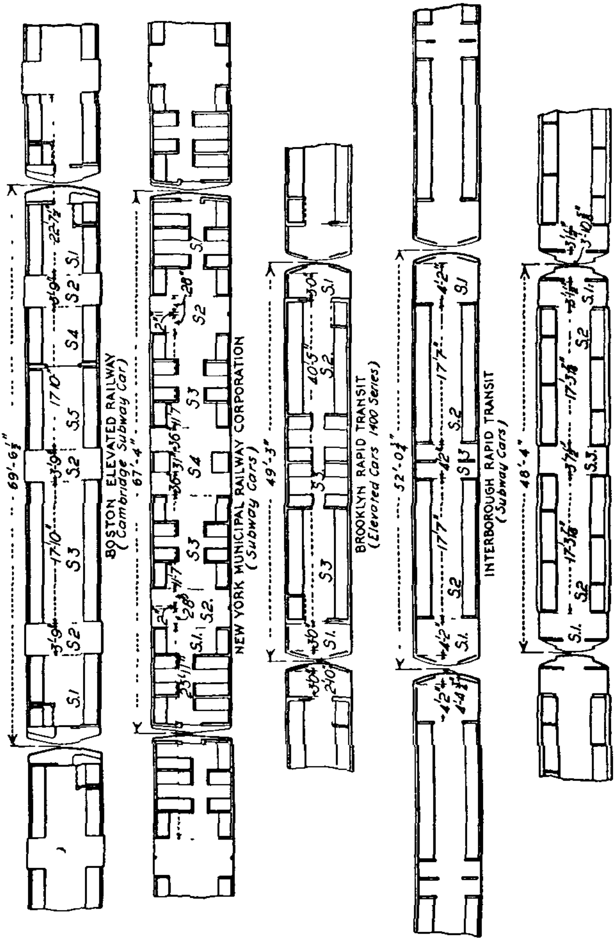
See notes on page 514.

Table of Comparative Data for Various Elevated and Subway Cars.—(*Concluded*)

Road and class of service	Weight in pounds: Total: Light	Weight in pounds: Total: With max. pass. load (seated and standing)	Weight in pounds: Per foot of train (light)	Weight in pounds: Per passenger seat light: Rush hour service (min. seating)	Weight in pounds: Per passenger seat light: Med. traffic (med. seating)	Weight in pounds: Per passenger seat light: Light traffic (max. seating)	Weight in pounds: Per max. pass. capacity (seated and standing	Per cent weight on drivers: Light	Per cent weight on drivers: Loaded (max. capacity)
Hudson & Manhattan—Original type. Tunnel service.	74,100	96,220	1533		1684		469	56.4	54.9
	148,200	192,440							
	222,300	288,660							
	296,400	384,880							
	370,500	481,100							
	444,600	577,320							
	518,700	673,540							
	592,800	769,760							
Hudson & Manhattan—Pennsylvania Newark service.	73,000	95,120	1510		1658		462	54.1	53.1
	146,000	190,240							
	219,000	285,360							
	292,000	380,480							
	365,000	475,600							
	438,000	570,720							
	511,000	665,840							
	584,000	760,960							
Long Island Railroad Subway—Suburban Flatbush Avenue division.	79,564	101,124	1550		1530		516	61.0	58.6
	40,000	59,040	860		715		294	...	.....
	159,128	202,248	1550		1530		516	61.0	58.6
	199,128	261,288	1335		1245		448	48.7	45.4
	278,692	362,412	1390		1315		466	52.2	49.2
	318,692	421,452	1290		1190		434	45.6	42.2
Long Island Railroad Subway—Suburban Pennsylvania station.	104,400	130,440	1620		1470		562	58.7	56.4
	208,800	260,880							
	313,200	391,320							
	417,600	521,760							
	522,000	652,200							
	626,400	782,640							
	730,800	913,080							
	835,200	1,043,520							

See notes on page 514.

ing to passenger capacity must be given than on the shorter run high speed city lines. On most subway and elevated lines it is physically impossible to seat all rush-hour passengers throughout



the trip even with a maximum length of train and the shortes possible headway. Since the rapid transit line cannot suppl seats to all of the rush-hour passengers, an arrangement must b made which will provide maximum standing and seating room fc the heavy and light hours, respectively. Figures 20 to 24, inclusive, show typical floor plans with door and seating layouts of various

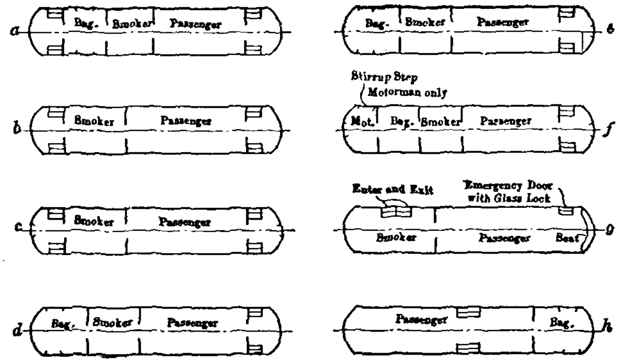


FIG. 25.—Outline plans for interurban passenger cars.

types of city rapid transit cars, and the tables on pp. 512 to 516, inclusive, give various data relative to the weights, capacity, motor equipment, etc., of these cars.

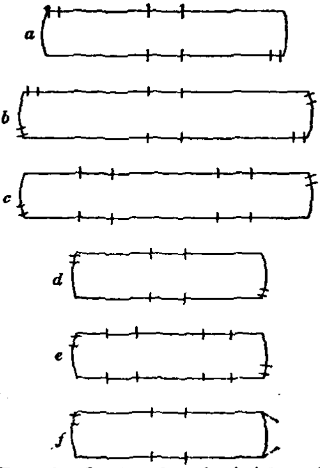


FIG. 26.—Outline plans for freight and express cars.

Interurban Cars. In the design of interurban cars it usually is desirable for the comfort of the passenger on the longer trips to provide a full equipment of cross seats. It is generally deemed desirable also to provide a separate smoking compartment and a compartment for baggage and light express matter, the necessity or desirability of these compartments becoming greater as the headway becomes longer and the distance between terminals greater. It is also desirable on long runs, and in some states, compulsory, that toilet facilities be provided. The layouts of several cars for heavy interurban service are shown by Figs. 15, 16, 17 and 18; the general dimensions and weights of these cars are shown in the table on page 509.

A Joint Committee on Transportation Engineering, A.E.R.E.A., found that when considered with regard to entrances there are eight different general styles of interurban cars designed for train operation. Those shown by Fig. 25, *a* to *e*, inclusive, are the most common types; *a*, *b*, and *c* are double-end and the others are single-end cars. Where there is any possibility of train operation, end doors should be provided. Swing doors are preferable to sliding doors for all but the baggage compartment because they allow more rigid and strong bulkhead construction, cause less trouble and have a lower maintenance cost.

For light interurban service several companies have adopted cars as light as 15 to 18 tons, fully equipped, with layouts and dimensions similar to those shown for suburban service by Figs. 3, 12, 13 and 14; the general dimensions and weights of which are shown in the table on pages 508 and 509.

Freight and Express Cars. Where shipments are made in less than carload lots there should be two large doors on each side of the car. Fig. 26 shows typical outline plans for freight and express cars. Cars *a*, *b*, and *c* are motor cars and the others are shorter designs for trailers. Car *a*, having one large and one small door on each side, is satisfactory when the car is not so long that time is lost in carrying freight between door and end of car. In *b* the small end doors facilitate the handling of long pieces of freight. Car *c* affords the most rapid handling of freight. In *d*, *e* and *f* the corner doors permit passage between cars. Car *f* is provided with an end door for the handling of automobiles and other bulky freight.

Types of Framing. The various types of framing at present in use may be described briefly as follows:

First. Wooden framing with sills reinforced with steel plates; this type is now practically obsolete, having been superseded by the more extensive use of steel for the principal members, a scientific application of pressed and commercial shapes of steel resulting in a stronger and more durable frame.

Second. Composite framing using steel plates for the lower panels, wood or thin steel for the upper side panels, and wood for the posts and superstructure. A heavy steel plate for the lower outside panel of a car is desirable from a maintenance standpoint on account of its durability and the resistance it offers against collisions and abrasions. With this type of construction a light wooden superstructure is the most suitable, the upper side panel being of poplar, agasote or thin steel, applied in sections so as to facilitate renewals.

Third. A composite framing using built-up steel girder side construction and steel floor framing with wood or thin steel as the outside sheathing.

Fourth. Semi-steel construction using thin steel sheets on the outside of the body up to the window sill, steel floor members with wood for side posts and roof construction.

Fifth. All steel body, including side posts with thin sheet steel for outside sheathing and with a steel body plate riveted to the top of the posts to form a girder of the side of the car.

Sixth. The curved side body in which the side girder sheet is curved as shown in Fig. 30. The widest dimension is at the belt rail, and the side posts are shaped in sharply. This gives good vehicular clearance with maximum width at seat line and clearance at the top of car to allow for side sway. The side posts between windows extend only down to the belt rail and are not used to support the roof structure. Wide steel end panels form corner piers which support the weight and take the strain of the roof.

By the use of properly designed members in the steel construction of the types last listed above, it is possible to build a light body, and probably no other method produces the same strength at a corresponding weight. An objection to these types of construction is that if the thickness of steel is to be governed by the required strength only, the material will be so thin as to be dented easily by slight collisions. Riveting also becomes difficult, and unless corrosion can be positively eliminated, the life of the structure will be comparatively short. If steel of proper thickness to overcome these objections is used, the weight of the car will be very greatly increased. This emphasizes the extreme desirability of a design to make accessible for frequent painting all steel parts so as to eliminate corrosion. Some experimental construction has been

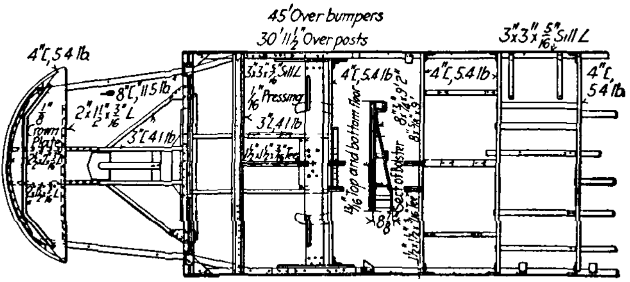


FIG. 27.—Steel underframe of Boston light weight double truck car.

made using monel metal and various forms of non-corrosive steel for various parts. There is as yet no general agreement as to the advantages of such special material and in some cases inferiority for structural purposes is considered.

Construction Details. Figure 27 shows the details of the steel underframe of the Boston light weight double truck car including a section of the body bolster and showing the stiffening included to care for couplers. Figs 28 to 31, inclusive, show details of upper framing in four different cars and show various methods of combining the steel frame and wooden parts. The Minneapolis car shown in Fig. 31 uses a continuous steel channel combining side post stiffening and roof members.

Aluminum in Car Body Construction. Aluminum in its various forms and alloys is rapidly becoming an important material in car construction. For use as stanchions and grab handles it may be obtained in the form of a hard drawn alloy which is quite rigid. For electrical conduit it may be obtained in an annealed form. For long unsupported stanchions it is found that the 1-in. size is not quite stiff enough for the service, and usually it is better practice to use 1¼-in pipe size for this purpose. Although aluminum retains a fairly good finish after continuous use and eliminates the necessity for painting stanchions and railings, it may tend, while new, to blacken the hand of a passenger. By careful cleaning after installation, this difficulty can, to a certain degree at least, be overcome. At least one entire installation of aluminum air pipe has been made, on a sample car built in Chicago. Some difficulty was experienced in cutting threads, and it was also found difficult to pull the pipe up tightly enough to hold the air without breaking it off at the threads. Some trouble was anticipated due to the effects of vibration, but after nearly two years of service, no serious difficulty has developed. Precautions were taken to prevent corrosion due to local galvanic action where the pipe enters steel or iron fittings, by painting the threads with heavy insulating varnish instead of the customary pipe fitting compounds.

In the form of the alloy known as "duralumin," rolled sheet aluminum is available in heat-treated form, and develops a tensile strength of 50,000 to 60,000 lb. per sq. in., which is ample for many car structural requirements. The alloy contains approximately 3 to 4½ per cent copper, 0.4 to 1 per cent magnesium, 0 to 0 7 per cent manganese, and the remainder aluminum, with some slight traces of iron and silicon in the form of impurities. The ultimate production of this class of material in the form of heat-treated rolled sections may have a considerable application for car structural purposes.

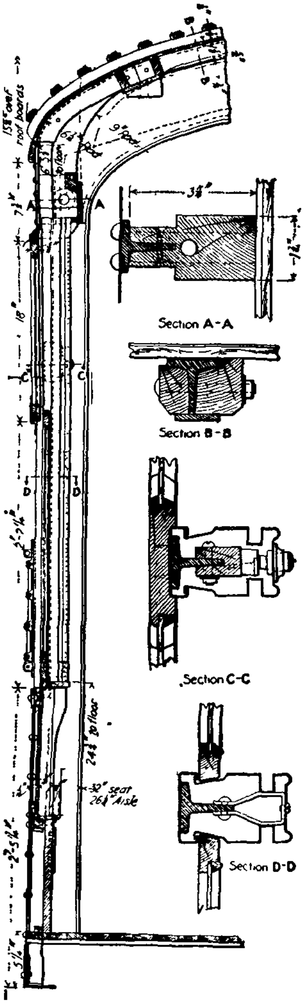


FIG. 28.—Construction details, Baltimore light weight single truck car.

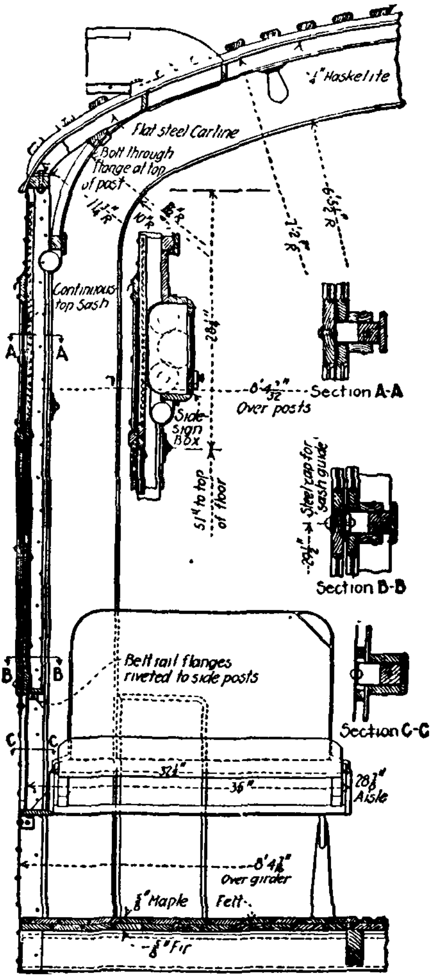


FIG. 29.—Construction details, Chicago light weight double truck car.

Aluminum in a considerable range of alloys is available for castings, and these are being applied quite generally and to a constantly increasing extent for car body construction. The castings may be heat treated, and in that form develop physical

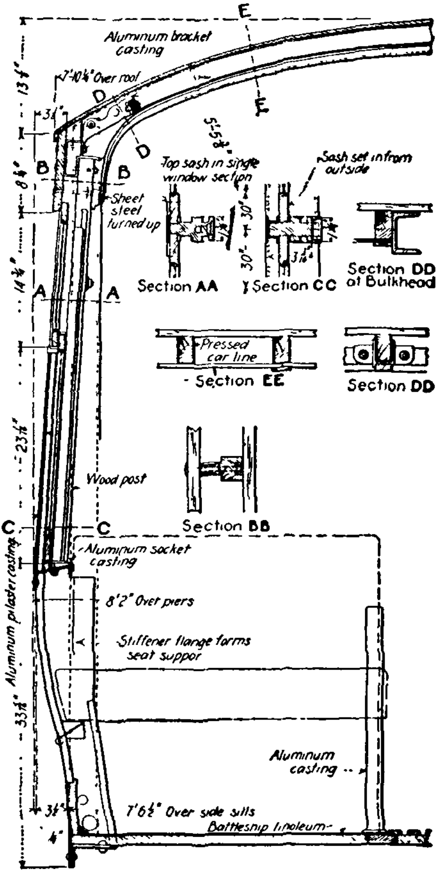


FIG 30—Construction details, curved side car.

properties which enable this material to be substituted for brass and in some cases for malleable iron, at considerable saving in weight. On a recent lot of cars in Chicago, the controller frames and tops were made of aluminum at a saving of 150 lb. per car and with no increase in cost.

Painting. The *Electric Railway Journal* sent out a questionnaire to a large number of electric railways in 1923, requesting information as to practice in car painting. The following data is from the replies received:

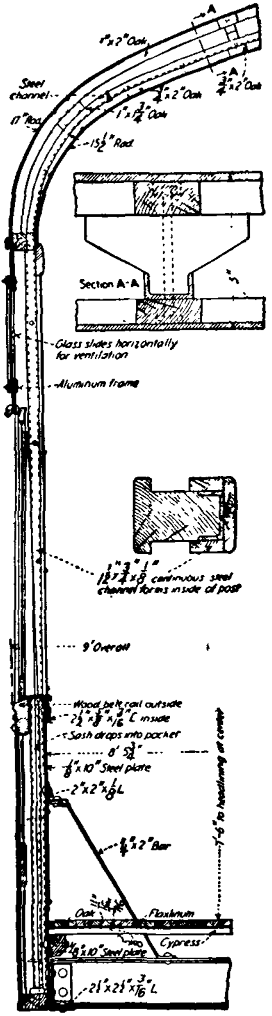


FIG. 31.—Construction details, Minneapolis car.

As to frequency of painting, 37 per cent reported less than two years; 47 per cent, two to three years; 12 per cent, five to six years; and 4 per cent, over six years. It appears to be the general practice to burn off paint from wooden cars and to use a paint remover on steel cars. In reply to a question on this subject, sixteen companies reported that they followed this practice. Ten others reported that they burned paint off, and seven replied that they used a paint remover. Other replies stated that the outside paint is burned off and a paint remover is used on the inside; that a sand blast, together with a paint remover is used; that the wood is burned off and the metal parts are sand blasted; and that the panels are burned off and a paint remover is used on other parts.

Answers to a question regarding the work done between general paintings indicate that most railways touch up and revarnish their cars between what they consider general paintings, without removing the old paint. Of the forty answers received, thirty companies replied that they did touch up and varnish between general painting. Six replied that they did not. Three replies stated that their practice was to touch up and revarnish only after accidents, and one replied that their company did very little touching up.

In regard to the difference in painting practice with steel and wooden cars, replies indicated that on the majority of roads, the only difference is in the priming coat. Thirty-eight companies which had both wooden and steel cars reported on this. Twenty-eight replied that the priming coat constituted the only difference, and ten replied that they used practically the same process for both steel and wooden cars.

An attempt was made to obtain information as to the extent that spray painting was being used and the various parts that were being painted by the spray system. Replies indicated that two-thirds of the railways were not employing the spray painting system. Of the remaining one-third that were using it, seven companies replied that they were spraying the trucks and underbodies of cars. Two replied that they sprayed trucks only, two that they sprayed platforms, hoods and trucks; two more that they sprayed box and flat cars, and an additional two that they were spraying trucks and work cars. Of the remaining replies indicating the extent to which the spray process was used, one company reported using the spray method for the entire outside of the car. Another stated that of the seven coats which were applied on the car, five of them were applied by the spray method, and in addition, the roof, truck, pilot, steps and bumpers were painted by the spray method. An investigation at the shops of some of the companies which are employing the spray method quite extensively indicates that the economies resulting from the use of this system are quite pronounced, and that manufacturers are co-operating toward supplying paints and varnishes which can be used and applied readily by the spray method. Spray painting is particularly adapted for work in restricted places and is economical in the quantity of material needed.

Aside from the special systems which are used by some railways, the various painting systems can be classified in general into three classes. These are (1) the flat color and finishing varnish system, (2) the color varnish and finishing varnish system, and (3) the enamel system. Of the forty replies received in answer to the questionnaire, fourteen railways reported as using the enamel system, thirteen used the color varnish and finishing varnish system, seven used the flat color and finishing varnish system, two reported using both the flat color and finishing varnish and the enamel systems and four reported the use of special systems.

After removing the paint or, where this is not entirely removed, after the loose paint and that which shows signs of cracking has been removed, those using the *flat color and finishing varnish system* apply a priming coat. This is followed by a glazing or knifing coat when holes and nailheads are puttied and spotted. In general this is followed by two rough coats or surface coats. The number, however, varies with different properties; some apply but one coat, and others three. Next follow two or three color coats, and after the lettering and striping usually two coats of finishing varnish are applied. Some roads, however, use only one coat of finishing varnish. Variations frequently will be found, as when cars are in fairly good condition some of the various coats may not be necessary.

Where the *color varnish and finishing varnish system* is used, the color varnish coats replace the flat color coats. Some roads use but one coat of color varnish; others two, and some use a combination, applying first a flat color coat followed by a color varnish coat. After the striping and lettering, the cars are finished by the application of either one or two finishing varnish coats.

Where the *enamel system* is used, the enamel coats replace the flat color or colored varnish coat. From the information received it appears that the most general practice is to apply two enamel coats. Some roads add a coat of finishing varnish over the enamel, and two roads reported the use of a clear enamel coat over the colored enamel coat. A large number of railways do not use finishing varnish with the enamel system. There are various combinations of these systems, as some roads use a flat body coat before applying the enamel coat, and other combinations are possible depending upon conditions.

As cars constitute the point of contact between the traveling public and the railway company, it is essential that the general appearance of the interior and exterior be kept in a condition pleasing to the eye. The use of paint to improve the appearance of rolling stock is then a most important consideration. One of the best mediums for selling transportation is to adopt bright, pleasing colors. Another reason for using brightly colored cars is that of safety, and many companies adopt a combination of colors to give the cars the greatest possible visibility. In order to operate cars at high speed and at the same time to promote the safety of the public, it is desirable that an approaching car should be distinguished as far as the eye can see. Other reasons for keeping rolling stock properly painted are for sanitary effect and to improve lighting conditions. In order to ascertain how far paints are used by electric railways for these two latter-named purposes, companies were asked specifically if they used paint to improve lighting conditions and for sanitary reasons. Eighty-three companies replied to this question, fifty-one of which (61 per cent) said that these were important considerations.

Door Operation. In city and suburban service the operation of the outside doors of both sliding and folding types generally is by some mechanical means operated from the motorman's or conductor's position. Such doors are frequently operated mechanically through an arrangement of shafts, rods and bell cranks or bevel gears, and often are operated pneumatically by means of door engines, consisting of cylinders with pistons properly connected to the operating mechanism, the air control being by suitable valve at the operator's position.

Car Couplers. The selection of coupler equipment is limited by three conditions: (1) the short radius of track curves, (2) the abruptness of changes in track grade, and (3) the construction and dimensions of cars. The center line radius of track curves should be as great as conditions permit, 40 ft. or greater if possible, and never less than 35 ft. If short radius curves must be used, spiraled approaches offer effective corrective possibilities. The greatest coupler swing on a given car and curve is when the car is fully within the curve and the coupler anchorage of the adjacent car is directly over the point of tangency of the track curve. The amount of change in grade is of little importance if sufficient distance is available for making the change. It is the change in grade which must be made in a short length of track that affects coupler clearance. Such conditions usually obtain at railroad crossings where street and railroad grades are permanently established at different levels, at the approaches to some bridges, and at points where streets dip below railroads or other overhead structures. Other special conditions must be considered such as temporary crossovers, emergency hose bridges used at fires, or transfer tables with short approaches. As one car platform rises or falls with respect to the adjacent platform, its coupler anchorage lifts or depresses, and this movement affects the clearance between coupler head and car buffer, as the straight line between anchorages must be maintained at all positions the two cars may assume. The projection of the coupler beyond car buffer when in center line position should always be sufficient to keep the hood rims of coupled cars separated at the bottom of the most abrupt track grade and should never be less than one and one-half times the coupler draft gear travel. Coupler selection is dependent further upon car dimensions and design. The coupler equipment demands a certain space under the car platform, and the nature and importance of its service requires that this clearance space may not be invaded even by such important items as car steps, brake apparatus, fenders or door operating mechanisms. When truck center spacing, as well as the length of car, is fixed, the only two variables left are coupler length and anchorage position, and this leaves small opportunity to secure a favorable maximum swing of couplers or entirely acceptable clearance space for car steps. The contour of car ends should be such that coupled cars be separated as little as possible. The buffer radius should approximate the coupler length and preferably be slightly less than swinging length of coupler. Such platforms permit close spacing of cars and improve corner clearance on curves. The stresses on car underframes when coupled in trains differ from those of single car operation, and sills for anchorage mounting should be capable of withstanding and transmitting the loads imposed in normal operation and the shocks incident to coupling.

Automatic Couplers. The automatic coupler should have the following qualities: (1) automatic coupling without necessity of preliminary adjustments; (2) simple and safe means of uncoupling; (3) ample vertical and horizontal alining ability of coupler head in coupling; (4) no loose parts subject to loss or misplacement; (5) ability to operate under all weather conditions and after long periods of idleness; (6) freedom from relative movement at coupler faces; (7) strength to withstand abnormal operating conditions and abuse; (8) provision for free vertical movement as well as horizontal swing of coupler; (9) provision for limited torsional movement to permit train to travel curves having elevated outer rail; (10) simplicity of construction; (11) ease of repair. Unless interchange of traffic demands a coupler of the M.C.B. type, the following points should be added: (12) automatic coupling of air brake pipe lines, and (13) provision for attachment of automatic electric couplers.

The Tomlinson automatic coupler is one of the widely used types. One of several modern forms is shown in Fig. 32. This coupler consists of a cast steel head or drawbar fitted at its face with a hardened steel forging in the form of a hook which constitutes the chief guide member for alining purposes during coupling, and by snapping into engagement with a similar hook when coupler faces are brought together acts as the tension member in haulage. Projecting dowel pins at the coupler face serve to aline the air line rubber gaskets which are above and below the hook on the center line of the coupler face. A heavy draft spring is housed within the enlarged rear end of the coupler body and absorbs both tension and compression shocks. The coupler proper terminates at the rear end in a two-piece socket clamp which encloses the machined ball of the coupler anchorage. The two halves of this coupler socket interlock to form a retaining cup for lubricating oil contained in a well within the anchorage ball. This ball and socket joint provides in one bearing for all the necessary movements of coupler head. Adjusting bolts and shims permit the socket to be adjusted for any wear which may occur. The uncoupling chain hangs from the end of the projecting cam handle. The carrier supports the projecting coupler head, when uncoupled, at a proper height to facilitate coupling, and is supported by springs which permit the coupler head to move downward, with clearance above to permit lifting of the coupler from its spring support. An overhead supporting bar, bent to curved outline, permits the carrier to swing to either side on track curves.

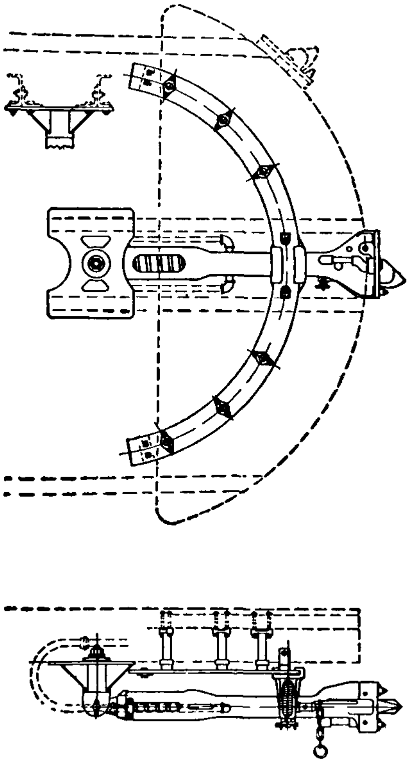


Fig. 32.—Typical coupler mounting on car.

An efficient device for establishing electric circuit connections between cars is an essential feature of modern automatic coupler equipment. The most simple condition of motor car and trailer operation requires the extension of buzzer and single stroke bell circuits from the motor car for signaling purposes, as well as a lighting and a heater circuit. Usually door signal circuits are added to increase speed and safety of operation. The operation of multiple unit motor cars in trains has always required some form of multiple conductor jumper cables, the functions of which may be combined with electric couplers. The electric couplers, as shown in Fig. 33, operate on the push-button principle. The electrical contact parts are insulated in molded composition blocks which are mounted in metal cases and bolted directly to the machined side faces of the coupler head. Their proper operation is assured by perfect preliminary alinement and the straightforward coupling movement of the mechanical couplers. This method of mounting has proved to be most practical, as it combines easy accessibility for installation and inspection with maximum track clearance. It is general practice to retain cut-out cocks in the air brake pipe lines back of the hose jumpers leading to coupler, and to insert disconnecting switches between car wiring and electric couplers. The fact that both air line cocks and electric coupler switches must be operated whenever cars are coupled or uncoupled leads to an obvious combination and interlocking of these parts which simplifies mounting arrangements and enforces their proper operation. The disconnecting switch has a rotating drum, operated by pull rods attached to the ends of an external operating handle, which makes and breaks contacts with metal fingers arranged on each side of the drum. At each end this drum terminates in a socket which serves to throw the air cock which is mounted between pipe lugs on the switch end. The mounting is such that both air cocks are open when the switch is closed and closed when the switch is open. By this arrangement exposed electric coupler contact points on uncoupled couplers must always be dead, as the car could not operate with the switch closed because the air cocks would then be open and the air brake would act automatically to lock the wheels. The disconnecting switch must be mounted in some accessible space convenient for air piping connections and the attachment of switch handle operating rods, and preferably it should be located where wheel wash will be avoided and periodic inspection can be made conveniently.

M.C.B. Type Couplers. There has been adopted as standard by the A.E.R.E.A. for use on interurban cars a coupler of the vertical plane type which has the same contour lines of knuckle and guard arm as, and which will automatically couple with, standard steam railroad couplers of the M.C.B. type. The draft rigging and drawbar supports for these couplers should be such that, with sudden changes in grade, the vertical displacement of the couplers with reference to each other will not be sufficient to cause the knuckles to become disengaged. The length of the radial coupler, measured from the center of the pocket-pin to the pulling face, should be 54 in. This length will apply to both interurban cars and to cars in city service. On city cars, where the bumper arrangements will permit, a pocket casting should be placed on the top of the bumper, the center of the pocket to be 35 in above the top of the rail, and the casting to be of ample strength and properly braced, so that by means of a suitable bar city cars may be coupled on a level with the automatic couplers of interurban cars. For this purpose it is advisable at present to maintain a link slot and coupling pin hole in the knuckle of the automatic couplers. The following are the A.E.R.E.A. standard specifications for couplers for interurban cars where the interchange of equipment is involved:

1. All couplers must be made to couple automatically by impact, and to uncouple without the necessity of going between the

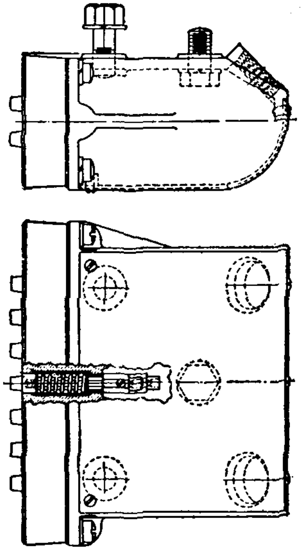


FIG. 33.—Electrical train line connection (14 point) for automatic car coupler.

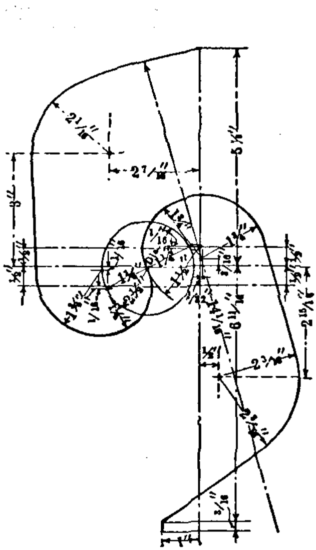


FIG. 34.—Contour line, M.C.B. standard coupler.

cars, with M.C.B. and all other types of M.C.B. contour couplers, whether used by steam or electric roads.

2. A device should be adopted for holding couplers on center within the required limits when inter-coupling with steam railroads.

3. An open knuckle for shackle bar connection should be used.

4. The draft gear, where possible, should meet M.C.B. requirements, *and the drawbar anchorages should be equivalent in* strength to M.C.B. equipment and requirements.

5. Couplers must not uncouple when cars are being pushed around a curve of 35-ft. center radius.

6. There should be an arrangement to release and open the knuckle without requiring the operator to pass between the cars.

7. The face of the knuckle vertically should be 16 in. maximum.

8. The height of the drawbar center should be 31½ in. minimum and 34½ in. maximum above the head of the rail.

9. The coupling center of couplers must have a minimum projection of 6 in. beyond buffer faces at any point between the working limits of couplers.

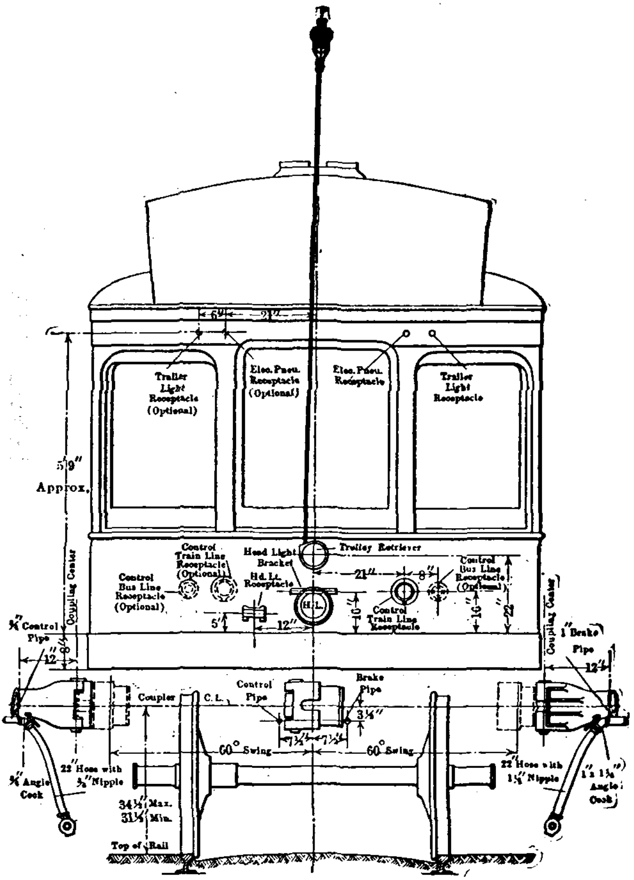


FIG. 35.—A.E.R.E.A. recommended location of end connections for interurban cars.

10. Couplers should be placed on both ends of the cars.

The contour line for automatic couplers adopted as standard by the Master Car Builders' Association is shown by Fig. 34.

Standard Heights of Couplers, Platforms and Bumpers. Following are A.E.R.E.A. standards. Height of city car couplers, 20 in. from top of rail to center of coupler. Height of interurban car platforms, 51 in. above top of rail. Height of bumpers for interurban cars, 43 in. from top of rail to bottom of bumper; for city cars, 31 in. from top of rail to top of bumper; width of city car bumpers, 6 in. Bumper on city cars to include pocket to permit coupling with interurban cars. Where possible, provision to prevent telescoping in case of collision between city and interurban cars.

End Connections for Interurban Cars. The locations of the various end connections for interurban cars, adopted as recommended practice of the A.E.R.E.A., are shown in Fig. 35. In this figure the locations shown for control train and bus line electropneumatic and trailer light receptacles are such as to clear end doors where these

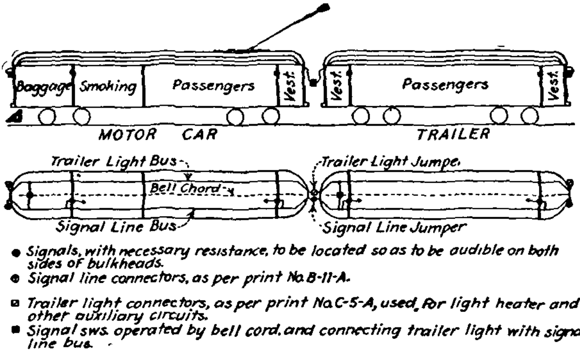


FIG. 36.—C.E.R.A. standard location of signal whistles and wiring.

are employed in the event conditions permit; if it is otherwise required, their position may be altered, but the re-location should be on the vertical center, as shown. The receptacles indicated by dotted lines are optional when not required; the trailer light receptacles are optional when bus lines are used. The headlight receptacle and bracket and trolley retriever bracket are optional when not required.

The Standardization Committee of the Central Electric Railway Association has proposed a standard location of electropneumatic signal whistles and wiring, as shown by Fig. 36, and also a standard trailer light connector, as shown by Fig. 37.

Miscellaneous Equipment for Interurban Cars. A Committee on Maintenance and Inspection, A.E.R.E.A., recommended that interurban cars should never be put in service without the following miscellaneous equipment:

Three sets of flags (red, white and green); telephone, where standard to the road; classification and marker lamps where oil lamps are used; two trolley pickups; one coupler or pulling bar; one pull rope; coupling link and pin; extra supply of air pump and light fuses (also car and control fuses where used); fire extinguisher in working order; one extra trolley pole, fully equipped, on top of the car; one extra trolley rope, or, better still, one extra retriever equipped with rope; one trolley retriever in its place on the rear dash; one headlight in its place on the front dash, for signal use. Fuses and torpedoes should be on each car. The crew should have both red and white lanterns in good condition, and sufficient tools to change the trolley pole or make other light repairs. Interurban cars should also carry a wrecking outfit with axe, saw, jack and crowbar.

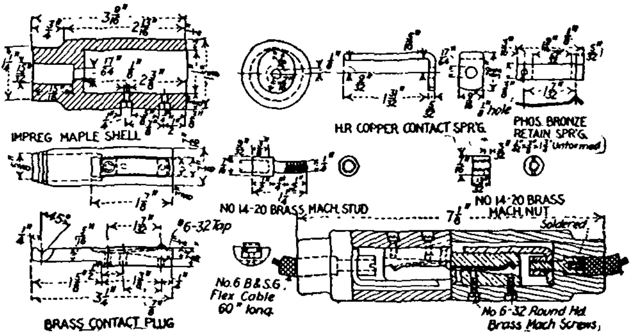


FIG. 37.—C.E.R.A. standard trailer light connector.

Track Sanders. Satisfactory results from the use of sand will depend very largely upon the efficiency of the sanding device itself and the character of material with which it is supplied. Owing to the restricted conditions on an electric car, it is usually very difficult to install the older types of mechanical gravity sanders with anything like a satisfactory arrangement. This difficulty is also augmented on account of the very sharp curves and other conditions which usually prevail or at least restrict the operation to some few particularly bad locations. This has led to the general use of pneumatic sanders as being superior to the gravity type in that they are more readily applied and have the advantage of distributing the sand evenly and expeditiously at the proper point on the rail directly ahead of where the front pair of wheels makes contact with the rails. The sand should be applied only in sufficient quantity to give maximum traction and braking power, and it is especially important that the application should be just previous to or at the time of the application of the brakes and before the braking power is high enough to skid the wheels. This applies particularly to emergency brake applications and is one of the special advantages of the pneumatic type over any of the gravity sanders. The pneumatic type, however, requires very careful installation and arrangement of the piping connections in order to insure reliable and positive results when called upon under the most trying conditions of weather and roadway. It also is necessary to use a device requiring the minimum amount of air and sand as well because of the tendency to overload the compressor with minor pneumatic devices which were not taken into consideration when the capacity of the air compressor was determined. The flexible connection between the sand box, usually carried inside the car, and the discharge pipes attached to the trucks is usually a source of considerable trouble and requires careful attention. The style of the sand valve should be such as to avoid useless waste of air. It should also be located conveniently near the brake valve so that the two operations of applying sand and setting the brakes can be done in emergency at practically the same time.

Character of Sand. The best quality of good sharp quartz sand, thoroughly dried and screened and free from dirt or soil, should be used. Dirty sand is more susceptible to moisture and consequently its tendency to clog up the pipes is greater, aside from the harm it may do after reaching the rail. Lake sand has been found quite satisfactory and is extensively used in pneumatic sanders on account of its fine, even grain and freedom from foreign matter. It is also easily dried and screened and is generally economical. The character of the sand is worthy of more important consideration than apparently is usually given to it. In some cases it is entirely lacking in the essential qualities and has a tendency not only to defeat the object for which it is applied, but actually to assist in creating a more serious condition.

Car Heating. The usual methods of heating cars are (1) coal stoves, direct method; (2) coal stoves, indirect method, similar to hot air furnace used for heating of houses; (3) hot air heaters, air blast, motor driven; (4) hot water heaters; (5) electric heaters. The characteristics of the various methods should be considered with regard to the following points: (1) ability to heat car to uniform temperature; (2) first cost, completely installed on car; (3) maintenance; this will include repairs, renewals, replacements, etc.; (4) cost to operate; in the case of hot water heaters, this will include fuel and labor only; in the case of electric systems, power only, and in the case of the hot air blast heater using coal, fuel, labor and power; (5) weight of system complete as installed on the car ready to operate; (6) fire risk; (7) reliability; (8) regulation; this refers to ability to regulate the heat to outside temperature; (9) space occupied; (10) appearance; (11) attention required from car crew; (12) cleanliness, which will include freedom from dust, ashes and obnoxious gases; (13) adaptability and relation to ventilation systems.

Aside from the ordinary coal stove, the three principal heating systems are the hot air heater, hot water heater and electric heater.

Hot Air Heater. In this system the air is heated by a coal fire and forced through suitable ducts along the side of the car by motor-driven fans. By its use it is possible to secure quite uniform heating of the car. The heat being applied along the floor line results in dry floors, which is a strong point in its favor. The first cost is about the same as that of an equivalent electric system wired in conduit and equipped with thermostatic control. The cost to operate, including only the items of coal, labor of attendance and cost of electricity for the motor is comparatively low. The weight depends upon type and size of car, but is practically the same as electric, and less than hot water heaters. The fire risk is practically the same as in the case of hot water heaters. The regulation is not so good as it should be, but will undoubtedly improve as the apparatus is further developed. When this system is used in conjunction with exhaust ventilation, the regulation is better. The space occupied is a little greater than with hot water heaters. The appearance compares favorably with other types of coal heaters. Considerable attention is required from the car crew from time to time in order to keep fire in proper condition, but where the heater can be placed near the conductor or motorman, this is readily accomplished. It is not so clean as the electric system, but compares favorably with hot water. As this system is designed to provide for ventilation, it is readily adapted to that end.

Hot Water Heater. This system possesses many valuable characteristics, among which are independence of the electric power supply, which is quite a consideration in interurban work where long runs are made and the power supply is subject to interruption. By the use of this type of heater it is possible to heat the car very uniformly. The efficiency of hot water heaters will fall off materially if the pipes and coil are not kept reasonably free from scale and other deposits. The first cost is greater than that of the hot air system. The maintenance is higher than that of the electric system. The cost to operate, including only the items of fuel and labor, is approximately the same as for the hot air heater. The weight of the hot water apparatus is high and has long been one of its chief drawbacks, but the latest types show improvements in this respect. The fire risk is substantially the same as for the hot air heater. In case of accident there is a hazard from the fire in the coal stove. The reliability is very good. The regulation is comparatively poor. It takes some time for water to take up heat and, conversely, it takes some time for water to lose its heat. The space occupied is considerable and, except on single-end cars, this space is valuable as seating or standing room. The hot water heater with its expansion drum, water glass gage, etc., does not add to the appearance of a car, except where it is practicable to partly enclose the apparatus. Attention is required from time to time, but the work is small and where the heater can be located close to one of the crew, it does not take him from other duties. The hot water heater as usually installed produces dust, and very frequently obnoxious gases. The heating elements being pipes located one above the other near the floor line, it is easy to adapt this system of heating to any practical scheme of ventilation.

Electric Heater. It is perfectly possible to secure uniform temperature throughout the car. The first cost is lower than that of any of the other modern systems. When the electric heater is carefully installed with wiring in conduit, the maintenance is very low, being considerably less than that of any of the other modern systems. The cost to operate, which includes energy only, is variable, depending on the size of car, the range of temperature, the cost of energy, and whether the peak load on power stations comes in the heating seasons or not. In general, the cost of operation, as defined above, is high. There are many cases, however, where this method of heating will show the greatest net economy. The weight is the least of any of the modern heating systems, except in some cases, where it is practically the same as the hot air heater. Where the wiring is properly installed in conduit the fire risk is practically nil. This system is very reliable. The regulation is best of all, it being possible to follow closely and without trouble rapid changes in outside temperature. The space occupied is very small and is not useful for standing or seating capacity. The appearance of such parts as are exposed is very good. The electric heating system requires the minimum amount of attention from car crew. This type of heater is clean and free from dirt or obnoxious gases. The heating units being subdivided and located under the car seats or along the truss plank are readily adaptable to any practical system of ventilation.

In the installation of electric heaters, it is preferable to have a comparatively large number of heaters rather than a few, even though the power consumption is on the same basis, on account of the better distribution of the heat. For localities where the temperature reaches zero or lower, it is well to have about 4.5 watts per cubic foot of car body, otherwise it may be difficult to maintain a comfortable atmosphere in the cars when low temperatures prevail.

Comparative Costs of Car Heating. As a comparison in costs of heating a car by the three modern systems, the following estimates are given. The figures are based, in general, on results obtained in practice.

Basic Data (all systems)

Length of car body to be heated (inside) approx.	40 ft.
Width of car body to be heated (inside) approx.	8 ft. 3 in.
Height of car body to be heated (inside) approx.	7 ft. 6 in.
Cubical contents of car body to be heated (inside)	2400 cu. ft.
Weight of car without passenger load	34,000 lb.
Average daily car mileage	150 mi.
Average season car mileage	22,500 mi.
Average length of heating season	150 days
Average temperature during Oct., Nov. and April	38 deg. F.
Average temperature during Dec., Jan. and Feb.	24 deg. F.
Average lift of temperature above outside air	35 deg. F.
Average temperature to be maintained	65 deg. F.
Heat units required per hour per cu. ft. of air to be heated*	13
(above equals a heater capacity of about)	10 kw.
Kw-hr. consumption per car mile (2-motors city service)	2.25
Watt hours per ton mile	132
Weight hot air equipment, including air duct	460 lb.
Weight hot water equipment, including piping and water	1600 lb.
Weight electric heating equipment, including wiring and conduits	400 lb.
Cost of coal per net ton at car house	$13.00
Cost of current per kw-hr. at car	$0.01
Cost of haulage per lb. per season of 150 days	$0.015

* Includes allowances for all radiation and air change losses; also for inefficiency of heaters due to re-heating through operation of thermostat.

Cost of Hot Air Heating System per Season of 150 Days

Interest charge—6 per cent on $200 per complete equipment, installed	$12.00
Depreciation charge—7 per cent on $200 cost per equipment	14.00
Maintenance per equipment @ 0.007 per heating hour	25.20
Attendance, including kindling of fires, removal of ashes and handling of coal from bin to car @ 12c per day	18.00
Current cost—to operate blower motor = 275 watts per hour × 18 hours per day × 150 days × $0.01 per kw-hr	7.43
Fuel cost—full heat for 90 days = 80 lb. hard coal per day × 90 days = 3.6 tons × $13.00 per ton	46.80
Fuel cost—3/4 heat for 60 days = 60 lb. hard coal per day × 60 days = 1.8 tons @ $13.00	23.40
Fuel cost—banking fires 6 hours per day 15 lb. hard coal per day × 150 days = 1 ton hard coal @ $13.00	13.00
Kindling—at $0.03 per day for 150 days	4.50
Haulage—460 lb. × $0.015 per lb. for 150 days	6.90
Labor—removing heater at termination of heating season and replacing in fall	2.00
	$173.23
Average cost per day for season of 150 days	$1.16

Increased insurance rate on car barns and rolling stock is 2c per $100.

Cost of Hot Water Heating System per Season of 150 Days

Interest charge—6 per cent on $300 per complete equipment installed	$18.00
Depreciation charge—7 per cent on $300 per complete equipment installed	21.00
Maintenance—per equipment @ $0.005 per heating hour	18.00
Attendance—including kindling of fires, removal of ashes and handling coal from bin to car @ $0.12 per day × 150 days	18.00
Fuel cost—full heat 90 days = 90 lb. per day × 90 days = 4 tons @ $13.00	52.00
Fuel cost—3/4 heat 60 days = 65 lb. per day × 60 days = 2 tons @ $13.00	26.00
Fuel cost—banking fires—6 hours per day × 15 lb. hard coal × 150 days = 1 ton @ $13.00	13.00
Kindling—at $0.01 per day for 150 days	1.50
Haulage—1600 lb. (approx.) × $0.015 per lb. for 150 days	24.00
Labor—removing heater at end of heating season and replacing in fall	4.00
	$195.50
Average cost per day for season	$1.30

Increased insurance rate on car barns and rolling stock is 2c per $100.

COST OF ELECTRIC HEATING SYSTEM PER SEASON FOR 150 DAYS

Interest charge—6 per cent on \$175 cost of 10 kw. electric heating complete, installed, including wiring, conduits and thermostatic control..........	\$10.50
NOTE: 10 kw. equipment is approximately equivalent in heating capacity to the hot air and hot water equipment covered by above tables.	
Depreciation charge—7 per cent on \$175 per equipment..................................	12.25
Maintenance—\$0.005 per heating hour on 1775 hours operating per season..........................	8.88
Haulage cost—400 lb. @ \$0.037 per lb. for 365 days..	14.80
Current cost—full heat = 10 kw. × 18 hours per day × 90 days × \$0.01 per kw-hr. ÷ 80 per cent time heat is on full..........................	162.01
Current cost—⅔ heat = 10 kw. × 18 hours per day × 60 days × \$0.01 per kw-hr. ÷ 60 per cent time heat is on..................................	64.80
Current cost—To heat up car before taking out of car house in morning = ½ hour per day × 10 kw. × 120 days × \$0.01..............................	6.00
NOTE: This item must be included to compare with hot air and hot water heaters in which fire is either maintained all night or built up early in the morning before car leaves barn.	
	\$279.24
Average cost per day per season of 150 days......	1.86

From the above it is seen that, *under the conditions assumed*, the relative total economy of the three principal heating systems is as follows: Hot air system, first; hot water, second; and electric, third. If the conditions are different from those taken as typical, different results will be obtained; for example, if, under the above assumptions, the peak load on power stations came in summer time, then the electric heating system might show the greatest total economy, due to the fact that in the latter case the proper charge for energy would be only the fuel cost, while if the peak comes in the heating season, the charge for energy should include all fixed charges on power plant *and distribution system*. It is, therefore, readily apparent why it is impossible to say off-hand what the cost of car heating is unless all the conditions are known. As stated before, in the choice of a system for heating cars some consideration other than that of total economy may govern, such as appearance, space factor, ability to regulate, fire hazard, or the requirements of the State or municipality.

In figuring the power consumption of electric heaters, the following method will probably give the most accurate results: Obtain from the weather bureau temperature readings for each winter for several years. Plot a curve showing variation of temperature for each day of the heating season. Find what circuit in heaters is used for various temperatures and plot a power curve which will indicate the average kilowatts per day.

In the use of hot water or hot air heaters there is a tendency on the part of the car crew to use less coal than is necessary to keep the cars at a uniform temperature during the time they are in service, while with electric heaters the tendency is to put on three points where two points would suffice. This may give rise to false ideas of the relative costs of the various heating systems.

Thermostatic Control of Electric Heaters. A number of devices have been designed for the automatic regulation of electric heaters, the successful operation of which very greatly reduce the cost of electric heating, in some cases the tests showing a saving of some 50 per cent. One type of thermostatic control is illustrated in Fig. 38, by reference to which the operation of the control circuit is explained as follows: Normally current flows through resistance *A*, the magnet coil of switch *E* and resistance *B* to ground, and switch *E* is held closed. Current then flows direct from the trolley through the blowout coil and the contacts of switch *E* and then through the heaters to the ground. As the car warms up, the column of mercury in the thermometer rises. When it touches the upper platinum contact, a current of very low voltage is shunted around resistance *B*, through fuse *C*, the coil of relay *D*, and the thermometer, to

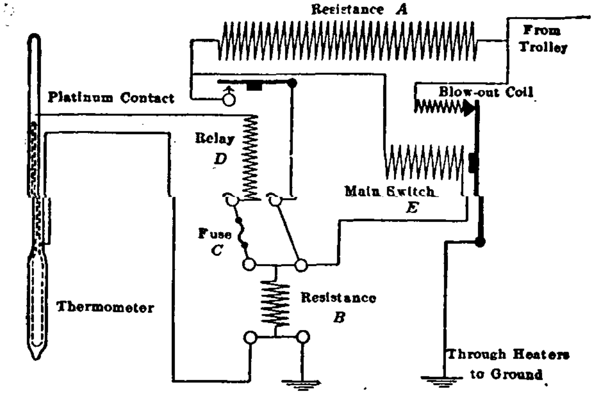


FIG. 38.—Electric thermostat control for car heaters.

ground. Relay *D* is thus energized and its contacts close. This short-circuits the magnet coil of switch *E*, and that switch opens by gravity, cutting current off the heaters. When the temperature falls so that the mercury of the thermometer leaves the platinum contact, the thermometer circuit is broken, and relay *D* is no longer energized. Therefore, its contacts open, breaking the shunt around the coil of switch *E*. The latter is then energized and closes, turning current on the heaters. A rise and fall of less than 1 deg. is sufficient to turn the heaters off or on as required.

Before the adoption of thermostatic control for electric car heaters in Chicago, tests were made to determine the probable energy saving. Of the three types of regulators tested, two utilized a mechanical expansion element operating a relay circuit through which the main heater switch was controlled; the third a mercury contact thermometer with relay circuit. One of the types tested possessed the advantage of double adjustment so that cars might be partially heated when standing in the yard just before being put into service. Another was so wired that current remained on the heating circuits in case of any failure of the thermostat and thus was under ordinary control by trainmen. Three of the test cars had monitor roof with natural ventilation and three arch roof and motor-driven ventilators to eliminate as much as possible the effects of variance in car body design and ventilation. The current used in heating these cars was compared with that used in standard cars of the respective types not equipped with the thermostat. With the latter, an effort was made to conform to the ordinance requirement of 50 deg. F. as closely as possible, although it was found that more heat was ordinarily used than authorized by the operating department due to the impossibility of close adjustment by hand. The amount of energy used by the heaters represents in approximate figures from 1/10 to 1/6 the average energy requirement of the car for propulsion alone. During the coldest winter weather this heating load averages per car as follows: 1st point, 3.43 kw.; 2nd point, 6.87 kw.; 3rd point, 10.30 kw.; resulting in an increase in total peak load at the power station from 15 to 20 per cent above non-heating requirements. The tests showed that the average saving over non-automatic control, when expressed in per cent, varied from nothing at 15 deg. F. where full heat was required, to a maximum where no heat was required, viz., 50 deg. F.:

Natural ventilation,	temperature 20 deg., average saving 21 per cent
	temperature 30 deg., average saving 40 per cent
	temperature 40 deg., average saving 56 per cent
	temperature 50 deg., average saving 69 per cent
Forced ventilation,	temperature 20 deg., average saving 8 per cent
	temperature 30 deg., average saving 23 per cent
	temperature 40 deg., average saving 44 per cent
	temperature 50 deg., average saving 65 per cent

As some trainmen keep heat on above 50 deg. the results actually showed some saving above the 50 deg. limit. Expressed in actual energy, the results were fairly constant throughout the range of the test:

Natural ventilation, saving 2.2 kw. per car average
Forced ventilation, saving 1.2 kw. per car average

Ventilation of Cars. Authorities differ as to the amount of air to be supplied per person per hour, in order to provide a reasonable standard of air purity. An ordinance of the city of Chicago on this subject calls for the supplying of 350 cu. ft. per hour per passenger (based on maximum standing and seated load); provided, however, that the air in the car shall at no time show more than 12 parts of CO_2 in 10,000 parts. It was found possible to meet this requirement by any one of several ventilating systems.

Monitor Deck Windows. The usual method of obtaining ventilation in electric railway cars has been to provide a monitor deck or clear story in which there are a number of small windows that can be opened. The ventilation afforded by these small windows is largely by dilution; that is, such air as may enter serves to freshen the air in the car. The chief objection to the system is that, under some conditions of operation, strong draughts are created which are objectionable to passengers.

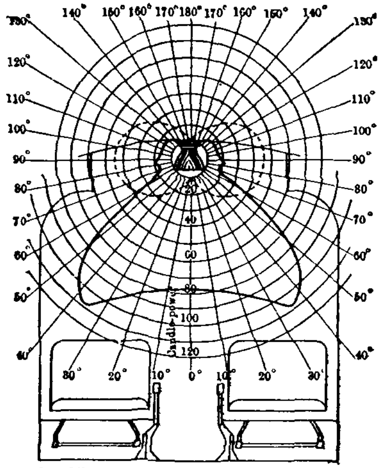
Types of Ventilators. Ventilation systems other than monitor windows have been worked out principally along two lines: those operated by the movement of the car through the air and those operated by motors. The first are usually called the automatic systems and the second, mechanical systems. Automatic ventilators are usually "exhauster" and should be so designed as to exclude rain and snow and to prevent gusts of air coming into the car. The action of automatic systems is, of course, variable, depending on the velocity of the car and the direction and force of the wind. The mechanical systems, of which there are two principal kinds, the "exhauster" type and the "blower" type, are positive and practically independent of motion of the car or velocity of the wind.

Mechanical System of Ventilation. A typical mechanical system consists of a motor-driven exhaust fan located on the vestibule roof, or other practical point, an exhaust chamber formed in the upper ceiling of the car, openings located at various points in the ceiling and intakes located at several points in the floor and connected to the electric heaters. The cold air thus is made to pass over the heating surface before coming into the car body which is generally appreciated as a very desirable feature. Tests show that consumption of energy for heat is not increased by this method. The fan-motor set consists of a ½-h.p. motor, direct-connected to a specially designed 9-in. cone fan. This fan will handle about 33,000 ft. of air per hour under normal conditions of line voltage. The motor is connected direct to the 500-volt trolley circuit through a standard combination snap switch and fuse, and is started and stopped by means of this switch. The motor and fan are mounted in a suitable metal housing which is connected to the exhaust chamber. The fan discharges through protected openings in each side of the housing. The exhaust chamber in the upper part of the car is formed by dropping down the ceiling about 4 in. from the roof framing and is continuous from end to end of the car body. Communication between the car interior and the exhaust chamber is provided by a number of openings, each containing a circular adjustable register. By proper adjustment of the registers a uniform velocity of air through all of them is obtained. There are several intakes, half being located on each side of the car under the seats in such a manner as to be readily connected to the electric heaters. The connection between the screened opening through the car floor and the electric heater is made with a pressed metal duct. The size and number of the intakes is such as to permit of a maximum velocity of the air of about 400 ft. per minute, which is hardly perceptible to passengers.

Automatic System of Ventilation. The automatic systems installed are of several different kinds, but all depend upon aspirator action for their operation. One of these automatic systems which has shown fair results comprises a number of exhausters located along each side of the monitor roof and attached to panels placed in the monitor or deck window openings. An opening in the panel communicates with the exhauster. These exhausters are also designed for use with plain arch roof cars. Intakes similar to those described in connection with the mechanical system are located in the floor and provide a supply of fresh air. The exhausters are rectangular sheet-metal boxes projecting outwardly from the panels to which they are secured, having openings top and bottom, and provided on the middle of each side face with V-shaped projections. The V projections are placed horizontally on the faces of the exhausters and split the air into two streams, one following upward and the other downward. The air streams flow past the openings of the exhauster and by induction draw the air out from the car.

Location of Air Intake. There is a tendency to abandon intakes near the floor line owing to street dust. On a number of European lines, notably the Budapest suburban system, louver type ventilators are installed over the side sash instead of using a monitor roof.

Car Lighting. Electric car lighting, formerly exclusively by means of the carbon filament lamp, has recently been made much more economical by the use of the tungsten lamp, which gives an efficiency of 1.25 watts per candle-power and has a useful life equal to or greater than that of the carbon lamp. On account of its resistance characteristics, the tungsten filament is much less susceptible to changes in candle-power with varying voltage than the carbon, so that it will stand a higher overvoltage than the latter, and will also give a fair illumination at low voltage under which the carbon lamp gives practically no light. Tests made by the Bay State Street Railway (*Elec. Ry. Jour.*, 1912–1913) indicate the economy of using a small number of large tungsten units rather than a large number of small units, as was the almost universal practice with carbon lamps.



Dotted line curve represents distribution from bare 56-watt Mazda lamps.
Solid-line curve represents distribution from 56-watt Mazda lamps equipped with a prismatic clear reflector.

FIG. 39.—Light distribution—Tungsten lamp with and without reflector.

Fig. 39 shows the comparative light distribution from bare lamps and lamps with reflectors.

The Committee on Equipment, A.E.R.E.A., 1914, stated that the minimum acceptable intensity of car illumination is 1.5 foot-candles, and in order to provide for a voltage drop to approximately 80 per cent normal, the car lighting system should be so designed as to give an average intensity of illumination of at least 3.75 foot-candles at normal voltage in a plane 36 in. above the floor. (See table, page 543.)

Size and Number of Lamps, and Types of Reflectors Required to Produce a Minimum Intensity of 3.75 Foot-candles at Approximately 36 In. from Car Floor, at Rated Voltage, for Various Lengths of Car Bodies. (A.E.R.E.A. Committee on Equipment, 1914)

Lamps		Length of car body					
No. of units in body of car	Kind	25 ft.	30 ft.	35 ft.	40 ft.	45 ft.	50 ft.
5	64-watt carbon.	....	....	....	....	....	....
	23-watt Mazda.	Not	sufficie	nt watt	age	Too wid	e space
	36-watt Mazda.	....	....	....	....	....	....
	56-watt Mazda.	Prism H. opal	Prism H. opal	....	....	....	....
	94-watt Mazda.	M.opal L. opal	M.opal L. opal	Prism H. opal M.opal	Prism H. opal M.opal	....	....
10	64-watt carbon.	....	... Not	sufficie	nt watt	age....	....
	23-watt Mazda.	Prism	Prism	Prism	....	....	....
	36-watt Mazda.	H. opal	H. opal	H. opal			
	56-watt Mazda.	M.opal L. opal	M.opal L. opal	M.opal L. opal	Prism H. opal M.opal L. opal	Prism H. opal M.opal	Prism H. opal
	94-watt Mazda.	Bare	Bare	Bare	Bare	L. opal	M.opal L. opal
15	64-watt carbon.	....	....	....	....	....	....
	23-watt Mazda.	Prism H. opal M.opal L. opal	Prism H. opal M.opal	Prism H. opal	Not su	fficient	wattage
	36-watt Mazda.	....	L. opal	M.opal L. opal	Prism H. opal M.opal L. opal	Prism H. opal M.opal	Prism H. opal
	56-watt Mazda.	Bare	Bare	Bare	Bare	L. opal	M.opal L. opal
	94-watt Mazda.	....	....	....	....	Bare	Bare
20	64-watt carbon.	....	... Not	sufficie	nt watt	age...	....
	23-watt Mazda.	Prism H. opal M.opal L. opal	Prism H. opal M.opal L. opal	Prism H. opal M.opal	Prism H. opal M.opal	Prism H. opal	....
	36-watt Mazda.	Bare	Bare	L. opal	L. opal	M.opal L. opal	Prism H. opal M.opal L. opal
	56-watt Mazda.	....	Bare	Bare	Bare	Bare	Bare
25	64-watt carbon.	....	... Not	sufficie	nt watt	age....	....
	23-watt Mazda.	Prism H. opal M.opal L. opal Bare lamp	Prism H. opal M.opal L. opal	Prism H. opal M.opal L. opal	Prsim H. opal M.opal L. opal	Prism H. opal M.opal	Prism H. opal
	36-watt Mazda.	....	Bare	Bare	Bare	L. opal	M.opal L opal
	56-watt Mazda.	....	....	....	....	Bare	Bare

Storage Battery Lighting. Especially on interurban lines with long headways, economy sometimes dictates the use of less feeder copper than will maintain a voltage which is at all times satisfactory for car lighting. In order to obviate the annoying effect of a change in intensity of illumination caused by such fluctuations, the Lehigh Valley Transit Company equipped a number of interurban cars with ten cells of A-6 Edison storage batteries arranged in two trays of five each, each tray weighing slightly under 100 lb. This battery gives on discharge 240 ampere hours. The cars are equipped with twenty 20-watt, 16-c.p., 12-volt tungsten lamps with holophane reflectors. The lamps are wired in multiple and connected to the battery through an ampere-hour meter which is compensated to run 20 per cent slower on charge than on discharge and with a pointer showing the limiting capacity of the battery. The battery and ampere-hour meter are arranged to be removable for charging, which is done by a small motor generator set giving 50 amperes at 125 volts, which is set up in one corner of the car house and which charges six sets of batteries at one time. The batteries are pulled out of their compartment under the car onto a small truck when the cars turn in at the car house on the last trip at night, and the sets are all placed in series for charging and cut out one at a time as they come fully up to the charge, which is determined by the individual ampere-hour meter. The twenty 20-watt lamps on each car require approximately 33 amperes and the batteries will deliver that current for 7½ to 8 hours continuously, which meets the requirements for one day's run. The initial voltage of the ten Edison cells is about 13 volts at the beginning of discharge which gradually drops to 10 volts at the end of the discharge, and it is stated that there is not a sufficient variation in the candle-power of the lamps to be appreciated without the aid of a photometer. It is stated that the batteries give no more trouble in handling and charging than the ordinary arc headlight and that the system produces an absolute direct lighting system in the cars unaffected by the variation in or even entire failure of the power supply. Although the charging is somewhat inefficient, the total energy consumption including all losses is still considerably less than with the ordinary carbon lamp equipment.

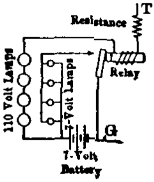


FIG. 40.—Train marker lamp wiring, Indianapolis.

Electric Lighting of Train Marker Lamps. As it is absolutely essential that train marker lamps remain lighted even during a period of temporary failure of power supply, there has been some hesitancy on the part of electric railways in departing from the oil-burning lamp for this purpose. When electric lamps are used, some method should be provided for maintaining the light during such periods of failure of power supply, and this is generally done by some arrangement of storage battery. A scheme as used by the Terre Haute, Indianapolis & Eastern Traction Company, is shown diagrammatically by Fig. 40. A 7-volt storage battery is used in series with four 110-volt, 16-c.p. lamps and a relay in the light circuit opens an auxiliary circuit which consists of four 7-volt, 4-c.p. lamps connected in parallel and operating in series with the storage battery when the trolley current is off. One 110-volt lamp and one 7-volt lamp are installed in each marker.

Indicating Tail Lights. The Nichols-Lintern indicating tail lights employ two lanterns, one provided with a red lens, the other with a green lens. The lanterns are mounted on the ends of the car, one on each side of the trolley catcher. The circuits controlling the lamps in these tail lights are interlocked with the control apparatus so that a red lamp is lighted when the car is stationary or coasting, with power off; both red and green lights are burning when the controller is in series or half-speed position; with the controller in full multiple or full speed position the green light alone

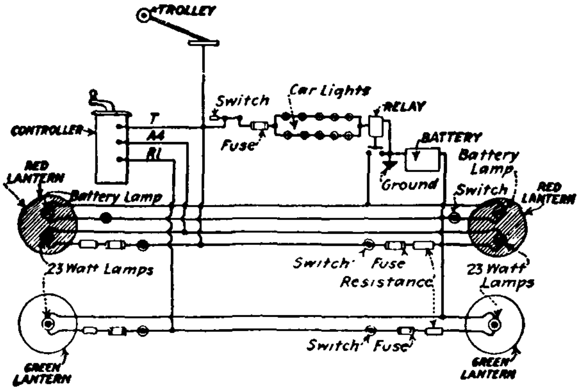


Fig. 41.—Wiring diagram for indicating tail lights.

burns. Fig. 41 shows the connections necessary to accomplish these results. As shown in Fig. 41, the lights are arranged with an auxiliary battery circuit which supplies current to a lamp in the red lens in case the main power supply should fail, as when the trolley leaves the wire; this feature may be included or not as desired. The safety and economical features claimed for this device are: in congested sections cars can operate closer to the car ahead without danger of accidents; by having an indication as to the operation of the car ahead, the motorman is able to coast more and apply his brakes less than otherwise; automobile and truck drivers behind cars equipped with these indicating tail lights are able to regulate their speed to that of the car and thus avoid collisions.

SECTION X

TRANSMISSION AND DISTRIBUTION

A.I.E.E. Standardization Rules. The A.I.E.E. standards (1922 revision) which apply particularly to electric railway transmission and distribution are as follows:

Contact Conductors. A contact conductor is that part of the distribution system other than the traffic rails, which is in immediate electrical contact with the circuits of the cars or locomotives.

Contact Rail. General: A contact rail is a rigid contact conductor.

Overhead Contact Rail. An overhead contact rail is a contact rail which is above the elevation of the maximum equipment line.

(The maximum equipment line is the contour which embraces cross-sections of all rolling stock under all normal operating conditions.)

Third Rail. A third rail is a contact conductor placed at either side of the track, the contact surface of which is a few inches above the level of the top of the track rails

Center Contact Rail. A center contact rail is a contact conductor placed between the track rails, having its contact surface above the ground level.

Underground Contact Rail. An underground contact rail is a contact conductor placed beneath the ground level.

Gage of Third Rail. The gage of a third rail is the distance measured parallel to the plane of running rails, between the gage line of the nearer track rail and the inside gage line of the contact surface of the third rail.

Elevation of Third Rail. The elevation of a third rail is the elevation of the contact-surface of the third rail, with respect to the plane of the tops of running rails.

Third Rail Protection. A third rail protection is a guard for the purpose of preventing accidental contact with the third rail.

Trolley Wire. A trolley wire is a flexible contact conductor, customarily supported above the cars.

Messenger Wire or Cable. A messenger wire or cable is a wire or cable running along with and supporting other wires, cables or contact conductors. A primary messenger is directly attached to the supporting system. A secondary messenger is intermediate between a primary messenger and the wires, cables or contact conductors.

Classes of Construction. *General:* Overhead trolley constructions are classed as Direct Suspension and Messenger or Catenary Suspension.

Direct Suspension. A direct suspension is the form of overhead trolley construction in which the trolley wires are attached, by insulating devices, directly to the main supporting system.

Messenger or Catenary Suspension. A messenger or catenary suspension is the form of overhead trolley construction in which the trolley wires are attached by suitable devices, to one or more messenger cables, which in turn may be carried either in *Simple Catenary, i.e.*, by primary messengers, or in *Compound Catenary, i.e.*, by secondary messengers.

Supporting Systems. *General.* Supporting systems for trolley wires shall be classed as follows:

Simple Cross-span Systems. Simple cross-span systems are those having at each support a single flexible span across the track or tracks.

Messenger Cross-span Systems. Messenger cross-span systems are those having at each support two or more flexible spans across the track or tracks, the upper span carrying part or all of the vertical load of the lower span.

Bracket Systems. Bracket systems are those having at each support an arm or similar rigid member, supported at only one side of the track or tracks.

Bridge Systems. Bridge systems are those having at each support a rigid member, supported at both sides of the track or tracks.

Transmission System.* When the current generated for an electric railway is changed in kind or voltage, between the generator and the cars or locomotives, that portion of the conductor system carrying current of a kind or voltage substantially different from that received by the cars or locomotives, constitutes the transmission system.

Distribution System.* That portion of the conductor system of an electric railway which carries current of the kind and voltage received by the cars or locomotives, constitutes the distribution system.

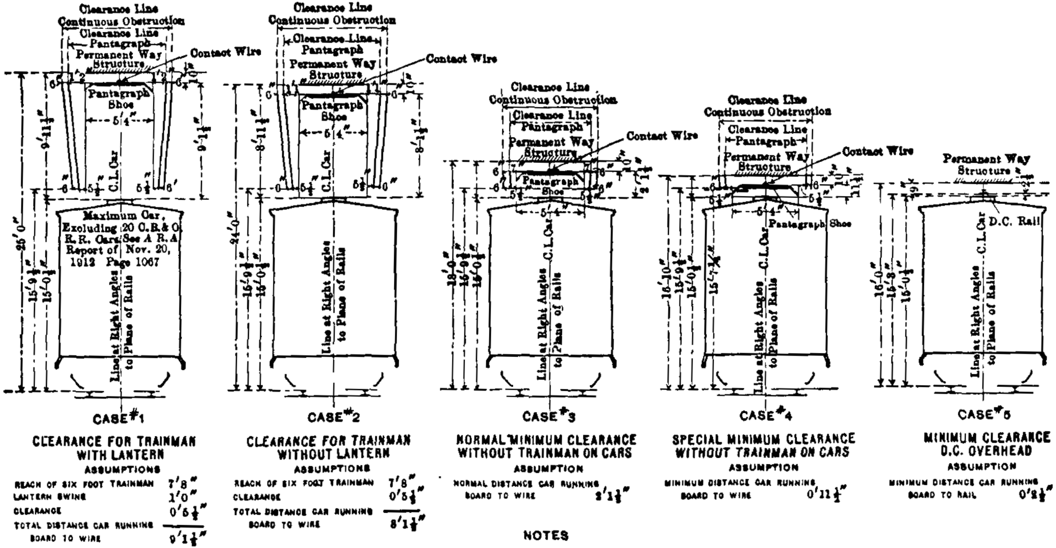
Substation. A substation is a group of apparatus or machinery which receives current from a transmission system, changes its kind or voltage, and delivers it to a distribution system.

Standard Height of Trolley Wire on Street and Interurban Railways. (*Same as A.E.R.A. Standard.*) It is recommended that supporting structures shall be of such height that the lowest point of the trolley wire shall be at a height of 18 feet above the top of rail under conditions of maximum sag, unless local conditions prevent. On trackage operating electric and steam road equipment and at crossings over steam roads, it is recommended that the trolley wire shall be not less than 21 ft. above the top of rail, under conditions of maximum sag.

Standard Gage of Third Rails. The gage of third rails shall be not less than 26 in. and not more than 27 in.

Standard Elevation of Third Rails. The elevation of third rails shall not be less than 2¾ in. and not more than 3½ in.

* These definitions are identical in sense, although not in words, with those of the Interstate Commerce Commission, as given in their Classification of Accounts for Electric Railways.



NOTES

Momentary Obstructions such as Signal Blades may approach Pantagraph Clearance Line.

Sway of Pantagraph Based on One Inch Difference in Hight of Car Springs, One-half Inch Difference in Elevation of Track Rail, and Sway of Six Inches Either Side at Twenty-two Feet above Top of Rail for Pantagraph Itself.

These Diagrams show Minimum Clearances; Additional Clearances will be Required to Provide for Special Features of Design, Sag between Points of Support as Affected by Length of Span and Temperature Changes, and also for Steady Strains, Pull Offs etc., if any.

All Hights to be Measured at Right Angles to Plane of Rails at Center Line of Track.

... of clearances for overhead working conductors. A.E.R.A. and A.R.A. recommended design.

Overhead Trolley Construction

The following notes on overhead trolley construction are principally taken from the Recommended Specifications of the American Electric Railway Engineering Association.

Classes. Overhead trolley construction may be classed as (*a*) direct suspension, (*b*) catenary suspension. Direct suspension comprises construction in which the trolley wires are attached, by suitable devices, directly to the main supporting system. Catenary suspension comprises construction in which the trolley wires are attached, by suitable devices, to one or more messenger cables whch in turn are carried (*a*) in simple catenary, by main supporting system; or (*b*) in compound catenary, by secondary messengers which in turn are carried by the main supporting system.

Supporting systems for direct or catenary suspension may be classed as:

(*a*) Simple span, comprising those systems having at each point of support a single flexible member attached at both sides of the track or tracks.

(*b*) Compound span, comprising those systems having at each point of support two or more flexible members attached at both sides of the track or tracks, the upper member carrying part or all of the vertical load of the lower member.

(*c*) Bracket, comprising those systems having at each point of support an arm or similar rigid member attached at one side only of the track or tracks.

(*d*) Bridge, comprising those systems having at each point of support a rigid member attached at both sides of the track or tracks.

Supporting structures are the devices which sustain the supporting system, and may be poles, whether of wood, steel, or concrete, towers, buildings, trees, or any other form of support, together with their anchors, guys, braces and similar reinforcing attachments. The type of supporting structure will be governed largely by local conditions. In general, natural wood (or concrete) poles are used for all interurban construction and wherever else practicable; steel poles may be used in streets if so desired; sawed poles and tree attachments should not be employed, and building attachments should be used only when local conditions make desirable, in which case special precautions and construction will be necessary. (See page 552.)

Pole Framing. Before setting, wood poles should be roofed, butts squared, entire pole rough shaved, knots smoothed, gains and faces made, and roof, gains and faces given a coat of preservative or paint. The size, number and location of holes, faces and gains vary. In general: holes, unless specifically noted otherwise, should be of same size as bolt or rod for which intended. Faces should be of sufficient area and of proper shape to receive their fittings, and should have about ¼ in. margin outside; they should be slightly hollowed to prevent rocking of fitting. Gains should be square with axis of pole, ¾ in. minimum depth, of width to secure snug fit of arm, and slightly hollowed to prevent rocking. The roof should have a pitch angle of 45 deg. and should be either a wedge with edge parallel to line when pole is set, or a cone.

Pole Clearances. On private right of way and elsewhere when practicable, side supports should be set with a minimum clear distance of 7 ft. from center line of track to face of support at level of top of rail, and center supports should have a minimum clearance of 7 ft. from center of track, this clearance to be increased if necessary on curves to allow for rail elevation and car overhang. Where curb lines are established, poles should be set just behind the curb itself unless local ordinances or conditions prescribe other location.

Pole Spacing. Poles on tangents should be normally spaced not less than 90 ft. and not more than 110 ft. apart. Poles on curves should be set as nearly as practicable in accordance with the table on page 560.

Pole Setting—Depth of Holes. *Pole holes in level ground* should have depths as follows:

Length of pole (feet)	Depth of hole In rock or concrete	In earth
30	5 ft. 0 in.	6 ft. 0 in.
35	5 ft. 6 in.	6 ft. 0 in.
40	5 ft. 6 in.	6 ft. 6 in.
45	6 ft. 0 in.	6 ft. 6 in.
50	6 ft. 6 in.	7 ft. 0 in.
55	6 ft. 6 in.	7 ft. 6 in.
60	7 ft. 0 in.	8 ft. 0 in.
65	7 ft. 0 in.	8 ft. 6 in.
70	7 ft. 0 in.	9 ft. 0 in.

In very compact soil pole holes may have depths intermediate between those for same lengths in rock or concrete, and in earth. The depth of a hole on sloping ground should be measured from the lower side of the hole; and in very steep slopes and in loose or otherwise doubtful material the depth should be increased over the standard depth by an amount to be determined for each case on the ground.

Barrel Holes. In material which caves freely one or more barrels or the like may be used as casing, and driven down as the material is dug from inside. Such barrels or casings may be of wood or steel and should be of sufficient size to give clear tamping room of at least 6 in. around the pole without cutting the latter.

Rake. Wood poles with brackets should in general have a rake from the track of 6 in. in 24 ft.; steel poles with brackets, of 3 in. in 24 ft. Wood poles with span should have a rake from the track of 12 in. in 24 ft.; steel poles with span should have a rake from track of 6 in. in 24 ft. When the strain is from the track, as with poles on the inside of a curve, brace poles or head guys should be used, and standard rake maintained. Double bracket poles should be set without rake; other poles between tracks, and poles under outside jurisdiction may be so set if necessary or required.

Keys. All span poles as well as bracket poles on curves of radius less than 2400 ft., and such other poles as may be designated because of unusual load conditions, should be provided with suitable keys of wood, stone, or concrete, at least 4 in. thick with a cross-section not less than 32 sq. in. One key 2 ft. long should be placed on edge behind the pole at the bottom of the hole; the other key 4 ft. long should be placed on edge in front of the pole just below the surface of the ground.

Refilling Holes. In setting poles the excavated material, if suitable, should be replaced in thin, even layers, and firmly and thoroughly tamped, not more than one shoveller serving three tampers, until the hole is completely filled, after which the earth should be well packed around the pole in a small mound, and if on a slope there should be made on the lower side, a berm at least 6 ft. wide from pole to edge. In rock holes the broken rock should be used to thoroughly wedge the pole in place. Black loam and similar poor material should be replaced by suitable material.

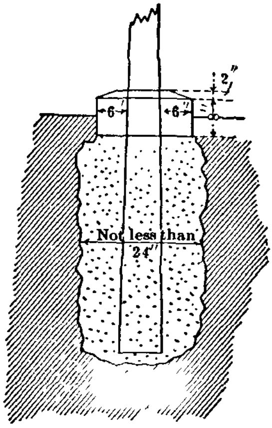


FIG. 2.—Concrete pole setting.

Concrete Settings. Poles subject to heavy lateral strains which cannot well be met by guying or bracing may be set in concrete mixed wet and consisting of one part Portland cement, three parts clean sharp sand, and five parts good hard gravel or broken stone of size to pass screen with holes 2 in. diameter, and to be retained by screen with holes ¾ in. diameter. Concrete settings should have a diameter at least 12 in. greater than that of the pole, and should completely fill pole hole to a level 6 in. below surface of the ground. In parking strips the authorities may require the concrete to finish at this level, but such latter practice is undesirable. Wherever practicable the concrete should be finished as indicated in Fig. 2.

Span Wires Attached to Buildings. Building owners in crowded districts sometimes prefer to have span wires attached to their buildings rather than to have poles in front of the buildings. Eyebolts for the purpose are sometimes placed in the building during construction. As a means of suspension such an arrangement is satisfactory but it must be borne in mind that it subjects the build-

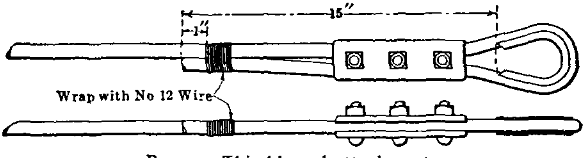


FIG. 3.—*Thimble end attachment.*

ing to unusual strains and the wires are more dangerous in case of breakage as they are over the sidewalk and will sweep it with more force than when attached to poles.

Span and Guy Attachments. Span and guy attachments may be made up with such of the following forms as may be desired:

Thimble End. (Fig. 3.) Thimble end may be made by bending strand around thimble of proper size. The strand end should extend 15 in. beyond thimble, and should be secured by a three bolt clamp close to thimble, and by a serving of about 10 turns of No. 12 galvanized wire 1 in. from end of strand.

Two-turn Wrap. Two-turn wrap may be made by taking strand around pole twice. If at end of span or guy, the end of the strand should extend 30 in. beyond the face of pole and should be secured to main part by a three bolt clamp 15 in. from face of pole and by a serving of about 10 turns of No. 12 galvanized iron wire 1 in. from end of strand. If hitch is at an intermediate point in span or guy it should be secured by a three bolt clamp on the outside parts of the strand, the center bolt of clamp being replaced by a lag screw into the pole as shown by Fig. 4.

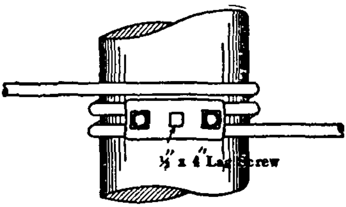


FIG. 4.—Three-bolt clamp hitch.

Close Tie. (Fig. 5.) Close tie may be made by bending strand tightly around attachment, leaving a free end about 15 in. long. One wire of this end should be unlayed back to the attachment and served on main part and remainder of end; the other wires should be in turn unlayed back to the last wrap and served over main part and remainder of end until latter is all served on.

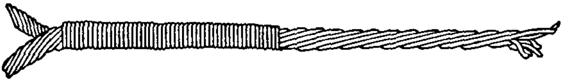


FIG. 5.—Close tie.

Temporary Tie. (Fig. 6.) Temporary tie may be made like permanent tie of same kind, but end should be left long enough to allow for adjustment and permanent make up and should be secured to main part by one or more servings of wire. In making temporary ties bend should be of as large radius as possible until permanently secured.

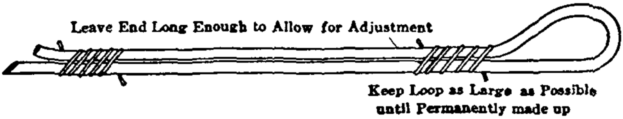


FIG. 6.—Temporary tie.

Anchors. (Fig. 7.) Anchors in earth should consist of a wooden deadman 4 ft. long, at least 6 in. thick, and having a cross-section not less than 48 sq. in., buried at least 4 ft. below the surface with not less than 2 ft. of rock if reasonably obtainable well packed into hole, and the earth filling above thoroughly tamped. The guy rod should pass through center of deadman, and must lie in line of pull of guy to prevent bending. Patent anchors of holding capacity equal to the breaking strain of the strand to be used with them, and having rugged parts sufficiently large to allow a reasonable amount of corrosion without reduction in holding capacity, may be used in place of rod and deadman where conditions are favorable. Anchors in rock should consist of eye-bolt securely

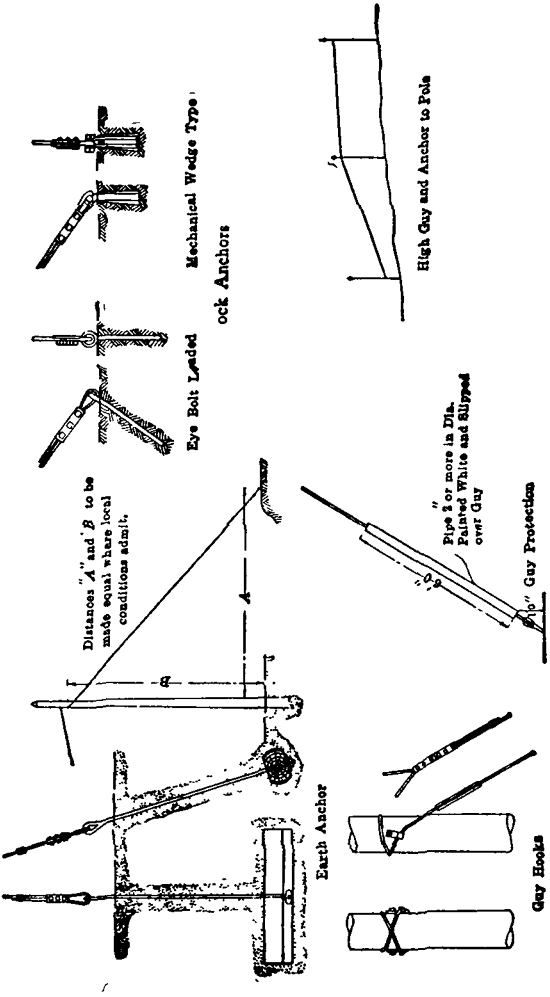


FIG. 7.—Methods of attaching guys to poles and attachments.

leaded or sulphured for entire length of shank in hole 1/8 in. larger in diameter than bolt, and inclined at right angles to pull of guy. In rock of sufficient strength to safely withstand the action, mechanical wedge type eye-bolts may be used, and the lead or sulphur omitted. Where practicable guys may be anchored to adjacent pole at point not less than 7 ft. above ground.

Guys. (Fig. 7.) Guys should be used where practicable on wood poles on curves of radius less than 900 ft., on poles to which are attached strain plate guys, trolley guys and head guys; and wherever any side strain exists. They should be of galvanized seven-wire steel strand with thimble end at anchor attachments, and two-turn wraps at proper height for attachment. The lead or horizontal distance from pole at ground line to guy at same level, whenever practicable, should be equal to the distance from ground line at pole to guy wrap. Guys located where there is a liability of persons or animals running into them should be made conspicuous by a piece of pipe 2 or more inches in diameter and 6 ft. long, slipped over guy, resting on anchor rod eye and painted white. Where guy is already installed, a wooden casing, 3 in. diameter or square and 6 ft. long, may be used in place of the pipe. The halves should be well white leaded, and should clamp the guy tightly when screwed together, the bottom resting on anchor-rod eye. Guy hooks attached one on each side of pole at level of guy wrap by a through-bolt at right angles to line of pull should be used where local conditions compel the use of a lead less than one-fourth the distance from ground line at pole to guy wrap. Where guys cannot be run directly to secure attachments they may be carried high on adjoining poles to a point where anchorage can be obtained.

Cross-arms. (Fig. 8.) Cross-arms should be given one coat of preservative or two coats of paint before pins are installed. Pin shanks should be dipped in the preservative or paint, and while wet firmly seated in hole and secured by a wire nail 2 in long driven through side of arm into pin. All cross-arms should be held to pole by a through-bolt driven from back of pole toward and through arm, having a washer at each end, with hole at back of pole properly counterbored to secure good seat for washer. Cross-arms for light duty service should be steadied by strap braces secured to pole by a lag screw, and to side of arm away from pole by carriage bolts on center line of arm with nuts next to the braces and a washer under bolt head. Arms 36 in. in length should have braces 20 in. long, fastened to arm 12 in. from center; arms over 36 in. long should have braces 30 in. long, fastened to arms 19 in. from center. Cross-arms for heavy duty service should be steadied by an angle brace, fastened to bottom of arm by carriage bolt and to pole by a through-bolt. The lowest feeder, telephone or signal cross-arm should have its center not less than 21 ft. above top of rail; other feeder, telephone or signal cross-arms should be spaced at least 24 in. from center to center. If the pole also carries a transmission line there should be a clear distance of at least 6 ft. between the top feeder, telephone or signal arm and the lowest transmission arm. Double arms for feeder should be used at ends of curves and on intermediate poles of curves of radius less than 500 ft. Double arms must be

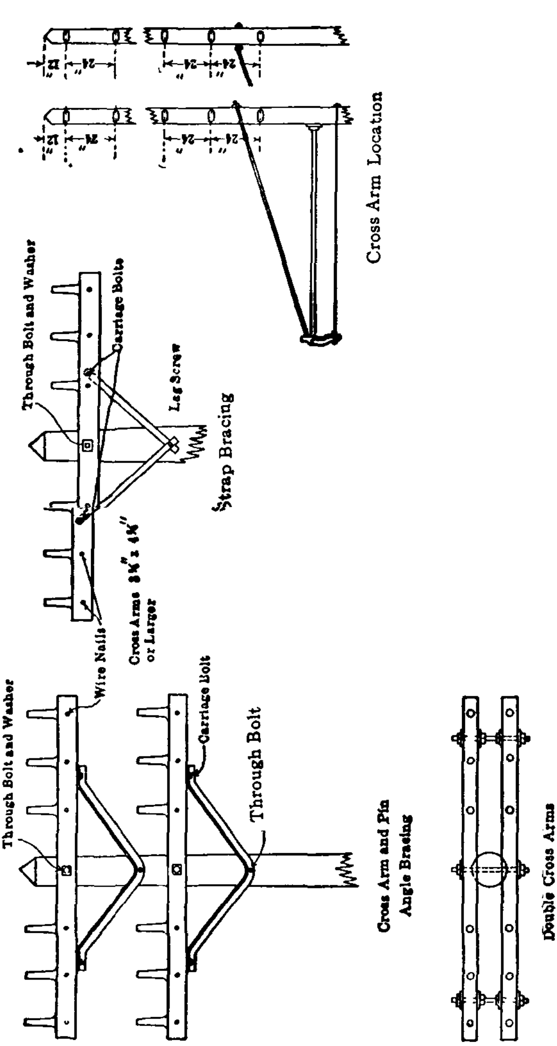


FIG. 8.—Cross-arm attachments.

parallel and at the same height. Both arms should be fastened to the pole by the same bolt, and should be firmly tied together by spacing bolts with nut and washer each side of each arm, located on the center line of arm, 8 in. from end.

Choice of Supporting System. Bracket support on side poles should be used for all single track where local conditions do not prevent, except for curves of radius less than 300 ft. Bracket support on center poles may be used on double track where practicable. Span support should be used on single track for curves of radius less than 300 ft. and where local conditions do not permit use of brackets; on double track, including turnouts, not employing center bracket poles; and for more than two tracks. Compound spans should be used when necessary to support the overhead of a series of tracks too closely spaced to permit poles between. Bridge support should be used only in special cases.

Height of Trolley. Supporting structures should be of such height that the lowest point of trolley wire in streets and on interurban lines will be at a height of not less than 18 ft. above the top of rail under conditions of maximum sag, unless local conditions prevent; on trackage operating electric and steam road equipment and at crossings over steam roads the trolley wire should be not less than 21 ft. above the top of rail under conditions of maximum sag.

Brackets. Brackets should be of sufficient length to allow 8 in. between hanger or strain insulator and the end casting. When poles are on outside of curve the length of bracket should be sufficient to allow for effect of pole rake and rail elevation. The eye-bolt should be installed at level of trolley wire; the pole casting with center a distance above center of eye-bolt equal to distance center to center between span eye and arm socket of end casting; and the oversupport rod a distance in inches above pole casting of three times the length of arm in feet. On wood poles eye-bolts and over-support rod should pass through pole and pole casting should be attached by two lag screws. On steel poles, eye for strand, socket for arm and pole attachment for oversupport rod should be carried by special fittings, clamping to pole. Eye-bolt should be installed pulled out to full length; the nut, on wood pole, seated against a washer; the arm should be given an upward rake from the horizontal of 1 in. in 4 ft. of length; the intermediate casting should be clamped on arm so that trolley wire comes midway between it and end casting; and the steel strand should be close tied into eye-bolt and into end castings with 2 in. of slack to permit hanger installation. On steel poles strain insulators should be cut into strand on either side of hanger, and between end and intermediate castings, to give double insulation. Brackets on curves should be installed as on tangents except that pull-over with attached strains should be close tied in strand in approximately correct position, but ties at eye-bolt and end casting should be temporary until final dressing of overhead.

Spans. Spans should consist of seven-wire strand, and in case of steel poles, should have strain insulator cut in not less than 5 ft. from pole; where pole carries high tension circuits, a strain insulator should be used of suitable strength and creeping surface. Where a foreign line crosses close to the span two strain insulators should be used, one at either side of foreign line, to ensure that if latter falls it shall be on a dead section. Spans on tangents should be close tied into eye-bolts at height above trolley wire not more than one-tenth the distance from track center to pole but in no case should the factor of safety for the span wire be less than two under the conditions to be expected. On wood poles eye-bolts should be at least 12 in. below top of pole, and should be installed at full length, seated against washers. Spans on curves should be installed as on tangents except that pull-over with attached strains should be close tied in strand in approximately correct position and temporarily tied in eye-bolts until the final dressing of overhead. In case of two or more tracks, strand between pull-overs should have temporary tie at one end until final dressing.

Trolley Wire. Trolley wire may be run out by mounting the reel on an arbor on which it can turn freely, and leading the wire to an anchor or to trolley already installed. Tension may be maintained by a brake on rim or side of reel, but under no circumstances should braking be done against the copper. The wire should be pulled to proper sag and temporarily tied to brackets or spans by rope or other soft insulating ties. Particular care should be taken to prevent twisting, kinking or bruising the wire. Parallel faced clamps should be used; chains, cam come-alongs or other short grip devices should not be employed. The sags should be as follows, and in pulling up great care should be taken that the corresponding tensions are not exceeded:

Sag and Tension of Trolley Wire (100 ft. Spans)

Temperature, deg. F.	Size of Wire							
	0		00		000		0000	
	Sag, in.	Tension, lb.	Sag, in.	Tension, lb.	Sag, in.	Tension, lb.	Sag, in.	Tension, lb.
0	2½	1920	3	2020	2¾	2780	2¾	3500
30	3	1600	3½	1730	3¼	2350	3¼	2960
60	3½	1370	4¼	1420	4¼	1800	4¼	2260
90	4½	1070	5½	1100	5¼	1450	5¼	1830
120	6	800	7¾	760	7¼	1050	7¼	1330

The table values are for spans of 100 ft.; for any other span the sag for the tension given in the table is proportional to the squares of the lengths. For example, for span of 50 ft., the sag for a given temperature is equal to 50 squared divided by 100 squared, or one-quarter the corresponding table value for that temperature.

After the trolley wire has been temporarily tied up with the proper sags, and the line has been anchored, the line ears and hangers may be located accurately and attached, clinch ears being thoroughly closed down to give secure grip and smooth running surface, and the mechanical ears well seated in grooved wire, the clamp screws then being slightly upset to prevent backing out.

Trolley Wire Splices. Splices should be of a type and so installed obstruction to the passage of trolley wheel. Grooved wire should be kept in perfect alinement, and if the splice is of the soldering type, it should be thoroughly sweated on without annealing the wire. *The free ends of the wire may be bent back sharply at the* outlet, and cut off forming a hook with end ⅛ in. long.

Trolley Wire Guys. In bracket construction trolley wire guys should be installed at the ends of curves and on long curves and tangents at equal intervals as nearly as possible, but not to exceed 1500 ft. apart. Trolley wire guys should be seven-wire steel strand attached to strain plate supported at a bracket by double pull-over with proper insulation and led both ways to next adjacent poles. Each guy should have a strain insulator cut in it 5 ft. from the strain plate, and should be secured to proper pole by a two turn wrap at height of bracket arm. Where practicable, the strain of these guys should be taken by anchor guys in the line of the pull; if this is impracticable, high guys should be used. Great care should be taken to ensure equal pulls on the guys, and especially that the strain plate is not twisted out of line; the ties should not be made up permanently until the final dressing of the overhead.

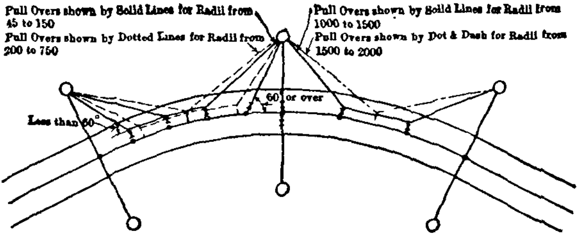


Fig. 9.—Location and arrangement of pull-offs in trolley wire curves.

Curves. (Fig. 9.) Curves should be made up with straight-line clinch ears for round wire, or double clip mechanical ears for grooved wire, attached to suitably insulated pull-over bodies. Support should be by span except where rest of line is in bracket and radius is greater than 300 ft. Between supports curves should be held to line by seven-wire steel strand, with single body pull-offs for single track or for outside one of several tracks, and double body pull-offs elsewhere, spaced as follows:

Radius of curve (feet)	Spacing of pull-offs (feet)	Number of pulls between supports	Distance apart of poles (feet)
40	7	4	35
50	8	4	40
60	9	4	45
70	10	4	50
80	11	4	55
90	12	4	60
100	13	4	65
125	14	4	70
150	15	4	75
200–500	20	3	80
750	25	3	100
1000	33½	2	100
1500 2000	50	1	100
Above 2000	100	0	100

The pull-overs in each span should have bodies and strains held radially to curve by a lacing of seven-wire steel strand at least 6 in. away from trolley wire, which lacing, however, may be omitted from any pull making an angle of 60 deg. or more with the ear to which it is attached. With an odd number of pull-over bodies the middle one should have pull-off strand to each pole. Intersections and complicated special work, particularly in city streets, will usually require special and individual study and treatment. Figures 10 to 16, inclusive, give typical overhead layouts for such special work.

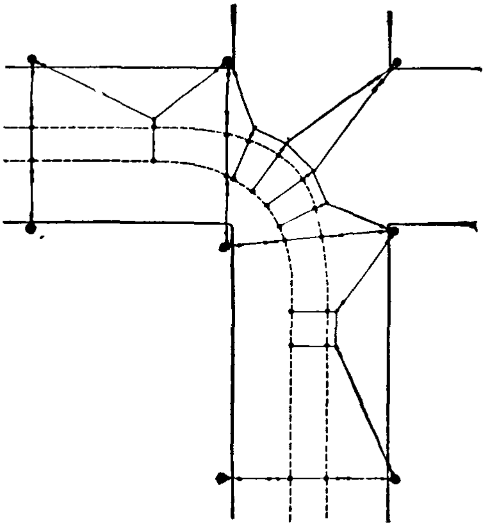


FIG. 10.—Trolley on double track curve.

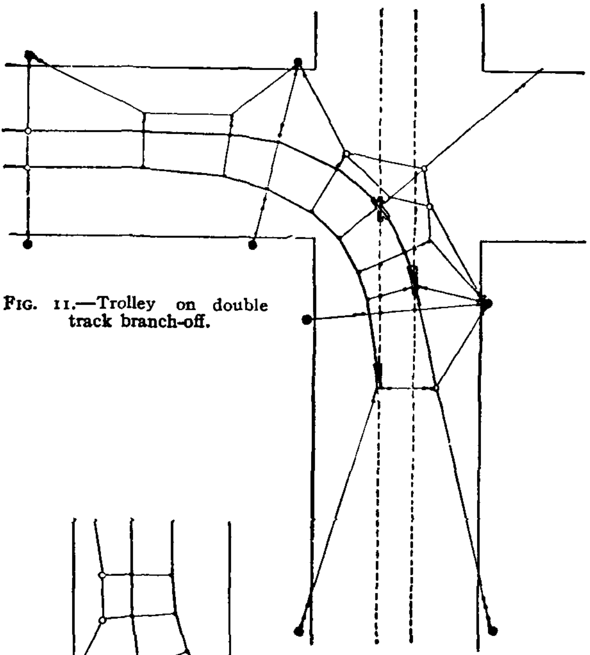


FIG. 11.—Trolley on double track branch-off.

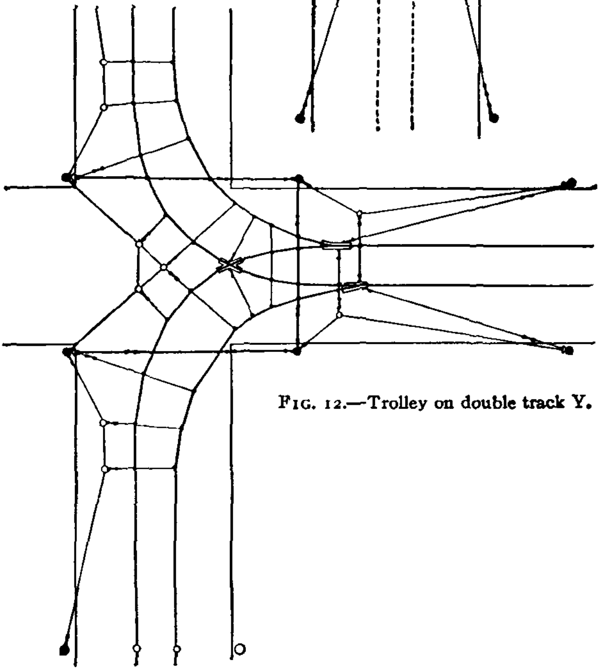


FIG. 12.—Trolley on double track Y.

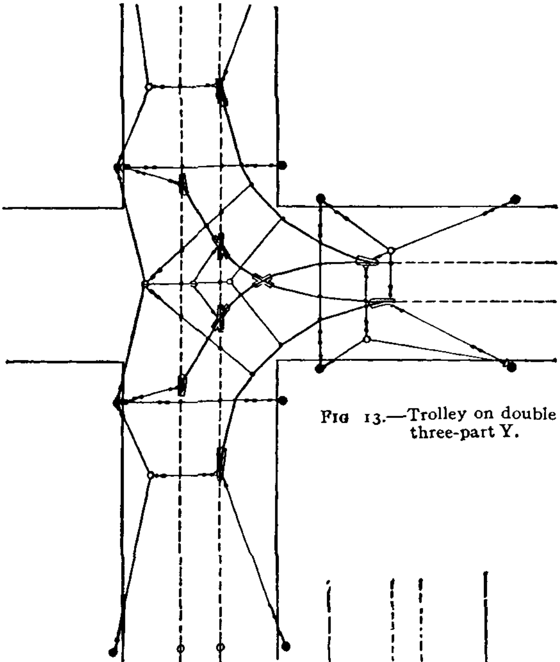


Fig 13.—Trolley on double track three-part Y.

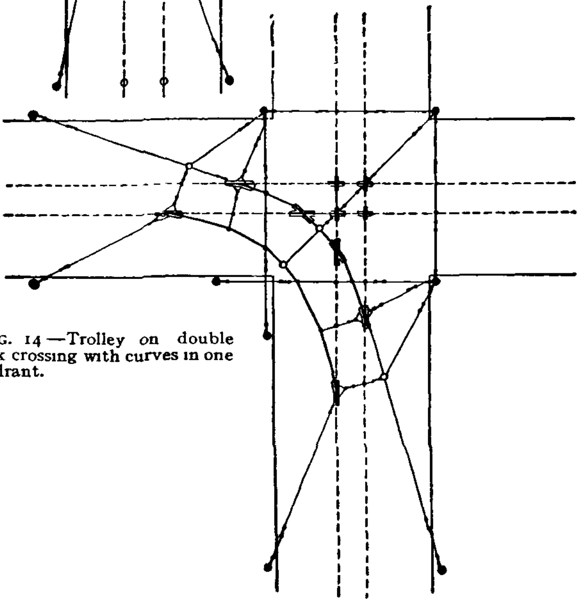


Fig. 14—Trolley on double track crossing with curves in one quadrant.

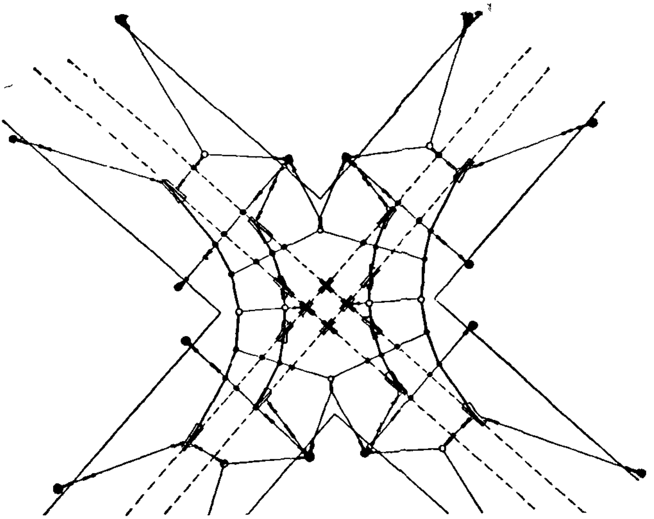


FIG. 15.—Trolley on double track crossing with curves in opposite quadrants.

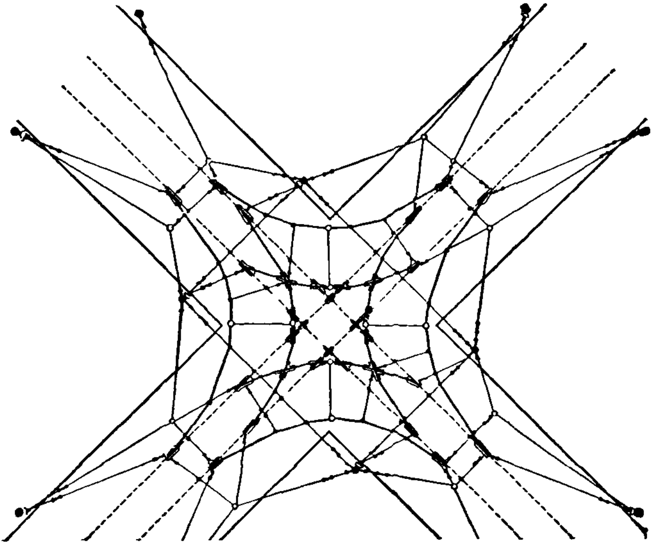


FIG. 16.—Trolley on double track crossing with curves in four quadrants.

Offset of Trolley on Curves. Curves should be dressed with uniform deflection at pull-overs and should offset to the inside of curve an amount given by the expression

$$S = \frac{EH}{G} + R - \sqrt{R^2 + P^2 - Q^2 - L^2}$$

in which

S = radial offset of trolley wire toward center of curve
E = super-elevation of outer rail
H = height of trolley wire above rail
G = track gage
R = radius of curve
P = distance from center of car to pivot of trolley base
Q = distance from center of car to center of truck
L = horizontal distance from pivot of trolley base to point of contact between trolley wheel and trolley wire.

NOTE: All values in terms of feet.

The total offset should be uniformly tapered off from full value at inside easement point of track to no offset at outside easement point. If track is not eased, start with full offset at distance inside end of curve as given below, and run to no offset at point at equal distance outside end of curve.

Radius of curve	Start offset easement
Up to 100 ft.	20 ft. from end of curve
100 to 500 ft.	40 ft. from end of curve
500 to 1000 ft.	60 ft. from end of curve
Above 1000 ft.	100 ft. from end of curve

Frogs. Frogs should be installed with both main line and branch trolley wires led straight through the latter to end 6 ft. beyond frog in eye of strain insulator, to other end of which is close tied a seven-wire steel strand, secured to pole at a level as nearly that of frog as will allow safe clearance over other wires. The frog itself should be held on each side by a guy of seven-wire steel strand with strain insulator attached to frog and secured to proper poles at the point of span attachment. Frogs should be temporarily located on center line of main track and one-third distance from track switch points to track frog point back from track switch points, and if need be should be shifted to suit local conditions and equipment. Until final location is made, trolley wire should be clamped just firmly enough to prevent slipping without bruising or kinking. Care must be taken that frog is not located too far back of track switch points, as such location, while often giving satisfactory running, will result in excessive wear of trolley wire.

Crossings. Crossings wherever practicable should be installed without cutting either of the line wires, which latter should be clamped just firmly enough to prevent slipping without bruising or kinking until the crossing has been saisfactorily located. Where wire must be cut, at least 3 ft. of free end should be left outside clamp until the final adjustment, after which the end should be cut off close to the clamp.

Feed Taps. (Figs. 17 and 18.) Feed taps should be at points of support and should consist of proper size triple braid, weatherproof stranded connection from feeder, feed yoke well soldered on at proper point, and straight-line ear soldered to the trolley wire and

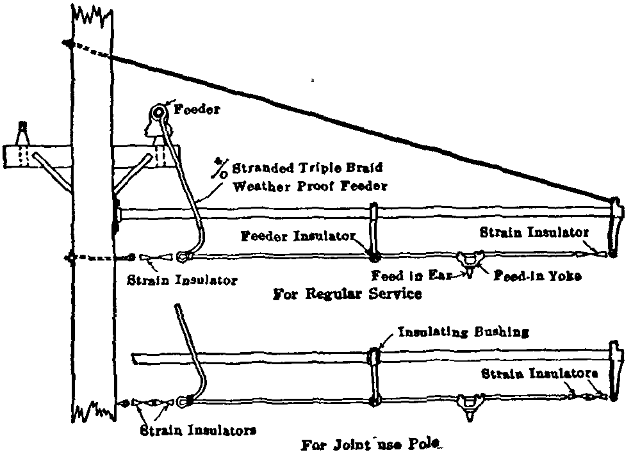


FIG. 17.—Feed tap in bracket construction.

bolted to the yoke. In general, feed taps should be located every 1000 ft. With bracket support the feeder connection should run from the feeder to which it should be well soldered, to strain attached to span eye-bolt in pole, thence, replacing the usual steel

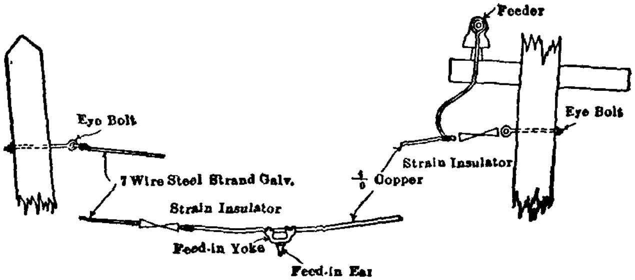


FIG. 18.—Feed tap in span construction.

strand through insulated intermediate casting to strain attached to end casting of bracket. Bracket tube should be of sufficient length to allow at least 8 in. of connection cable between feed yoke and end strain, and intermediate casting should be so located that feed yoke is midway between it and end casting. With span support the *feeder connection should run from feeder to strain attached to span* eye-bolt in pole, thence, replacing the usual steel strand, to close tie in strain 5 ft. beyond trolley farthest from feeder. Span should be completed by seven-wire steel strand close tied into strain and into pole eye-bolt, and should sag not greater than one-tenth the distance from trolley wire to pole.

Feed wire may be run out by mounting the reel on an arbor on which it can turn freely, and if practicable, run along the line on a car or wagon. Where local conditions necessitate pulling feeder on end over the cross-arms, great care must be taken to prevent injury, especially to insulated feeder, rollers or snatch blocks of ample size being used at each arm. Feeder should be strung, for spans of 100 ft., with sags not less than those of the following table:

ALLOWABLE SAGS IN FEEDERS FOR DIFFERENT TEMPERATURES

Temperature, Fahrenheit (degrees)	Copper (in.)	Aluminum (in.)
0	10	5
30	13½	10
60	17	16
90	20½	21½
120	23½	26

The table values are for either bare or weatherproof feeder of any size from 0000 to 2,000,000 circular mils For spans other than 100 ft in length the sag should be in the same ratio to the table values as is the square of the span to 100 squared.

Feeder should be installed in the top groove of the insulator on tangents and in the outer side groove on curves and at angles, and should be tied in with No. 6 soft drawn wire of the same metal as feeder. Top tie (Fig. 19) may be made with two tie wires each 15 in. long. The first wire should be looped around insulator in side

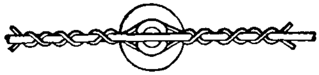


FIG 19.—Top tie for feeder.

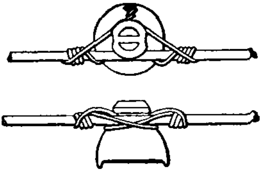


FIG. 20.—Side tie for feeder.

groove, the ends crossed and twisted once under cable, then brought up, one each side and wrapped around cable, crossing above and below until all is on. The other wire should be used to make a similar tie on opposite side of insulator. Side tie (Fig. 20) may be made with one tie wire 15 in. long which should be looped around inside groove at back and ends, carried under cable, then working from center with one end each side, over top, down back, under, up front, and so on for four complete turns, then around back of insulator where ends should be twisted together securely. The cable side of the insulator is front.

On tangents feeder may be carried on single arms having insulators on wood pins; on angles less than 10 deg. by single arms, and on angles greater than 10 deg. by double arms, in either of the latter cases having porcelain insulators on metal pins.

Splices in solid feeder should be made with an approved connector; in stranded feeder should either be made with an approved connector, or of the wrapped cable type, tapered uniformly to size of original feeder at ends. In insulated feeder either type should be smoothly taped to the equivalent covering. The wrapped cable type splice (Fig. 21) is made by stripping the ends to be spliced for 24 in., unlaying and brightening 18 in. of these bare ends, cutting out the core strand of each, and passing wires of one end between those of other end and laying parallel with main cable. The wires of one strand either side of middle of splice are then close served over main portion and rest of wires of that end; a second pair of strands is similarly served on, and so on, until the last pair of strands serve simply on the main portion. The splice should then be sweated full of solder and smoothly taped to equivalent of original insulation

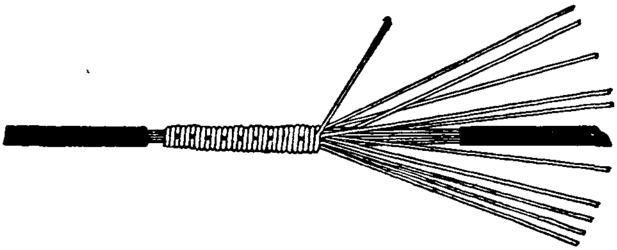


Fig. 21.—Cable type feeder splice.

if cable is insulated. Lead covered or specially insulated or covered cables should be spliced under special instruction, and then only by men experienced in that particular class of work.

Choice of Feeder. Bare feeder should be used on private right-of-way and wherever else practicable; weatherproof feeder where required and where numerous trees or other obstacles would cause grounds if bare cable was used.

Feeder Anchors. Where long spans, long heavy grades, or other conditions cause unusually heavy strains, feeders should be anchored by mechanical clamps secured to eye-bolt in cross-arm through a suitable strain. Insulated feeder should be bared so that clamp takes direct hold on the metal. If heavy pull is anticipated, take guy from cross-arm to next pole.

Section Insulators should be installed at a span suspended from hanger.

Section Switch Box should be bolted to back of pole, using bolts of proper length, each with washer under head and nut, latter being inside box, which should rest on cross-arm carrying feeder sectionalized. The feeder should be dead-ended in strains attached to eye-bolt at proper point in feeder cross-arm and then enter box. If feeder drops from dead end point it should be carried lower than bushings in box and then up to prevent drip from entering.

Automatic Sectionalizing Switch. Fig. 22 shows the circuits in the automatic sectionalizing switch of the General Electric Company. The switch is connected across the section insulator by taps *G* and *H*. Circuit breaker *B* upon closing energizes section *B* and current passes through tap *G*, contactor operating coil *X* to contact stud on relay which is then open circuited. On closing breaker *C*, section *C* is energized and current passes through tap *H* and relay operating coil *W* to ground, closing the relay disk *V*. This in turn completes the circuit through the contactor operating coil *X*, causing the contactor to close. This completes the circuit across the insulator, thus placing the feeders in multiple. The switch will not operate until both breakers feeding the sections to which it is connected are closed. With a heavy load on one section (while the

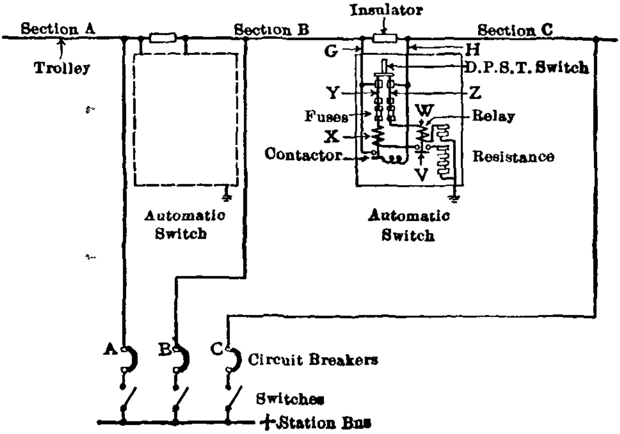


FIG. 22.—Circuits of automatic sectionalizing switch.

automatic switch is in operation), current from the adjacent sections will be fed across the section insulators, thus increasing materially the efficiency of the entire distribution. Where an automatic or hand-operated sectionalizing switch is used, tests should be made periodically to determine any changes necessary in feeder sizes in order to divide the load properly between the feeders.

Lightning Arresters where required by local conditions should be installed at feed taps, just below the feeder cross-arm, and should be connected to the feed tap close to its attachment to the feeder by solid insulated No. 4 copper wire. The ground connection should be of solid No. 4 copper wire, securely fastened to the back of pole and either (*a*) extended as a ground coil, (*b*) well soldered into a pipe ground or (*c*) well soldered to the track rails. In any case at least 8 feet of the lower exposed portion should have a non-metallic protection. Any changes in the direction of ground wire should be made by easy curves.

Line lightning arrester ground wires should be connected to a good earth ground and to the track rail, except:

(1) Where the current flow (see note) on the connection from track to earth would exceed an average of one-fourth ampere during any twenty-four hour period, and

NOTE: In checking up this current flow, the algebraic sum of currents flowing first in one direction and then in the other shall be used in determining the current flow. *It is assumed that a resultant of more than one-fourth ampere average in either direction should be avoided.*

(2) Where *alternating current track circuit block signals of the* double rail type are used, the connection to the track rail should be omitted.

Particular care must be taken to ensure that the ground is effective; unless a good ground is secured the arrester cannot give protection. Earth grounds should be secured as follows: Where permanently moist earth is assured at reasonable depth the ground may consist of a one-half inch pipe driven at least 3 feet into the moist earth. Where there is doubt as to the condition of the soil, excavate. If permanently moist earth is reached, install pipe ground; if otherwise, install a flat coil containing 40 lineal feet of solid No. 4 bare copper wire bedded in not less than 7 cubic feet of charcoal.

Soldering. Where soldering is necessary, it should be done with *non-corrosive paste or with stearin;* the use of acid or corrosive salts should be strictly forbidden; and great care should be taken to prevent overheating and annealing.

Linemen. Only such men should be employed on overhead construction as have had experience and are so skilled in the work that those details which make up good practice will be attended to without the necessity for specific and detailed instruction. Such details include: grading poles to bring tops to approximately the same line; setting cross-arms square to tangent line, and bisecting the angle at breaks; setting brackets square to line and all with same rake; cutting in strains at the same relative points; installing spans so that eye-bolts and strands line up; setting chamfered nuts with flat side to bearing; seating insulator pins firmly to shoulder; seating washers square with bolt hole; screwing in lag screws at least the last half; finishing off all splices, fastenings and ends.

Sag and Horizontal Pull in Span

Single Track Line —The following table shows the sag resulting in spans of various lengths when subjected to the tensions given, a standard span weight of 50 lb being assumed This is a convenient standard weight for a 5/16 in steel strand span which supports a length of 100 ft of 00 trolley with ear and insulator.

SAG IN SPAN WIRE—SINGLE-TRACK LINE

Maximum Sag in Inches (c) for Weight (W) of 50 lb.

Pull on span (P) in pounds	Length of span (S) in feet										
	30	35	40	45	50	55	60	65	70	75	80
500	9 0	10 5	12 0	13 5	15 0	16 5	18 0	19 5	21 0	22 5	24 0
600	7 5	8 8	10 0	11 3	12 5	13 8	15 0	16 3	17 5	18 8	20 0
700	6 4	7 5	8 6	9 6	10 7	11 8	12 9	13 9	15 0	16 1	17 1
800	5 6	6 6	7 5	8 4	9 4	10 3	11 3	12 2	13 1	14 1	15 0
900	5 0	5 8	6 7	7 5	8 3	9 2	10 0	10 8	11 7	12 5	13 3
1000	4 5	5 3	6 0	6 8	7 5	8 3	9 0	9 8	10 5	11 3	12 0
1100	4 1	4 8	5 5	6 1	6 8	7 5	8 2	8 9	9 5	10 2	10 9
1200	3 8	4 4	5 0	5 7	6 3	6 9	7 5	8 2	8 8	9 4	10 0
1300	3 5	4 0	4 6	5 2	5 8	6 3	6 9	7 5	8 1	8 7	9 2
1400	3 2	3 8	4 3	4 8	5 4	5 9	6 4	7 0	7 5	8 1	8 6
1500	3 0	3 5	4 0	4 5	5 0	5 5	6 0	6 5	7 0	7 5	8 0
1600	2 8	3 3	3 8	4 2	4 7	5 2	5 7	6 1	6 6	7 0	7 5
1700	2 6	3 1	3 5	4 0	4 4	4 9	5 3	5 7	6 2	6 6	7 1
1800	2 5	2 9	3 3	3 7	4 2	4 6	5 0	5 4	5 8	6 2	6 7
1900	2 4	2 8	3 2	3 6	4 0	4 3	4 7	5 1	5 5	5 9	6 3
2000	2 3	2 6	3 0	3 4	3 8	4 1	4 5	4 9	5 3	5 6	6 0
2100	2 1	2 5	2 9	3 2	3 6	3 9	4 3	4 7	5 0	5 4	5 7
2200	2 2	2 4	2 7	3 1	3 4	3 8	4 1	4 4	4 8	5 1	5 5
2300	2 0	2 3	2 6	2 9	3 3	3 6	3 9	4 2	4 6	4 9	5 2
2400	1 9	2 2	2 5	2 8	3 1	3 4	3 8	4 1	4 4	4 7	5 0
2500	1 8	2 1	2 4	2 7	3 0	3 3	3 6	3 9	4 2	4 5	4 8

Sags are computed from the formula $c_1 = \frac{3WS}{P}$ in which $W = 50$ for single track lines

Double Track Line. The following table shows the sag resulting in spans of various lengths when subjected to the tensions given, a standard span weight of 100 lb being assumed This is a convenient standard weight for a ⅜-in steel strand span which supports a length of 100 ft of two 00 trolley wires, insulators and ears.

SAG IN SPAN WIRES—DOUBLE TRACK LINE
Maximum Sag in Inches (c) for Weight (W) of 100 lb

Horizontal pull on span (P) in pounds	Length of span (S) feet										
	30	35	40	45	50	55	60	65	70	75	80
50	120 0	150 0	180 0	210 0	240 0	270 0	300 0	330 0	360 0	390 0	420 0
100	60 0	75 0	90 0	105 0	120 0	135 0	150 0	165 0	180 0	195 0	210 0
150	40 0	50 0	60 0	70 0	80 0	90 0	100 0	110 0	120 0	130 0	140 0
200	30 0	37 5	45 0	52 5	60 0	67 5	75 0	82 5	90 0	97 5	-05 0
250	24 0	30 0	36 0	42 0	48 0	54 0	60 0	66 0	72 0	78 0	84 0
300	20 0	25 0	30 0	35 0	40 0	45 0	50 0	55 0	60 0	65 0	70 0
400	15 0	18 8	22 5	26 3	30 0	33 8	37 5	41 3	45 0	48 8	52 5
500	12 0	15 0	18 0	21 0	24 0	27 0	30 0	33 0	36 0	39 0	42 0
600	10 0	12 5	15 0	17 5	20 0	22 5	25 0	27 5	30 0	32 5	35 0
700	8 6	10 7	12 9	15 0	17 1	19 3	21 4	23 6	25 7	27 9	30 0
800	7 5	9 4	11 3	13 1	15 0	16 9	18 8	20 6	22 5	24 4	26 3
900	6 7	8 3	10 0	11 7	13 3	15 0	16 7	18 3	20 0	21 7	23 3
1000	6 0	7 5	9 0	10 5	12 0	13 5	15 0	16 5	18 0	19 5	21 0
1100	5 5	6 8	8 2	9 6	10 9	12 3	13 6	15 0	16 3	17 7	19 1
1200	5 0	6 3	7 5	8 8	10 0	11 3	12 5	13 8	15 0	16 3	17 5
1300	4 6	5 8	6 9	8 1	9 3	10 4	11 5	12 7	13 8	15 0	16 2
1400	4 3	5 4	6 4	7 4	8 6	9 6	10 7	11 8	12 9	13 9	15 0
1500	4 0	5 0	6 0	7 0	8 0	9 0	10 0	11 0	12 0	13 0	14 0
1600	3 8	4 7	5 6	6 6	7 5	8 4	9 4	10 3	11 3	12 2	13 1
1700	3 5	4 4	5 3	6 2	7 1	7 9	8 8	9 7	10 6	11 5	12 4
1800	3 3	4 2	5 0	5 8	6 7	7 5	8 3	9 2	10 0	10 8	11 7
1900	3 2	4 0	4 7	5 5	6 3	7 1	7 9	8 7	9 5	10 3	11 1
2000	3 0	3 8	4 5	5 3	6 0	6 8	7 5	8 3	9 0	9 8	10 5
2100	2 9	3 6	4 3	5 0	5 7	6 4	7 1	7 9	8 6	9 3	10 0
2200	2 7	3 4	4 1	4 8	5 5	6 1	6 8	7 5	8 2	8 9	9 5
2300	2 6	3 3	3 9	4 6	5 2	5 9	6 5	7 2	7 8	8 5	9 1
2400	2 5	3 1	3 8	4 4	5 0	5 6	6 3	6 9	7 5	8 1	8 8
2500	2 4	3 0	3 6	4 2	4 8	5 4	6 0	6 6	7 2	7 8	8 4

Sags computed from the formula $c = \frac{3W(S-a)}{P}$ in which $W = 100$ and $a = 10$ for double track.

The values in the above table are plotted in Figs 23 and 24 For other sizes of wire strand or trolley the weight of the span will be different The table on page 573 shows the "loading factor" or ratio of weight of various spans to the weight of the assumed standard span of 100 lb The sag given in the above table, when multiplied by the proper "loading factor," will give the sag in any desired span.

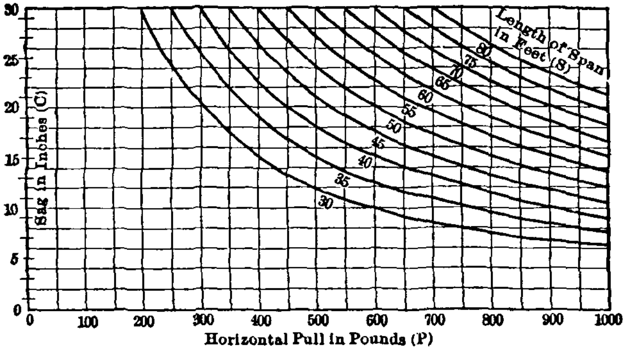


FIG. 23.—Sag and horizontal pull of double track span wires. (See also Fig. 24.)

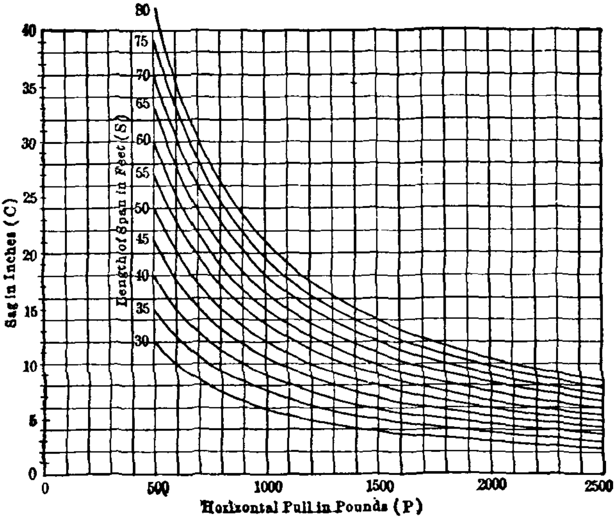


FIG. 24.—Sag and horizontal pull of double track span wires. (See also Fig. 23.)

Loading Factors for Calculating Sags in Double Trolley Span Wires

These factors are to be used as multipliers of the sags for standard loading given in the Table on page 571 to determine the sag for any other loading

Span wire	1/4-in.				5/16-in.				3/8-in.				7/16-in.			
Safe working stress in pounds	1150				1900				2500				3250			
Trolley wire	0	00	000	0000	0	00	000	0000	0	00	000	0000	0	00	000	0000
30 ft. length of span	0.85	1.01	1.22	1.49	0.87	1.04	1.25	1.51	0.90	1.06	1.25	1.54	0.93	1.10	1.31	1.60
35 ft.	0.85	1.02	1.23	1.49	0.88	1.05	1.26	1.52	0.91	1.08	1.27	1.55	0.96	1.12	1.33	1.62
40 ft.	0.86	1.03	1.24	1.50	0.89	1.06	1.27	1.53	0.93	1.09	1.28	1.57	0.98	1.14	1.35	1.64
45 ft.	0.87	1.03	1.24	1.51	0.90	1.07	1.27	1.55	0.94	1.11	1.30	1.58	1.00	1.16	1.37	1.66
50 ft.	0.87	1.04	1.25	1.51	0.91	1.08	1.28	1.56	0.96	1.12	1.31	1.60	1.02	1.18	1.39	1.68
55 ft.	0.88	1.04	1.25	1.52	0.93	1.09	1.29	1.57	0.97	1.14	1.33	1.61	1.04	1.20	1.41	1.70
60 ft.	0.89	1.05	1.26	1.52	0.94	1.10	1.30	1.58	0.99	1.15	1.34	1.63	1.06	1.23	1.43	1.72
65 ft.	0.89	1.06	1.27	1.53	0.95	1.11	1.31	1.59	1.00	1.17	1.36	1.64	1.08	1.25	1.46	1.74
70 ft	0.90	1.06	1.27	1.54	0.96	1.12	1.32	1.60	1.02	1.18	1.37	1.66	1.10	1.27	1.48	1.76
75 ft	0.90	1.07	1.28	1.54	0.97	1.13	1.33	1.61	1.03	1.20	1.39	1.67	1.12	1.29	1.50	1.78
80 ft.	0.91	1.08	1.29	1.55	0.98	1.14	1.33	1.62	1.06	1.21	1.40	1.69	1.14	1.31	1.52	1.80

In determining the safe working stress, a factor of safety of 2 was used.

The factors multiplied by 100 give the approximate total weight of the span and two trolley wires.

The following material makes up the span: 4 strain insulators; 2 straight-line hangers; 2 trolley ears; seven-strand galvanized iron wire.

Weights of Material

Straight-line hangers	3.25 lb. each
Ears	0.8 lb. each
1/4-in. galvanized wire, 7-strand	0.125 lb. per ft.
5/16-in. galvanized wire, 7-strand	0.21 lb. per ft.
3/8-in. galvanized wire, 7-strand	0.295 lb. per ft.
7/16-in. galvanized wire, 7-strand	0.415 lb. per ft.
0000 trolley wire per ft.	0.6398 lb.
000 trolley wire per ft.	0.5073 lb.
00 trolley wire per ft.	0.4024 lb.
0 trolley wire per ft.	0.3194 lb.
Strain insulators	2.25 lb. each

Catenary Construction

Endeavor to reduce the sag in overhead contact conductor by supplying several points of support in a given span resulted in the development of the catenary type of suspension. In its elementary form it consists in suspending the contact conductor by means of clips or hangers which are placed a few feet apart and which are hung from one or more messenger wires which hang nearly in a catenary curve between the main supports of the overhead construction or between intermediate supports suspended therefrom. There are a great many varieties of this general form of suspension, varying from the simple catenary from which the trolley wire is suspended directly, to very elaborate suspension systems. The purpose of the catenary construction is to support the contact conductor at points close together while maintaining a great distance between points of support of the whole system, thus reducing changes in height of the contact conductor to a practical minimum without introducing excessive tension in the suspension system. The particular need for such a working conductor system is found in high speed work. At the present stage of development the advantages which might be derived from the use of the catenary type of suspension in city work, except in special cases such as over railroad crossings, are in general considered to be outweighed by the complications in its application and operation. There are three general types of primary suspension, namely: (1) bracket, (2) cross span and (3) bridge. Figs. 25 and 26 show typical examples of working conductor suspension in heavy electric railway work.

Length of Span. Messenger spans for catenary construction are usually 150 ft. with wood pole support, and 300 ft. on steel bridges; hanger spacing is generally 15 ft. for pantograph operation, and 30 ft. for wheel operation.

General Details of Suspension. The earliest catenary consisted of a messenger from which the contact wire was suspended by a series of equidistant hangers. This worked well for short spans, but on long spans the great difference in length, and so in weight, of the mid and end hangers led to the development of compound catenary. The Siemens-Schukert type, one of the earliest forms, employs a main messenger, a secondary messenger suspended from the main messenger by hangers approximately 20 ft. apart, and a contact wire suspended from the secondary messenger by hangers approximately 10 ft. apart. The secondary messenger supports come at the quarter points between the main hangers or "droppers," which are of wire, about No. 8 gage equivalent, and are in two parts, linked eyes in the adjacent ends forming a flexible joint; these hangers attach rigidly to both main and secondary messenger. The secondary hangers attach rigidly to the contact wire, but loop over the secondary messenger so they are free to rise 2 or 3 in. This has proved very satisfactory in a number of Continental installations, and on the Midland Railway of England.

Multiple Catenary. Two main messengers have vertical sag and horizontal deflection toward each other. The hangers, spaced 10 ft. apart, are equilateral triangles of pipe, the two upper corners attaching to the messengers while the bottom one holds the contact wire; the entire system is at line potential, the supports at the bridges being insulators as well. As first installed on the New Haven road, the contact wire, No. 0000 grooved copper, was carried directly by the triangles. These points, however, were exceedingly rigid, while the midspan was comparatively soft, and when the speed of the train was such that the time interval between hanger points was the same as the vibration period of the collector, the latter chattered with increasing violence until changed conditions threw the pantograph "out of step."

Secondary Contact Wire. The trouble with the multiple catenary cited above was successfully obviated by the following improvement by E. H. McHenry: A steel contact wire of the same section as the copper is suspended from the latter by clips midway between the hangers, which keep the wires 1¾ in. apart, center to center. The clips are rigidly attached to both wires; it was feared that with the heavy shoe pressure a looped clip such as is used in the Siemens-Schukert form might permit the lower wire to turn sideways, in which case the shoe would foul the clips with disastrous results. The London-Brighton & South Coast Railway, to secure the same end, uses hangers of comparatively light wire, the small ones holding the wire by a loop, while the larger ones have jointed sides to give the desired flexibility. On this system the stresses are all low, the messenger having a sag of 6 ft. in vertical projection for a span of not quite 200 ft.

Twin Trolley. Twin trolley construction consisting of two trolley wires hung side by side and alternately suspended from the same messenger was first used on the Chicago, Milwaukee, and St. Paul electrification. On account of the fact that there are two trolley wires the current carrying capacity of the combination is greater than with a single trolley wire and on account of the uniform flexibility of the arrangement there are no hard spots causing the collector to leave the wire and cause arcing. This type of construction is suitable for both high speed operation and heavy current collection with pantograph collector.

McHenry-Murray Suspension. The extensive experience with the pioneer and McHenry forms under heavy service, and careful study of the behavior of other types, resulted in the McHenry-Murray catenary employed on the extension of the New Haven electrification. In this the main messengers, one for each track, have at the quarter points structural steel cross-bents which in turn carry insulators from which the secondary system is hung. This latter consists of the secondary messenger, carrying, through hangers 10 ft. apart, a pair of wires, upper of copper and lower of steel, held 1¾ in. center to center by clips which are midway between the secondary hangers. The main messengers have only to carry the mechanical load of the secondary system. The value of this grounded shield as a lightning protection has been strikingly shown in several electrical storms which caused trouble on the older forms, while the new construction in the same territory was unaffected.

Catenary Hangers. A great variety of hangers has been tried, but very few of the forms have persisted, the majority either tak-

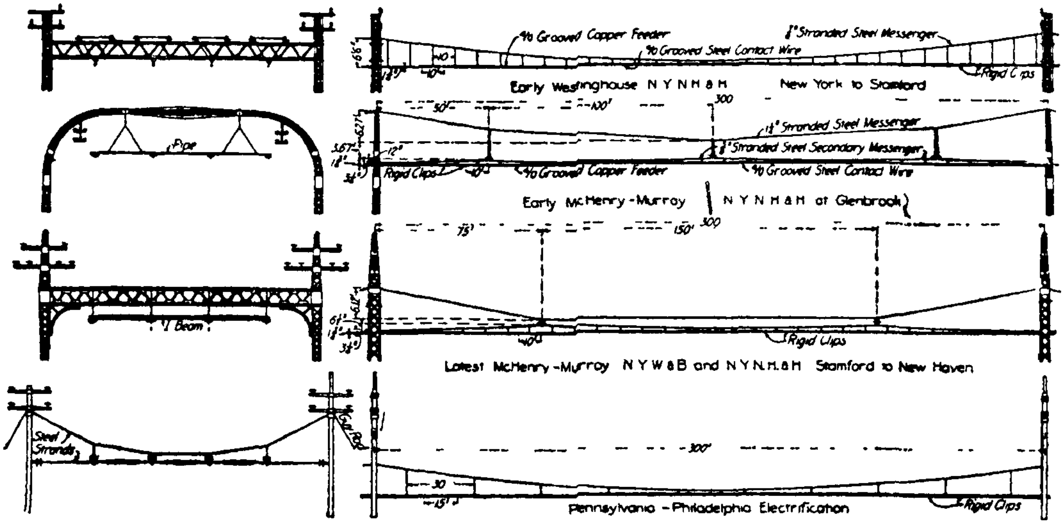


FIG. 25—Examples of typical catenary construction in America.

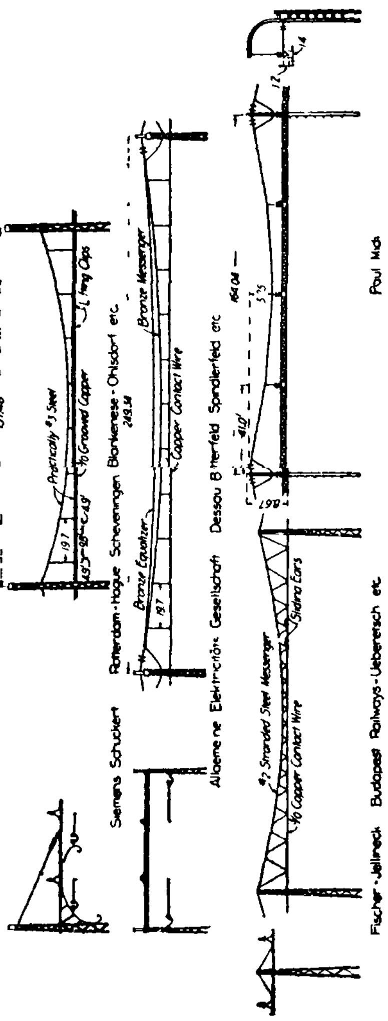


FIG. 26.—Examples of typical catenary construction in Europe

ing too much time to install or shaking loose in service, not a few doing both. That which seems to meet conditions best is a mechanical three-screw clip rigidly attached to a steel strap, the top of which is formed into a loop 3 in. or 4 in. long, permitting the contact wire to rise to that extent. The other types with cam, screw, toggle, wedge and other locking devices, many of them exceedingly ingenious, are for the most part of interest only as a matter of history. Early forms of strap hangers were hinged at the lower end, but this permitted the loop to tip over, and with each lift to crawl further out on the messenger until all the lift was taken up and the flexibility lost. To facilitate installation several ingenious ways of making the loop have been devised. In one the end is twisted back of the main stem, leaving a space of such shape that the hanger screws onto the messenger; another snaps into place, the spring of the shaft normally keeping the loop closed; while a third, also of the snap type, has the tail of the strap so bent up inside the loop that, once snapped on, the messenger cannot get into a position to wedge its way out.

Catenary Brackets. Brackets are of two classes: those made up of paired angles or channels, separated at one end to go either side of the pole, to which they fasten by a through-bolt, or to clasp a casting lagged or bolted to the face of the pole; and those consisting of a single I or T seating in a suitable pole casting. Paired member brackets offer facilities for the attachment of supports, insulators and the like; single member forms offer much less opportunity for corrosion. Brackets which clasp the pole and are through-bolted to it restrict the length of line affected by a break to a few spans either side, but the brackets at the break are usually badly crippled if not completely wrecked; brackets less rigidly secured swing clear, and while dropping a much greater length of line are usually injured little if any. Abroad, brackets on special iron poles are used for spans up to 300 ft., and frequently for double track work, either with center pole construction or carrying both trolley wires on the one long arm. In American practice two-track brackets are not so frequently used, spans or bridges being employed instead; the latter are also extensively employed on long-span foreign lines.

Cross Span Catenary. Cross span support has the marked advantage of causing minimum interference with the view of the motorman, but the sag must be great or the stresses will be very high. It has been used in some of the early light cantenary construction and the New York, New Haven & Hartford Railroad has made extensive use of this construction for yard work, and to a small extent for the main line, employing heavy lattice poles to sustain it. The span wire attaches to the main catenary messenger, while a steady strand attached at or near the contact wire prevents overturning under pantograph pressure. In some instances the span has contained a section of structural steel carrying insulators to which the messengers attach. The New Haven type has long wooden strains in the steady strand, very materially stiffening it.

Bridge Catenary. For heavy work most engineers have preferred a stiff cross member to the high or heavy poles required for cross span support. Bridges have a wide range of design, from the light paired angle irons bolted to wooden poles, one on each side of the track, as on the Midland Railway, England, and certain Continental lines, and the light but rigid entirely structural material frames of the Archbold-Brady type, to the very substantial structures of the early New Haven work. The importance of the traffic involved in the latter case and the lack of data as to behavior in such service justified the use of a conservative design in the pioneer installation.

Alinement of Contact Wire. Abroad the line is zigzagged from side to side, usually about 1 ft. each way, to distribute the wear over the top of the shoe. In America the line is almost always centered, probably for esthetic reasons, although it is found that the natural side sway of the pantograph is ample to give all necessary side travel. Even with center location it is essential that track and overhead departments maintain close cooperation lest a change in super-elevation without a corresponding line shift result in the shoe being thrown entirely clear of the contact wire because of unavoidable side play.

Turnout Diverters. At turnouts the diverging wire is lifted above the level of that of the main line, but for pantograph operation early American catenaries used a gridiron of wires to fill in the space between wires to a point where the end of the shoe could by no chance go over the branch wire. Later designs follow the foreign practice of merely raising the branch wire, which, however, is rigidly held to that elevation throughout the fouling space by special double hangers.

Catenary Curve Dressing. Curve dressing is complicated, particularly on flexible simple catenary. To prevent tipping there must be two attachments: if made to the hanger rod the friction on the messenger absorbs much of the flexibility; if one attachment is made to the hanger rod or contact wire and the other to the messenger the pull-together effect of paired strands is equally undesirable. The best solution is the use of a spreader which keeps the two parts parallel for at least 6 ft. or 8 ft. from the catenary, beyond which they come together in a common pull-off. On rigid hanger lines this is unnecessary and the strands are attached to the messenger and to the bottom of the hanger rod or to an attachment to the contact wire which, for pantograph work, is sufficiently offset to prevent fouling. Sharp curves with several pulls per span employ either a bridle between supports to which the pulls are run radially, or else the pulls are run to the supports in fans; light curves use either the bridle or fan method, or on heavy construction use a special pull-off pole. On light work flat curves offer so little side pull that the weight of the strands and insulator causes heavy sag; in one instance this was partly obviated by using a light single strand, which a short distance from the catenary was divided into two groups of three wires each, the seventh being served on to prevent further unlaying. One group of three wires was then taken to the messenger, the other to the trolley, while the insulator was cut in at the bridle. Foreign lines and a few American employ the rigid steady brace attached to the main support, and rarely to a pole specially set for it. The elimination of special pull-off poles is secured, and the disadvantage of the line loading by the steady brace is obviated by extending the bracket to form a hook to which a pull off strand is attached. In case the pole is on the outside of the curve the attachment is made directly to it. The use of cross-bents on the Harlem Branch electrification of the New Haven permitted the use of a bridle tied to these bents, pulling them partly to the desired offset from the chord of the messenger, a further shift of the insulators on the cross-bents bringing the contact wire over the track center. The most admirable device, however, is the Murray curve hanger, the head of which firmly grips the messenger, holding the shank at an angle of approximately 45 deg. from the vertical, while the lower end is a duplex clip, which is vertical. The lengths are such that the clips are true to line, making the chords but 10 ft., while the inclined shank permits the contact wire to yield to the pressure of the shoe against the torsion of the messsenger and the pull of the contact wire In connection with shortened support spacing to keep the hanger lengths within reasonable limits, this scheme, employed on the recent New Haven work and now used on the Pennsylvania, gives an almost perfectly true alinement.

Temperature Changes. Changes in temperature tend to cause corresponding length changes in the various members of catenary construction, but the actual result is largely affected by the character of the construction. The messenger, contracting or expanding, tends to lower or rise, the movement being a maximum at mid-span, and zero at supports, the contact wire tends to do likewise, but at a different rate because of its different sag. The hangers tie the two together and compel equal movement, giving to the contact wire in cold weather an inverted sag, so to speak, and in hot weather slack wire between the hangers. On the Syracuse, Lake Shore & Northern Railway, the contact wire, which was erected in winter, was given a very high initial tension and an inverted sag of 1 ft. to the standard 300-ft. span, so that the heavy summer traffic should have a fairly tight straight line. With this one exception, which, it is said, did not entirely meet expectations in this respect, American engineers have considered compensation devices not worth the complication, a belief well justified by the behavior of the lines which are in service without such devices.

Messenger Wire Design. The Amer. Elec. Ry. Eng. Assn. recommends a factor of safety of 4 between working and ultimate tension of messenger wire, and that the messenger be pulled up to such a tension when unloaded (measured by dynamometer) that when the load of trolley and hangers is added the trolley wire will hang in a straight line. Charts prepared by the Committee on Power Distribution show the relation between sag and tension in loaded and unloaded messenger, together with temperature correction. Figs 27, 28 and 29 reproduce such charts for steel messenger with 180 and 300 ft. spans and for copper messenger with 300 ft. spans, respectively. The Manual of the Am. El. Ry. Eng. Assn. shows similar charts for steel messenger with 150, 210, 240 and 270 ft. spans.

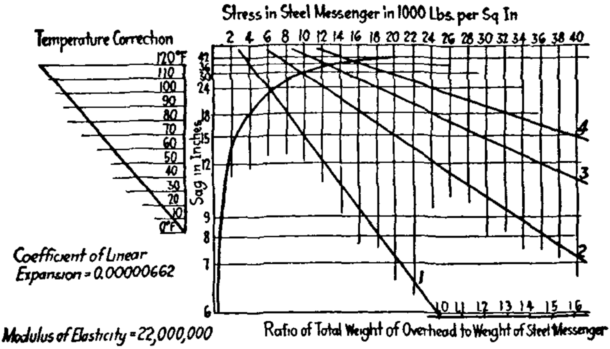


FIG 27.—Steel messenger sag tension curves for 180 ft. spans.

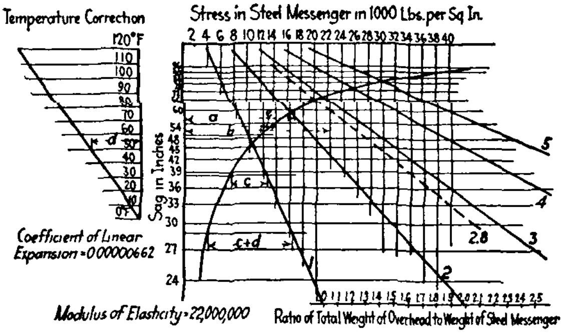


FIG. 28.—Steel messenger sag tension curves for 300 ft. spans.

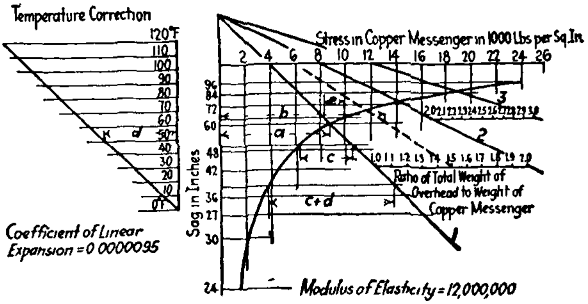


FIG. 29.—Copper messenger sag tension curves for 300 ft. spans.

On these charts the true length of a parabolic curve of various sags is expressed by the horizontal ordinate from the line of origin to the curved line. For instance, on the 300-ft. span diagram (Fig. 28) the true length of a parabola of 60 in. sag is represented by the distance $a + 300$ ft. The stretch of a steel messenger under its own weight is represented by a similar distance measured to the straight line marked 1 on the diagram. The stretch under multiple loads is marked by the lines 2, 3, etc., and intermediate ratios may be obtained by interpolation. For instance, the stretch of a steel messenger loaded with 2.8 times its own weight would be indicated by the distance b measured from the origin to a line interpolated on the diagram. Subtracting, $a - b = c$, then $c + 300$ ft. equals the true unstressed length of the strand necessary to obtain the given sag. In this instance for a final sag of 60 in. c will have a minus sign. Marking the quantity c on the edge of a card and keeping the card edge horizontal, move it along the length curve until it fits between this curve and the line of unit stretch, and read the required sag at the left, in this case 38½ in. This means that a steel messenger which will be ultimately loaded with 2.8 times its own weight should be strung to a sag of 38½ in. in order to have a final sag of 60 in. under full load. This would be the case with a 7/16 in. steel messenger weighing 0.415 lb. per foot to support a 4/0 trolley weighing 0.641 lb. and hangers weighing 0.1 lb. per foot of span. The total of these weights is 1.156 lb. per foot and the ratio of 1.156 to 0.415 equals 2.8. The stress per square inch in messenger may be read in the upper line of figures. In the case shown, the stress would be 22,000 lb. per square inch when loaded. This would represent a pull of 2640 lb. on the 0.1204 sq. in. area of a 7/16 in. steel cable.

Temperature effects may be obtained from the triangular scale at the left of the chart, and the quantity shown should be added to or subtracted from the quantity c. Let it be assumed that the messenger is to be sagged when the temperature is 50 deg. below normal. Measure the distance d on the temperature diagram at the 50 deg. point. As we are considering a lower temperature, the wire is shortened, in other words d is a minus quantity. As c is also minus, c and d may be added graphically and the marker moved along the length curve until it fits between this curve and the line representing stretch, and the required sag is read at the left, in this case approximately 29 in. If the stringing temperature were 50 deg. above the standard, the quantity d would have a plus sign and c, being minus, would be subtracted from it. This may be done graphically, the result being e which has a plus sign. Where the sign is plus the scale must be set above the intersection of the curve and the line expressing stretch. In a similar manner the effects of temperature on other spans may be studied.

On the diagram for the copper messenger on 300-ft. span (Fig. 29) there is indicated a similar calculation assuming a 500,000 c.m. messenger, a 4/0 trolley and hangers weighing 0.1 lb. per foot of span, using the same letters to indicate similar quantities. In this case the ratio of total weight to weight of messenger would be 1.5. If the normal sag in this case were to be 60 in. when loaded, for a stringing temperature 50 degrees below normal the messenger should be pulled to a sag of about 38 in. and for a temperature 50 deg. above normal a sag of 63 in.

In a similar manner the effects of ice load may be studied, expressing the total load as a ratio to the weight of the messenger cable and interpolating a line on the diagram to represent this ratio.

Catenary Hanger Spacing. The Amer. Elec. Ry. Assn. recommends that hangers be spaced 15 ft. apart where pantograph trolleys are used; 30 ft. spacing may be used where pole trolleys are used.

Catenary Hanger Length. Hanger lengths will depend on the length of span and the minimum distance established between lowest point in messenger and contact wire, and may be determined from the formula:

$$h = \frac{4dx^2}{S^2} + a$$

Where h = length of hanger, inches
S = length of span, feet
d = maximum sag in center of span, inches
x = distance from center of span to hanger, feet
a = fixed distance between trolley wire and the lowest point of sag of messenger, inches.

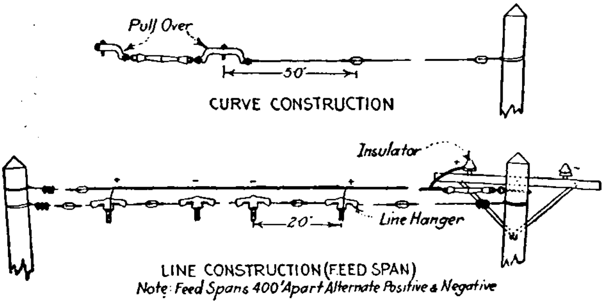


FIG. 30.—Types of span using standard line material for trackless trolley, used with swivel harp, wheel collector.

Overhead for Trackless Trolley. The construction of the overhead working conductors for the trackless trolley bus usually is quite similar to the standard 600 volt overhead trolley construction (page 550 et seq.), except that there are two parallel trolley wires, one positive and the other negative. The two trolley wires generally are carried 24 to 30 in. apart, with a strain insulator cut into the span wire between the hangers, although in one installation (Staten Island) the spacing is as low as 14 in. on straight line and 15 in. on curves, the two trolleys being carried by a single hanger, with porcelain insulators around the stud bolts. In an installation at Baltimore there are two pairs of 2/0 round copper trolley wires, the positive and negative wires of each pair being spaced 24 in. apart. The trolley wire is hung from a span by the standard straight-line hanger and clincher ear with porcelain strain insulators cut in the span between each positive and negative wire and also between each outside wire and the pole. Two 4/0 feeders, one positive and the other negative, are run on cross-arms about two-thirds of the total distance and each pair of trolley wires is tapped to the corresponding feeder at intervals of approximately 400 ft., that is, each wire is tapped to the feeder of the same polarity every 800 ft. For the purpose of turning the busses, loops have been built at each end of the line, using wood strain insulators and standard pull-overs. The wood strain insulators are of proper length to give the necessary clearance between the conductors. Fig. 30 shows both a straight-line and a curve-suspension span.

On some of the installations of trackless trolley line it has been impossible to turn the busses on a loop and it has been necessary to build a wye. The standard frogs and cross-overs are not designed so that a trolley wheel mounted in a swivel harp will follow in the desired direction and it has been necessary to put guides on the frogs and cross-overs so that the trolley wheel will operate properly; this difficulty also has been overcome by the use of pins or springs.

Heavy Electric Traction

There are three systems of power distribution which may be considered for heavy electric traction or the electrification of main line railroads: (1) direct current, (2) single phase alternating current, and (3) three phase alternating current. These are described in the Standard Handbook for Electrical Engineers as follows:

Direct current electrification consists of direct current motor equipment fed from overhead trolley or third rail which receives power from substations containing rotary converters or motor-generator sets. It is the system in almost universal use in city and interurban electric railways, but varies as to the voltage of the direct current supply. As a result of perfecting the commutating or interpole motor, it has been possible to wind motors for higher voltages, and 1500-volt direct current motors are now in commercial operation. These motors are connected two in series when fed from a 3000-volt trolley. Still higher voltage equipments are entirely possible, as indicated by factory experimental apparatus built and tested, and every requirement of the heaviest main line service can be provided for at moderate cost of locomotives and distribution systems. The chief disadvantage of the direct current system has been the relatively high cost of feeder copper and substations and their low efficiency, but this has been largely overcome by the adoption of higher voltages and the automatic substation. The advantages of this system include simplicity, low cost and high efficiency of the locomotives, the option of trolley or third rail distribution, the possible use of bi-polar gearless motor construction, and the use of balanced three phase power supply of any frequency and with no serious telephone or telegraph interference. Large direct current locomotives are in service upon the following main line electrifications:

P. R. R. New York Terminal	600 volts
B. & O. Tunnel Railway	600 volts
New York Central Railway	600 volts
Detroit Tunnel Railway	600 volts
Butte, Anaconda & Pacific Railway	2,400 volts
Canadian Northern Railway	2,400 volts
Chicago, Milwaukee & St. Paul Railway	3,000 volts

Single Phase. The use of a single phase alternating current permits the use of very high voltages on the trolley—as high in fact as there is any reason for using. In this country the voltages vary from 3300 to 6600 which are used on interurban lines and some of the earlier main line electrifications, to 11,000 which is now accepted as the standard for main lines. In Europe voltages as high as 15,000 are in use, but it is felt in this country that this is higher than necessary. With the exception of one small installation, 25-cycle current is used for all single phase railways in this country. In Europe, however, the preference is given to 15 or 16⅔ cycles. A frequency of 25 cycles has the advantage of being a standard for power distribution in many localities, and in lower cost of power station and substation apparatus. It has the disadvantage of greater size and weight of commutator type motors, and thus limits the output from a given space on a locomotive. On short single phase lines the energy may be fed directly from the generators to the trolley, but for long lines, it is necessary to use a high voltage for transmission and feed the trolley from step down transformers. These transformers are usually located in substations but are sometimes of the outdoor type. In either case regular attendants are not required.

Two types of equipment are at present operated from single phase trolleys: (1) Cars and locomotives having commutator type motors with series characteristics; and (2) split phase locomotives having induction motors which derive the additional phases from a phase converter in the locomotive cab. The phase converter consists of a polyphase stator and a squirrel cage rotor, the combination being in effect an induction motor fed from single phase supply and getting its polyphase field by means of its rotating armature. Both types employ step down transformers on the locomotive so that the motors have low voltage applied to them. The former type is particularly suited for passenger service with both multiple trains and locomotives. It is also well suited for high speed freight and switching service. In such service the variable speed characteristics are valuable. The split phase locomotive is particularly desirable where heavy grades are encountered which necessitate high tractive efforts and where the automatic regeneration of the motors on down grades saves power and decreases the likelihood of accidents.

The disadvantages attending the use of single phase current are in the relatively high cost and low efficiency of motive power, the cost of supplying single phase power and the inductive interference of the single phase trolley with neighboring telephone and telegraph circuits. The latter has been largely overcome by the installation of series track transformers, the primary of which carries the trolley current and the secondary the return current in the track. The great advantages of the single phase system of distribution lie in the high efficiency of the substations and trolley and the low cost of the distributing system. Large single phase locomotives with commutator type motors are in service on the following main line electrifications: New York, New Haven & Hartford Railway, Grand Trunk Ry., St. Clair Tunnel, Boston and Maine Railway—Hoosac Tunnel, Spokane and Inland Empire Railway. This system has also been adopted for the New York, Westchester & Boston Ry., and the Pennsylvania Railway, both using multiple unit trains, the former for New York and the latter for Philadelphia suburban service. The Norfolk and Western Ry. is using split phase locomotives with three phase motors for heavy mountain grade service. Liquid rheostats are used in starting these locomotives. The Pennsylvania R. R. has also built one split phase locomotive, with which it has been experimenting.

Auto-transformer Scheme of Single Phase Distribution. In the auto-transformer scheme (also known as the semi-balanced system) of single phase distribution developed on the New York, New Haven and Hartford R. R., the contact conductor of the 11,000-volt catenary system is connected to one side of a 22,000-volt single phase line, auto-transformers distributed along the track are connected across this 22,000-volt line, and the middle point of each of these auto-transformers is grounded to the track rails. Thus this system is similar to a three wire system except that the direct load is on one side of the circuit, the other side of the circuit receiving its share of the load through auto-transformers which act as balancers. By this system the operating economies of a contact conductor voltage of 11,000 and a transmission voltage of 22,000 are combined, and this has been done with the elimination of the disturbance which the ordinary single phase system, having one side grounded and the other side connected to the contact conductor, caused in adjacent telephone and telegraph circuits.

Three Phase. The three phase system of distribution is not used to any extent in this country, chiefly on account of the difficulties arising from the double overhead trolley which complicates the yards and cross-overs. It is used extensively in Italy, and the three phase locomotives in use there are the lightest electric locomotives of their capacity in the world. Three phase locomotives are usually used with about 3300 volts on the trolleys and this voltage is applied to the motors without the use of lowering transformers. The Cascade Tunnel on the Great Northern Railway has the only three phase locomotives in operation in this country.

High Conductivity Trolley Wire, Round and Grooved

(A.E.R.E.A. and A.S.T.M. Standard)

Material. The material of which high conductivity trolley wire is made should be electrolytic or low resistance lake copper wire bars of quality specified by standard specifications for electrolytic copper wire bars, cakes, slabs, billets, ingots, and ingot bars, or lake copper wire bars, cakes, slabs, billets, ingots, and ingot bars, of the American Society for Testing Materials, 1921 Book of Standards. Necessary brazes in trolley wire should be made in accordance with the best commercial practice, and tests upon a section of wire containing a braze should show at least 95 per cent of the tensile strength of the unbrazed wire. Elongation tests should not be made on test sections including brazes. The wire should be of uniform size, shape, and quality throughout, and should be free from all scale, flaws, splits and scratches not consistent with the best commercial practice.

Round Trolley Wire

(A.E.R.E.A. and A.S.T.M. Standard)

Dimensions. Dimensions of round trolley wire should be expressed as the diameter of the wire in decimal fractions of an inch, using not more than three places of decimals; *i.e.*, in even mils. Wire should be accurate in diameter. Variations of 1 per cent over or under nominal diameter are permissible.

Tensile Strength and Elongation. Round trolley wire should be so drawn that its tensile strength will not be less than the minimum values given in the following table:

Diameter, in.	Area, cir. mils	Tensile strength, lb. per sq. in.	Elongation in 10 in., per cent
0.548	300,000	47,000	4.50
0.460	211,600	49,000	3.75
0.410	168,100	51,000	3 25
0.365	133,225	52,800	2 80
0.325	105,625	54,500	2.40

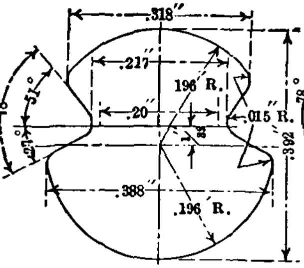
The elongation should be determined as the permanent increase in length due to the breaking of the wire in tension measured between bench marks placed upon the wire originally 10 in. apart. The fracture should be between the bench marks and not closer than 1 in. to either bench mark.

Inspection for Defects. For the purpose of determining and developing defects which may be prejudicial to the life of trolley wire, owing to its peculiar service, as compared to that of copper wire for other purposes, the following twisting tests should be made. Three twist tests should be made upon samples 10 in. long between the holders of the machine. The twisting machine should be so constructed that there is a linear motion of the tail stock with respect to the head, and the twist should be applied not faster than 10 turns per minute. All three samples should be tested to destruction and should not reveal under test any seams, pits, slivers or surface imperfections not consistent with the best commercial practice. At the time of fracture the wire should be twisting with reasonable uniformity. Wire should not be considered satisfactory which does not withstand at least 9 turns before breaking.

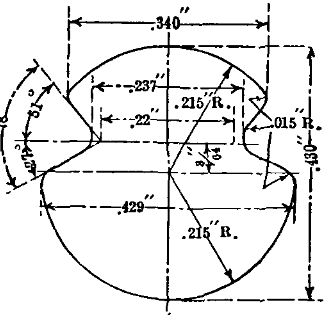
Resistivity. Electric resistivity should be determined upon fair samples by resistance measurements at a temperature of 20 deg. C.

(68 deg. F.). The wire should not exceed in resistivity 900.77 lb. per mile ohm.

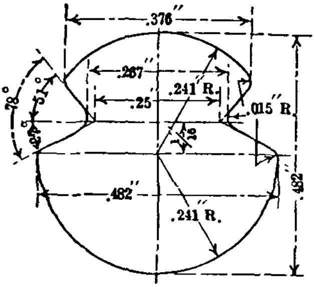
Density. For the purpose of calculating weights, cross-sections, etc., the specific gravity of copper should be taken as 8.89 at 20 deg. C.



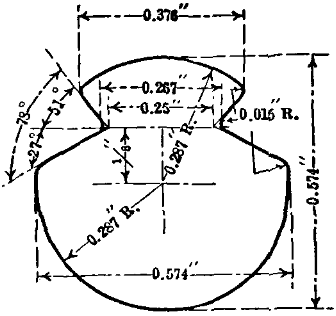
2/0
Nominal cir. mils 133,200
Actual area 0.1083 sq. in.
Actual weight 2201 lb. per mi.



3/0
Nominal cir. mils 168,100
Actual area 0.1314 sq. in.
Actual weight 2669 lb. per mi.



4/0
Nominal cir. mils 211,600
Actual area 0.1665 sq. in.
Actual weight 3382 lb. per mi.



300,000 c.m.
Nominal cir. mils 300,000
Actual area 0.2356 sq. in.
Actual weight 4786 lb. per mi.

FIG. 31.—Standard sections of grooved trolley wire.

Grooved Trolley Wire

(A.E.R.E.A. and A.S.T.M. Standard)

Standard sections are shown by Fig. 31. These are known as the "American Standard Grooved Trolley Wire Sections."

Dimensions. Dimensions of grooved trolley wire should be expressed as the area of cross-section in circular mils, the standard sizes being as follows:

300,000 cir. mils weighing 4,786 lb. per mile.
211,600 cir. mils weighing 3,382 lb. per mile.
168,100 cir. mils weighing 2,669 lb. per mile.
133,200 cir. mils weighing 2,201 lb. per mile.

Grooved trolley wire may vary 4 per cent over or under in weight per unit length from standard as determined from the nominal cross-section.

Physical Tests. Physical tests on grooved trolley wire should be made in the same manner as those upon the round wire (see page 587). The tensile strength of grooved wire should be at least 95 per cent of that required for round wire of the same sectional area; the elongation should be the same as that required for round wire of the same sectional area. The twist test should be omitted.

Resistivity. The requirements for resistivity are the same as those for round wire of the same sectional area (see page 587).

Low Conductivity, High Strength Trolley Wire. At this time (1924) no specification for special trolley wire of low conductivity and high tensile strength has the approval of the Amer. Elec. Ry. Assn. The committee on Power Distribution in 1923, for the purpose of giving member companies an opportunity to study definite data and so promote discussion, but not as a recommendation, tentatively suggested certain limiting values for high strength wire of 40 per cent conductivity as follows:

Diameter	Area C.M.	Tensile strength round wire		Min. twists in 10 in.	
		Actual	Lb. per sq. in.	Round	Grooved
0.460 in.	211,600	12,000	72,860	32	5
0.410 in.	168,100	10,000	75,740	34	6
0.365 in.	133,225	8,200	78,400	37	10
0.325 in.	105,625	6,700	80,760	42	12

Steel Trolley Wire. Steel wire is sometimes used as a secondary contact wire in catenary construction, and also has been used as the main trolley wire. In the latter case it must be supplemented by a larger cross section of feeder than is required by a copper trolley, but it has been claimed that the first cost is less, the life is longer, the reliability is greater, the temperature coefficient is less and the maintenance cost considerably less than copper. The Pacific Electric Railway has had more than 100 miles of steel trolley in use on both 600 and 1200 volt lines, and in both direct and catenary suspension. In places where rapid accelerating occurs, the steel wire appears to wear almost as rapidly as copper, at other places, considerably less. Galvanized steel wire of the 4/0 grooved section was used, and feed taps were installed much more frequently than with copper trolley. It was found that the maintenance on steel trolley was far below that of copper, since steel is much harder. The clips hold much better in the grooves, and the wire does not break if the trolley pole hits a span. There are no breaks due to crystallization at the ends of ears, splices or switches. In most cases, steel wire that has been accidentally broken and shorted on the rail can be restrung, whereas copper wire would become annealed. Steel is easier to pull up and can be handled for double the length of copper without sagging. Steel wire does not cut out on the sides when out of alinement nor does it pound flat at clips as does copper. Practically all the wear is due to burning and arcing at acceleration points.

As the correct welding of splices in steel trolley wire is so important, the method developed on the Pacific Electric system is worth description in some detail. The equipment consists of a Prest-O-Lite outfit of gages, welding torch and tips. The torch is style H, and the size of the tip No. 5, this size being most satisfactory for No. 4/0 double-grooved wire. The welding rods are of 1/4-in. or 3/16-in. diameter nickel steel. Fig. 32 shows the different stages of a weld. First the opposing ends of the wires to be joined are cut at an angle of 45 deg. and a space of 1/8 in. is left between them to provide for expansion. The flame of the torch, which is next applied, disposes of the galvanized scale which coats the wire. As soon as both ends of the wire are at the melting point, the welding rod is introduced, care being taken to keep both ends at about the same temperature. As the nickel steel is melted from the rod, it is fluxed with the steel trolley wire by means of the flame, starting at the bottom and building across and up. As soon as all necessary metal has been added, the weld while still red hot is filed down to a smooth underrun and shaped up as much as desired. Then cold water is poured on the wire, beginning about one foot on each side of the weld and running toward the weld. Finally the weld is reached and wetted until perfectly cold. The average length of time to weld the wire in the air varies from ten to fifteen minutes, according to accessibility.

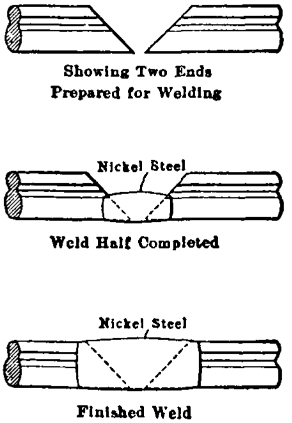


FIG. 32.—Weld joint in steel trolley wire.

Bending, heat and resistance tests were made on the galvanized wire by the Pacific Electric Railway, as follows: The bending test showed that the wire could stand six right-angle bends in a vise before breaking on the seventh bend. To find how much heat could be withstood and how much current could be carried, alternating current was passed through the wire until it was red hot. After cooling, the wire was found to be more flexible but still tough enough for use as trolley wire. It was given ten right-angle bends without breaking. Direct current, averaging 750 amp., was passed through the wire for five minutes. At the end of this time it was heated to a cherry red and the galvanizing had burned off, but the wire was otherwise uninjured. This latter test proved that under normal service conditions the wire would not be damaged by high temperature. However, even if the galvanized coating was burned off by a heavy ground, this would not be so serious as the annealing of grounded copper. Volt-ampere readings showed the resistance to be 0.000342 ohm per foot at 100 deg. F., or 6.53 times the resistance of No. 4/0 copper at 100 deg. F. As steel is 10.6 per cent lighter than copper, its resistance per unit of weight is 5.83 times that of copper.

Overhead Line Material. Standard specifications for various items of material for overhead construction, 750 volt direct current, direct suspension, have been recommended by the Amer. Elec. Ry. Eng. Assn., and are shown in the Manual of that association. The specifications cover iron and steel fittings, wood cross-arms, bronze castings, seven strand steel cable, switch boxes, tree and cable guards, wood insulator pins and brackets, porcelain strain and feeder insulators, and wood break strain insulators.

Selection of Poles

The selection of the pole for trolley line construction is dependent upon the type of line which is to be supported, and involves three elements, namely, height of pole, width of street for span construction, and sag desired in span. The allowable sag having been decided upon, the next step is to find the horizontal pull for the span wire. This is found as shown on pages 570 to 573, and the pole of the desired length must then be selected to withstand the pull thus determined. If the pole selected has a deflection of say 3 in. for the required pull, then the pole must be given a 3 in. rake if it is desired to pull it up vertically when the span is completed.

Steel Poles

Deflection.

$$x = \frac{Pl^3}{3EI}$$

in which x = deflection in inches, 18 in. from top of pole
P = horizontal pull, 18 in. from top of pole
l = length of pole from point of application of P to ground line (6 ft. from base end), in inches
E = modulus of elasticity of steel (29,000,000)
I = moment of inertia of base section.

Deflections from various loadings together with descriptions of poles are given in the following tables. The variations of lengths of

Length of pole, ft	Description of poles: Size and kind of pipe: Butt: Standard	Butt: Extra heavy	Middle: Standard	Top: Standard	Weight of pole, lb	Section lengths, ft: Butt	Middle	Top	Pole set 6′ 0″ in ground—load applied and deflection measured 18″ from free end, probable deflections when held and loaded as stated. Load in pounds applied 18″ from free end of pole, in: 100	500	600	800	1,000	1,200	1,400	1,600	1,800	2,100	2,400	2,700	3 000	Greatest safe load, lb	Length of pole, ft
28	5		4	3	385	18	7	6	1 23													443	28
28		5	4	3	495	18	7	6	90	4 52	5 42											603	28
28	6		5	4	509	18	7	6	65	3 24	3 89											692	28
28		6	5	4	681	18	7	6	46	2 28	2 73	3 64										993	28
28	7		6	5	645	18	7	6	39	1 93	2 32	3 09										992	28
28		7	6	5	906	18	7	6	26	1 27	1 53	2 04	2 55	3 06	3 57							1,525	28
28	8		7	6	791	18	7	6	25	1 23	1 48	1 98	2 47	2 97								1,366	28
28		8	7	6	1,058	18	7	6	17	84	1 01	1 34	1 68	2 02	2 35	2 69	3 03					1,990	28
28	9		8	7	951	18	7	6	17	83	99	1 32	1 65	1 98	2 31	2 64	2 97					1,817	28
28		9	8	7	1 218	18	7	6	12	59	71	94	1 18	1 42	1 65	1 89	2 12	2 48	2 83			2,527	28
28	10		9	8	1 137	18	7	6	11	57	68	90	1 13	1 36	1 58	1 81	2 04	2 37	2 71			2 432	28
28		10	9	8	1 393	18	7	6	08	41	50	66	83	1 00	1 16	1 33	1 49	1 74	1 99	2 24	2 49	3,205	28
29	5		4	3	393	18	7	7	1 45													422	29
29		5	4	3	503	18	7	7	1 07	5 35												575	29
29	6		5	4	520	18	7	7	76	3 83	4 58											658	29
29		6	5	4	692	18	7	7	54	2 67	3 21	4 28										947	29
29	7		6	5	659	18	7	7	45	2 26	2 72	3 63										946	29
29		7	6	5	920	18	7	7	30	1 49	1 79	2 38	2 98	3 58	4 17							1,450	29

29	8		7	6	811	18	7	7	29	1 44	1 73	2 31	2 88	3 46								1,304	29
29		8	7	6	1,078	18	7	7	20	98	1 18	1 57	1 96	2 35	2 74	3 14	3 53					1,900	29
29	9		8	7	974	18	7	7	19	96	1 15	1 54	1 92	2 30	2 69	3 07						1,733	29
29		9	8	7	1,241	18	7	7	14	69	82	1 09	1 37	1 64	1 92	2 19	2 46	2 88	3 29			2 410	29
29	10		9	8	1,166	18	7	7	13	65	77	1 03	1 29	1 55	1 81	2 06	2 32	2 71				2,320	29
29		10	9	8	1,422	18	7	7	10	48	58	78	97	1 16	1 36	1 55	1 74	2 04	2 33	2 62	2 91	3,060	29
30	5		4	3	406	18	9	6	1 67													403	30
30		5	4	3	516	18	9	6	1 23	6 16												550	30
30	6		5	4	539	18	9	6	88	4 39	5 27											629	30
30		6	5	4	711	18	9	6	62	3 08	3 70	4 93										906	30
30	7		6	5	682	18	9	6	52	2 60	3 12	4 17										903	30
30		7	6	5	943	18	9	6	34	1 71	2 06	2 74	3 43	4 12								1,387	30
30	8		7	6	840	18	9	6	33	1 65	1 99	2 65	3 31	3 97								1,245	30
30		8	7	6	1,107	18	9	6	23	1 12	1 35	1 80	2 25	2 70	3 15	3 60	4 05					1,815	30
30	9		8	7	1,008	18	9	6	22	1 10	1 33	1 77	2 21	2 65	3 09	3 54						1,655	30
30		9	8	7	1,275	18	9	6	16	79	95	1 26	1 58	1 90	2 21	2 53	2 85	3 32				2,305	30
30	10		9	8	1,205	18	9	6	15	74	89	1 18	1 48	1 78	2 07	2 37	2 66	3 11				2,220	30
30		10	9	8	1,461	18	9	6	11	56	67	90	1 12	1 34	1 57	1 79	2 02	2 35	2 69	3 02		2,920	30
31	5		4	3	413	18	9	7	1 95													387	31
31		5	4	3	523	18	9	7	1 44	7 18												527	31
31	6		5	4	549	18	9	7	1 02	5 09	6 12											603	31
31		6	5	4	721	18	9	7	72	3 58	4 29	5 72										867	31
31	7		6	5	696	18	9	7	60	3 00	3 60	4 80										865	31
31		7	6	5	958	18	9	7	40	1 98	2 38	3 18	3 97	4 77								1,325	31
31	8		7	6	860	18	9	7	38	1 90	2 29	3 05	3 81									1,192	31
31		8	7	6	1,127	18	9	7	26	1 30	1 55	2 07	2 59	3 11	3 63	4 15						1,740	31
31	9		8	7	1,031	18	9	7	25	1 27	1 52	2 03	2 54	3 05	3 56							1,585	31
31		9	8	7	1,299	18	9	7	18	91	1 09	1 46	1 82	2 18	2 55	2 91	3 28	3 82				2,205	31
31	10		9	8	1,234	18	9	7	17	85	1 02	1 36	1 70	2 04	2 38	2 72	3 06	3 57				2,125	31
31		10	9	8	1,491	18	9	7	13	64	77	1 02	1 28	1 54	1 79	2 05	2 31	2 69	3 07	3 45		2,795	31
32	5		4	3	427	19	9	7	2 17													371	32
32		5	4	3	545	19	9	7	1 61	8 02												505	32
32	6		5	4	568	19	9	7	1 14	5 70												578	32

TUBULAR STEEL POLES—DEFLECTIONS AND LOADS—(*Concluded*)

Length of pole, ft.	Description of poles: Size and kind of pipe: Butt: Standard	Butt: Extra heavy	Middle: Standard	Top: Standard	Weight of pole, lb.	Section lengths, ft.: Butt	Middle	Top	Pole set 6' 0" in ground—load applied and deflection measured 18" from free end, probable deflections when held and loaded as stated. Load in pounds applied 18" from free end of pole, in.: 100	500	600	800	1,000	1,200	1,400	1,600	1,800	2,100	2,400	2,700	3,000	Greatest safe load, lb.	Length of pole, lb.
32	....	6	5	4	749	19	9	7	.80	4.01	4.82	6.42	......	......	......	......	......	......	......	......	......	832	32
32	7	....	6	5	720	19	9	7	.68	3.37	4.05	5.40	......	......	......	......	......	......	......	......	......	830	32
32	....	7	6	5	995	19	9	7	.45	2.22	2.67	3.56	4.45	5.33	......	......	......	......	......	......	......	1,275	32
32	8	....	7	6	888	19	9	7	.43	2.14	2.57	3.43	4.28	......	......	......	......	......	......	......	......	1,143	32
32	....	8	7	6	1,169	19	9	7	.29	1.46	1.75	2.33	2.92	3.50	4.08	4.67	......	......	......	......	......	1,665	32
32	9	....	8	7	1,066	19	9	7	.29	1.43	1.72	2.29	2.86	3.43	4.00	......	......	......	......	......	......	1,520	32
32	....	9	8	7	1,348	19	9	7	.20	1.02	1.22	1.63	2.04	2.45	2.85	3.27	3.67	4.28	......	......	......	2,115	32
32	10	....	9	8	1,274	19	9	7	.19	.96	1.15	1.53	1.92	2.30	2.69	3.07	3.46	......	......	......	......	2,035	32
32	....	10	9	8	1,545	19	9	7	.14	.72	.86	1.15	1.44	1.73	2.02	2.31	2.59	3.03	3.45	......	......	2,680	32
33	5	....	4	3	439	19	10	7	2.48	......	......	......	......	......	......	......	......	......	......	......	......	357	33
33	....	5	4	3	555	19	10	7	1.83	......	......	......	......	......	......	......	......	......	......	......	......	487	33
33	6	....	5	4	582	19	10	7	1.30	6.49	......	......	......	......	......	......	......	......	......	......	......	555	33
33	....	6	5	4	764	19	10	7	.91	4.56	5.47	......	......	......	......	......	......	......	......	......	......	798	33
33	7	....	6	5	739	19	10	7	.77	3.83	4.60	......	......	......	......	......	......	......	......	......	......	797	33
33	....	7	6	5	1,015	19	10	7	.51	2.53	3.04	4.05	5.07	6.08	......	......	......	......	......	......	......	1,225	33
33	8	....	7	6	910	19	10	7	.49	2.43	2.92	3.80	4.87	......	......	......	......	......	......	......	......	1,098	33
33	....	8	7	6	1,192	19	10	7	.33	1.65	1.99	2.65	3.31	3.97	4.63	5.30	......	......	......	......	......	1,605	33

33	9	....	8	7	1,004	19	10	7	.33	1.62	1.95	2.60	3.25	3.90	4.55	....	....	†....	....	....	....	1,460	33
33	....	9	8	7	1,376	19	10	7	.23	1.16	1.39	1.86	2.32	2.78	3.25	3.71	4.18	....	....	....	....	2,035	33
33	10	....	9	8	1,308	19	10	7	.22	1.09	1.31	1.74	2.18	2.62	3.05	3.49	3.93	....	....	....	....	1,955	33
33	....	10	9	8	1,579	19	10	7	.16	.82	.98	1.31	1.64	1.97	2.30	2.63	2.95	3.28	3.94	....	....	2,580	33
34	5	....	4	3	454	20	10	7	2.75	....	....	....	....	....	....	....	....	....	....	....	....	343	34
34	....	5	4	3	575	20	10	7	2.03	....	....	....	....	....	....	....	....	....	....	....	....	467	34
34	6	....	5	4	601	20	10	7	1.44	7.21	....	....	....	....	....	....	....	....	....	....	....	534	34
34	....	6	5	4	792	20	10	7	1.01	5.08	6.09	....	....	....	....	....	....	....	....	....	....	768	34
34	7	....	6	5	763	20	10	7	.86	4.28	5.13	....	....	....	....	....	....	....	....	....	....	767	34
34	....	7	6	5	1,053	20	10	7	.56	2.81	3.37	4.49	5.62	....	....	....	....	....	....	....	....	1,177	34
34	8	....	7	6	939	20	10	7	.54	2.71	3.25	4.33	5.42	6.51	....	....	....	....	....	....	....	1,057	34
34	....	8	7	6	1,235	20	10	7	.37	1.85	2.22	2.96	3.70	4.44	5.18	....	....	....	....	....	....	1,543	34
34	9	....	8	7	1,128	20	10	7	.36	1.81	2.17	2.89	3.62	4.34	5.07	....	....	....	....	....	....	1,405	34
34	....	9	8	7	1,425	20	10	7	.26	1.29	1.55	2.07	2.59	3.11	3.63	4.15	4.67	....	....	....	....	1,955	34
34	10	....	9	8	1,348	20	10	7	.24	1.21	1.45	1.93	2.42	2.90	3.39	3.87	4.36	....	....	....	....	1,885	34
34	....	10	9	8	1,634	20	10	7	.18	.92	1.10	1.46	1.83	2.20	2.56	2.93	3.29	3.84	4.39	....	....	2,480	34
35	5	....	4	3	469	21	10	7	3.03	....	....	....	....	....	....	....	....	....	....	....	....	331	35
35	....	5	4	3	595	21	10	7	2.23	....	....	....	....	....	....	....	....	....	....	....	....	450	35
35	6	....	5	4	620	21	10	7	1.59	7.97	....	....	....	....	....	....	....	....	....	....	....	515	35
35	....	6	5	4	820	21	10	7	1.12	5.60	6.72	....	....	....	....	....	....	....	....	....	....	741	35
35	7	....	6	5	787	21	10	7	.95	4.73	5.68	....	....	....	....	....	....	....	....	....	....	738	35
35	....	7	6	5	1,091	21	10	7	.62	3.12	3.75	4.99	6.24	....	....	....	....	....	....	....	....	1,135	35
35	8	....	7	6	968	21	10	7	.60	3.01	3.61	4.8	6.02	....	....	....	....	....	....	....	....	1,018	35
35	....	8	7	6	1,278	21	10	7	.41	2.05	2.46	3.2	4.10	4.92	5.74	....	....	....	....	....	....	1,485	35
35	9	....	8	7	1,192	21	10	7	.40	2.01	2.41	3.2	4.02	4.82	....	....	....	....	....	....	....	1,355	35
35	....	9	8	7	1,474	21	10	7	.29	1.43	1.72	2.2	2.87	3.44	4.02	4.59	5.17	....	....	....	....	1,885	35
35	10	....	9	8	1,388	21	10	7	.27	1.35	1.62	2.16	2.70	3.24	3.78	4.32	4.87	....	....	....	....	1,818	35
35	....	10	9	8	1,689	21	10	7	.20	1.01	1.21	1.62	2.02	2.42	2.83	3.23	3.64	4.24	....	....	....	2,390	35

Steel Pole Pipe Table

Nominal diameter in inches	3		4		5		6	
	Standard	Extra heavy	Standard	Extra heavy	Standard	Extra heavy	Standard	Extra heavy
Weight of pipe, lb. per ft.	7.575	10.252	10.790	14.983	14.617	20.778	18.974	28.573
Actual diameter, in.:								
External $= D$	3.500	3.500	4.500	4.500	5.563	5.563	6.625	6.625
Internal $= d$	3.068	2.900	4.026	3.826	5.047	4.813	6.065	5.761
Thickness	.216	.300	.237	.337	.258	.375	.280	.432
Circumference, in.:								
External	10.996	10.996	14.137	14.137	17.477	17.477	20.813	20.813
Internal	9.638	9.111	12.648	12.020	15.856	15.120	19.054	18.099
Transverse areas, sq. in.:								
External	9.621	9.621	15.904	15.904	24.306	24.306	34.472	34.472
Internal	7.393	6.605	12.730	11.497	20.006	18.193	28.891	26.067
Metal	2.228	3.016	3.174	4.407	4.300	6.113	5.581	8.405
Moment of inertia $= I$. $I = .0491\ (D^4 - d^4)$	3.017	3.894	7.233	9.610	15.162	20.671	28.140	40.491
Modulus of section $= S$. $S = .0982\left(D^3 - \frac{d^4}{D}\right) = \frac{2I}{D}$	1.724	2.225	3.214	4.271	5.451	7.432	8.496	12.224
R = Moment of resistance. Inch pounds. $R = SF$ (F = 20,000 lb. sq. in.)	34,480	44,500	64,280	85,420	109,020	148,640	169,920	244,480

See notes on page 597.

Steel Pole Pipe Table—(Concluded)

Nominal diameter in inches	7		8		9		10	
	Standard	Extra heavy	Standard	Extra heavy	Standard	Extra heavy	Standard	Extra heavy
Weight of pipe, lb. per ft.	23.544	38.048	28.554	43.388	33.907	48.728	40.483	54.735
Actual diameter, in:								
External = D	7.625	7.625	8.625	8.625	9.625	9.625	10.750	10.750
Internal = d	7.023	6.625	7.981	7.625	8.941	8.625	10.020	9.750
Thickness	.301	.500	.322	.500	.342	.500	.365	.500
Circumference, in:								
External	23.955	23.955	27.096	27.096	30.238	30.238	33.772	33.772
Internal	22.063	20.813	25.073	23.955	28.089	27.096	31.479	30.631
Transverse areas, sq. in.:								
External	45.664	45.664	58.426	58.426	72.760	72.760	90.763	90.763
Internal	38.738	34.474	50.027	45.666	62.786	58.430	78.853	74.663
Metal	6.926	11.190	8.399	12.760	9.974	14.33	11.91	16.10
Moment of inertia = I $I = .0491\ (D^4 - d^4)$	46.515	71.370	72.489	105.720	107.580	149.63	160.73	211.95
Modulus of section = S $S = .0982\left(D^3 - \frac{d^4}{D}\right) = \frac{2I}{D}$	12.201	18.720	16.809	24.505	22.354	31.092	29.917	39.433
R = Moment of resistance Inch pounds $R = SF$ ($F = 20{,}000$ lb. sq. in.)	244,020	374,400	336,180	490,100	447,080	621,840	598,340	788,660

The above table is figured on tubing having a tensile strength of 45,000 lb. per square inch, an elastic limit of 24,000 lb. per square inch and a modulus of elasticity of 26,000,000. The greatest safe fiber stress is taken at 20,000 lb. per square inch.

The lower end of the poles was assumed to be buried 6 ft. 0 in. in the ground. Moment of inertia $= I = 0.0491\ (D^4 - d^4)$.

Deflection $= x = \frac{PL^3}{3EI}$. The deflections were measured and the load was applied 18 in. from the free end of the pole.

P = load applied 18 in. from the free end. l = length of the pole in inches from the ground line to the point where the load is applied. I = moment of inertia and E = modulus of elasticity.

the sections of a pole within commercial limits have but little effect upon its strength, stiffness, or weight. The section lengths given conform closely to those usually employed. In general a number of poles will satisfy the requirements, and choice must be made from those of the proper length. Usually the lightest weight pole would be selected unless the deflection was considered too great, in which case the choice would be determined finally by the deflection.

Steel Pipe Used in Making Poles. All steel pipe used in making up poles should be good quality of mild steel of uniform thickness, free from bad welds, cold shuts, outside slivers, cracks, flaws, open seams, rivets or other imperfections which would affect the strength, life or appearance of the pole. The tables on pages 596 and 597 give data on steel pipe which was used in making up the tables on pages 592 to 595.

Joints in Steel Poles. The joints should be 18 in. in length and should be made hot without reducing the diameter of the smaller pipe more than 1/8 in., and without any reduction in the thickness of either pipe. The completed joint should caliper the same outside diameter for its entire length, should be free from cracks or flaws, and should be water tight. Any pole when dropped three times from a height of 6 ft. upon a solid wooden block on a rigid base should not show any telescoping at the joints.

Ground Sleeve. A ground sleeve, if required, should be made of standard weight pipe 2 ft. in length; it should be shrunk on the butt section with the bottom of the sleeve 5 ft. 6 in. from the base of the pole.

Wood Poles

Classification According to Purpose, A.E.R.E.A. Standard. To determine the character of poles to be used in trolley line construction, they may be divided into three classes, A, B, C.

Class A. For span construction on streets or rights of way where a 35 ft. span is required, or for heavy feeder lines carrying from one to six cross-arms.

Class B. For span or bracket construction where spans are not more than 35 ft., or bracket line construction carrying two transmission circuits, one feeder arm, and two telephone and signal arms.

Class C. For constructing telephone, signal and other light auxiliary lines where no side strain is required.

Chestnut Poles

MINIMUM DIMENSIONS OF CHESTNUT POLES IN INCHES
(A.E.R.E.A. STANDARD)

Length of poles (feet)	Class A		Class B		Class C	
	Circumference		Circumference		Circumference	
	Top, in.	Six feet from butt, in.	Top, in.	Six feet from butt, in.	Top, in.	Six feet from butt, in.
25	24	36	21	31	20	30
30	24	40	22	36	20	33
35	24	43	22	40	20	36
40	24	45	22	43	20	40
45	24	48	22	47	20	43
50	24	51	22	50	20	46
55	22	54	22	53	20	49
60	22	57	22	56	....	....
65	22	60	22	59	....	....
70	22	63	22	62	....	....
75	22	66	22	65	....	....

Quality of Chestnut Poles and Time of Delivery. A pole should not have been cut over 2 years, nor should it have been cut when dead. It should not be shaved, nor should it have a crack or large season checks or short crook or a crook or sweep in two planes, or a short reverse curve. Poles over 30 ft. in length should have less sweep than the shorter poles and as follows (A.E.R.E.A. Standard):

Length	Sweep not over
35 ft.	10 in.
40 ft.	10 in.
45 ft.	11 in.
50 ft.	11 in.
55 ft.	12 in.
60 ft.	13 in.
65 ft.	14 in.
70 ft.	15 in.

The sweep is measured between the 6 ft. mark and the top of the pole.

All poles should have sound tops. Poles with double tops should be examined carefully for split tops or rot where the two parts join. All poles should have reasonably sound butts. Hollow butts should be carefully examined, and poles having them rejected, if the hole runs over 4 ft. Poles with hollow butts should be rejected if there is evidence of decay at the further end of the hole inside the pole. There should be no sap rot. There should be no "cat faces" unless they are sound and small and the pole has an increased diameter at the "cat faces." There should be no "cat faces" near the 6 ft. mark or within 10 ft. of the top. Poles should be examined carefully for black knots, hollow knots or projections caused by overgrown knots. Overgrown knots should be trimmed off and examined for internal rot. There should be no evidence of rot at knot holes nor should there be large hollow knots. There should be no top, butt or black knot which has been plugged. The pole should be free from woodpeckers' holes or evidence of ants, worms or grubs.

Eastern White Cedar Poles

MINIMUM DIMENSIONS OF EASTERN WHITE CEDAR POLES IN INCHES (A.E.R.E.A. STANDARD)

Length of poles (feet)	Class A*		Class B*		Class C*	
	Circumference		Circumference		Circumference	
	Top (inches)	Six feet from butt (inches)	Top (inches)	Six feet from butt (inches)	Top (inches)	Six feet from butt (inches)
30	24	40	22	36	18¾	33
35	24	43	22	38	18¾	36
40	24	47	22	43	18¾	40
45	24	50	22	47	18¾	43
50	24	53	22	50	18¾	46
55	24	56	22	53	18¾	49
60	24	59	22	56	....	....

* See page 598.

Poles should be of the best quality of either seasoned or live green cedar of dimensions given in the above table. Seasoned poles should have preference over green poles, provided they have not been held for seasoning long enough to have developed any objectionable timber defects. They should be reasonably straight, well proportioned from butt to top, both ends squared, the bark peeled and all knots and limbs closely trimmed. They should not be shaved. A pole should not have dead streaks covering more than one-quarter of its surface. A dead pole or one having dead streaks covering more than one-quarter of its surface should be rejected. A pole having dead streaks covering less than one-quarter of it surface should have a circumference greater than otherwise required. This increase in the circumference should be sufficient to afford a cross-sectional area of sound wood equivalent to that of a pole of the same class.

Fire Killed or River Poles. No dark red or copper colored pole which, when scraped, does not show good live timber should be accepted. No pole should have more than one complete twist for every 20 ft. in length nor should it have a large season check. There should be no "cat faces" unless they are small and perfectly sound and the poles have an increased diameter at the "cat faces," nor have "cat faces" near the 6 ft. mark or within 10 ft. of the top. A pole should contain no sap rot, evidence of internal rot as disclosed by a careful examination. There should be no black knots, hollow knots, woodpeckers' holes, or plugged holes, and no pole should show evidence of having been eaten by ants, worms or grubs, except that poles containing worm or grub marks below the 6 ft. mark may be accepted. A pole should have a sound top with no pencil holes. A pole having a double top or double heart should be free from rot where the two parts or hearts join. A pole should be free from ring rot (rot in the form of a complete or partial ring). A pole should not have a short crook or bend, a crook or bend in two planes or a reverse curve. The amount of sweep, measured between the 6 ft. mark and the top of the pole, that may be present in poles, is shown in the following table (A.E.R.E.A. Standard):

Length	Sweep not over
35 ft.	10½ in.
40 ft.	12 in.
45 ft.	10 in.
50 ft.	10 in.
55 ft.	11 in.
60 ft.	12 in.

Poles may have hollow hearts under the conditions shown in the following table which is Standard of the A.E.R.E.A.:

Average diameter of hollow heart	Add to butt requirements of		
	30 ft. poles	35, 40 and 45 ft. poles	50, 55, 60 and 65 ft. poles
2-in.	Nothing	Nothing	Nothing
3-in.	1-in.	Nothing	Nothing
4-in.	2-in.	Nothing	Nothing
5-in.	3-in.	1-in.	Nothing
6-in.	4-in.	2-in.	1-in.
7-in.	Reject	4-in.	2-in.
8-in.	Reject	6-in.	3-in.
9-in.	Reject	Reject	4-in.
10-in.	Reject	Reject	5-in.
11-in.	Reject	Reject	7-in.
12-in.	Reject	Reject	9-in.
13-in.	Reject	Reject	Reject

Scattered rot, unless it is near the outside of the pole, may be estimated as being the same as heart rot of equal area. Poles with cup shakes (checks in the form of rings) which also have heart or star checks may be considered as equal to poles having hollow hearts of the average diameter of the cup shakes.

Western White Cedar Poles

("Red Cedar," "Western Cedar," or "Idaho Cedar")

MINIMUM DIMENSIONS OF WESTERN WHITE CEDAR POLES IN INCHES—(CIRCUMFERENCE). (A.E.R.E.A. STANDARD)

Length of poles (feet)	Class A*	Class B*	Class C*
	Top 28	Top 25	Top 22
	Butt	Butt	Butt
30	37	34	30
35	40	36	32
40	43	38	34
45	45	40	36
50	47	42	38
55	49	44	40
60	52	46	41
65	54	48	43

* See page 598.

NOTE: "Top" measurement being the circumference at the top of the pole, and the "butt" measurement, the circumference 6 ft. from the butt.

Poles should be of the best quality of either seasoned or live green cedar. Seasoned poles should have preference over green poles, provided they have not been held for seasoning long enough to have developed any objectionable timber defects. Poles should be reasonably straight, well proportioned from butt to top, both ends squared, sound tops, the bark peeled, and all knots and limbs closely trimmed. Large knots, if sound and trimmed close, should not be considered a defect. A pole should not contain hollow or rotten knots, nor should it contain sap rot, woodpeckers' holes, or plugged holes, nor show evidence of having been eaten by ants, worms or grubs. A pole should not be dead nor should it have dead streaks covering more than one-quarter of its surface. A pole having dead streaks covering less than one-quarter of its surface should have a circumference greater than otherwise required. The increase in the circumference should be sufficient to afford a cross-sectional area of sound wood equivalent to that of a sound pole of the same class. A pole should not have more than one complete twist for every 20 ft. in length nor should it have a large season check or crack. No pole should have a short crook or bend, a crook or bend in two planes, or a reverse crook or bend. The amount of sweep measured between the 6 ft. mark and the top of the pole should not exceed 1 in. to every 6 ft. in length. A pole should have no "cat faces," unless they are small and perfectly sound and the pole has an increased diameter at the "cat face," nor have "cat faces" near the 6 ft. mark, or within 10 ft. of its top. A pole having cup shakes (checks in the form of rings) should contain no heart or star shakes which enclose more than 10 per cent of the area of the butt. A pole should not have butt rot covering in excess of 10 per cent of the total area of the butt. The butt rot, if present, must be located close to the center.

Reinforced Concrete Poles

Figs. 33 and 34 and the following formulas are from the Manual of the Amer. Elec. Ry. Assn. The square section is more efficient and economical than either the hexagonal or the octagonal. The two latter were developed for situations in which the square pole would not present a sufficiently artistic appearance.

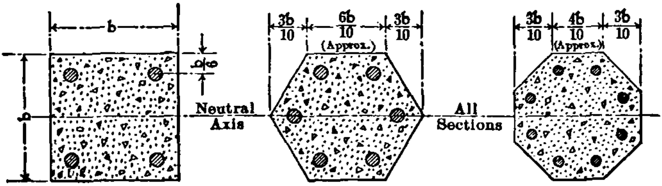


FIG. 33.—Proposed sections for concrete poles.

SQUARE

Deflection $x = \dfrac{30Pl^3}{E_s b^4}$

Area $= b^2$

Safe load, $P_1 = \dfrac{1750d^2b}{L}$ (Approx.)

HEXAGONAL

$x = \dfrac{40Pl^3}{E_s b^4}$

Area $= \dfrac{9b^2}{10}$ (Approx.)

$P_2 = \dfrac{9P_1}{10}$

$= \dfrac{1575d^2b}{L}$ (Approx.)

OCTAGONAL

$x = \dfrac{47Pl^3}{E_s b^4}$

Area $= \dfrac{8b^2}{10}$ (Approx.)

Safe load, $P_3 = \dfrac{8P_1}{10}$

$= \dfrac{1400d^2b}{L}$ (Approx.)

In which

x = deflection at a point 18 in. below top of pole, inches. (See note below)
P = horizontal pull at a point 18 in. below top of pole, pounds
l = length of pole from point of application of P to the ground line (6 ft. from base end), inches
L = length of pole from point of application of P to the ground line (6 ft. from base end), feet
E_s = modulus of elasticity of steel used
b = side of square pole or distance between parallel faces of hexagonal or octagonal pole at ground line, inches
d = diameter of steel rods, inches
Four per cent of steel used in all sections.

NOTE: The above reference states that coefficients 30, 40 and 47 of the formulas for deflection, respectively, should be increased 25 per cent for practical application.

The values in the table (pp. 605 to 607) from the "Miscellaneous Practice" of the American Electric Railway Engineering Association are calculated on the proposed design of a reinforced-concrete pole of square cross-section, shown in Fig. 34.

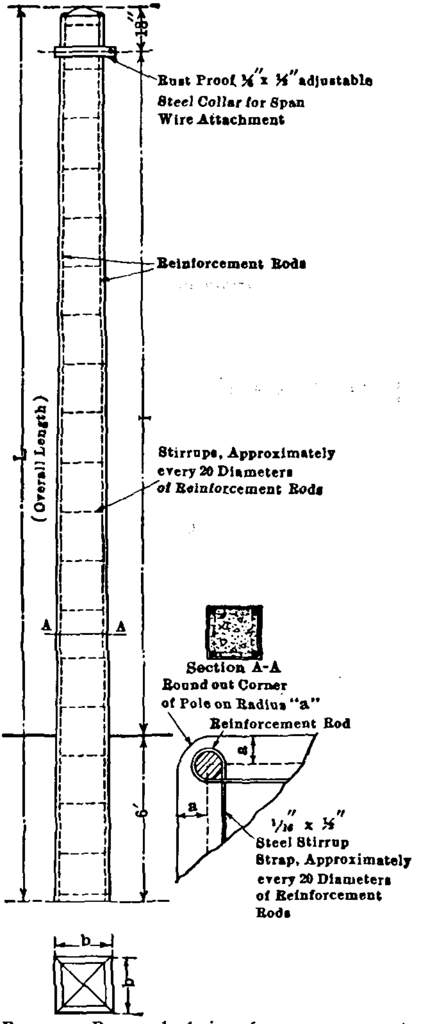


Fig. 34.—Proposed design for square concrete pole.

The general results of tests and experience of the Amer. Elec. Ry. Eng. Assn. Committee with concrete poles indicate certain facts as follows: failure of a pole is always due to stretching of the reinforcing rods on the tension side; a failure is always preceded by the appearance of hair-line cracks in the concrete on the tension side, at rather frequent and regular intervals from the ground line up; it is advantageous to use a high grade of reinforcing steel to secure the maximum tensile strength; plain round reinforcing rods are essentially as satisfactory as twisted or other rough rods because in general the rods will elongate before they slip in the concrete; a larger number of small rods is preferable to a smaller number of large rods as a better distributed reinforcement may be secured for a given amount of steel and a greater bonding contact surface is presented to the concrete; the reinforcement need not be uni-

Reinforced Concrete Poles—Deflections and Loads

Length feet	Description of pole: Butt section (In.)	Top section (In.)	Diameter reinforcement (In.)	Weight of pole in lb.	Probable deflection in inches at point of application of load—18 in. from top, with pole 6 ft. 0 in. in ground: 100	200	400	600	800	1,000	1,200	1,400	1,600	1,800	2,100	2,400	2,700	3,000	3,300	Greatest safe load in lb.	Length feet
28	5×5	3×3	9/16	525	2.7															121	28
28	6×6	4×4	5/8	812	1.5															190	28
28	7×7	5×5	3/4	1176	0.825	1.650														330	28
28	8×8	6×6	1	1624	0.435	0.870	1.755													683	28
28	9×9	7×7	1-1/8	2128	0.280	0.561	1.122	1.686												988	28
28	10×10	8×8	1-1/4	2688	0.180	0.360	0.720	1.080	1.440	1.800										1372	28
28	11×11	9×9	1-3/8	3304	0.127	0.255	0.510	0.765	1.020	1.275	1.530	1.785								1844	28
28	12×12	10×10	1-1/2	4090	0.090	0.180	0.360	0.540	0.720	0.900	1.080	1.260	1.440	1.605						2415	28
28	13×13	11×11	1-5/8	4746	0.066	0.133	0.267	0.400	0.534	0.669	0.802	0.936	1.068	1.230	1.404	1.605				3091	28
28	14×14	12×12	1-3/4	5558	0.048	0.096	0.195	0.292	0.390	0.486	0.585	0.675	0.780	0.876	1.020	1.179	1.314	1.458	1.605	3884	28
29	5×5	3×3	9/16	544	3.3															115	29
29	6×6	4×4	5/8	841	1.605															182	29
29	7×7	5×5	3/4	1218	0.864	1.726														314	29
29	8×8	6×6	1	1682	0.510	1.005	2.010													651	29
29	9×9	7×7	1-1/8	2204	0.315	0.630	1.260	1.905												942	29
29	10×10	8×8	1-1/4	2784	0.214	0.429	0.855	1.275	1.710	2.145										1308	29
29	11×11	9×9	1-3/8	3422	0.150	0.315	0.615	0.915	1.230	1.515	1.830	2.145								1758	29
29	12×12	10×10	1-1/2	4232	0.105	0.210	0.420	0.610	0.840	1.050	1.260	1.470	1.680	1.890						2302	29
29	13×13	11×11	1-5/8	4915	0.075	0.150	0.300	0.450	0.600	0.750	0.885	1.035	1.185	1.320	1.545	1.770				2945	29
29	14×14	12×12	1-3/4	5756	0.052	0.105	0.210	0.322	0.426	0.640	0.645	0.750	0.855	0.960	1.123	1.284	1.444	1.605		3706	29
29	15×15	13×13	1-7/8	6670	0.040	0.082	0.165	0.247	0.330	0.405	0.495	0.542	0.660	0.742	0.862	0.985	1.108	1.231	1.354	4578	29
30	5×5	3×3	9/16	562	3.904															112	30
30	6×6	4×4	5/8	870	1.875															174	30
30	7×7	5×5	3/4	1260	1.005	2.010														300	30
30	8×8	6×6	1	1740	0.570	1.140	2.280	3.480												622	30
30	9×9	7×7	1-1/8	2280	0.360	.0735	1.470	2.205	2.940											900	30

REINFORCED CONCRETE POLES—DEFLECTIONS AND LOADS—*Continued*

Length feet	Description of pole			Weight of pole in lb.	Probable deflection in inches at point of application of load—18 in. from top, with pole 6 ft. 0 in. in ground															Greatest safe load in lb.	Length feet
	Butt section	Top section	Diameter reinforcement		100	200	400	600	800	1,000	1,200	1,400	1,600	1,800	2,100	2,400	2,700	3,000	3,300		
	(In.)	(In.)	(In.)																		
30	10×10	8×8	1-1/4	2880	0.240	0.480	0.972	1.458	1.944	2.430	2.910	....	....	....	....	....	....	....	....	1250	30
30	11×11	9×9	1-3/8	3540	0.166	0.333	0.666	0.999	1.332	1.665	1.998	2.331	2.664	....	....	....	....	....	....	1680	30
30	12×12	10×10	1-1/2	4275	0.117	0.234	0.468	0.702	0.936	1.173	1.404	1.638	1.872	2.106	2.463	....	....	....	....	2200	30
30	13×13	11×11	1-5/8	5091	0.078	0.156	0.312	0.468	0.624	0.784	0.936	1.092	1.248	1.404	1.638	1.872	2.118	....	....	2816	30
30	14×14	12×12	1-3/4	5955	0.063	0.126	0.252	0.378	0.504	0.630	0.756	0.882	1.008	1.134	1.323	1.512	1.701	1.890	2.079	3539	30
31	5×5	3×3	9/16	580	4.440	....	....	....	....	....	...	....	....	....	....	....	....	....	....	108	31
31	6×6	4×4	5/8	899	2.130	....	....	....	....	....		....	....	....	....	....	....	....	....	167	31
31	7×7	5×5	3/4	1302	1.125	2.265	....	....	....			....	....	....	....	....	....	....	....	287	31
31	8×8	6×6	1	1798	0.645	0.129	2.580	3.885	....	....	....	....	....	....	....	....	....	....	....	596	31
31	9×9	7×7	1-1/8	2356	0.390	0.795	1.590	2.535	3.210	....	....	....	....	....	....	....	....	....	....	862	31
31	10×10	8×8	1-1/4	2976	0.277	0.555	1.110	1.665	2.370	2.775	3.337	....	....	....	....	....	....	....	....	1197	31
31	11×11	9×9	1-3/8	3658	0.190	0.381	0.765	1.140	1.665	1.905	2.280	2.667	3.330	....	....	....	....	....	....	1609	31
31	12×12	10×10	1-1/2	4418	0.133	0.267	0.540	0.795	1.065	1.335	1.770	1.870	2.145	2.403	2.805	....	....	....	....	2107	31
31	13×13	11×11	1-5/8	5247	0.094	0.190	0.375	0.570	0.780	0.954	1.161	1.335	1.560	1.719	2.010	2.331	2.580	....	....	2697	31
31	14×14	12×12	1-3/4	6154	0.070	0.141	0.300	0.420	0.600	0.708	0.849	1.005	1.200	1.273	1.475	1.800	1.905	2.124	2.340	3388	31
31	15×15	13×13	1-7/8	7130	0.054	0.109	0.225	0.330	0.435	0.549	0.660	0.765	0.891	0.990	1.155	1.314	1.435	1.647	1.815	4190	31
32	5×5	3×3	9/16	500	5.040	....	....	....	....	....	....	....	....	....	....	....	....	....	....	103	32
32	6×6	4×4	5/8	928	2.415	....	....	....	....	....	....	....	....	....	....	....	....	....	....	159	32
32	7×7	5×5	3/4	1344	1.312	2.625	....	....	....	....	....	....	....	....	....	....	....	....	....	276	32
32	8×8	6×6	1	1856	0.750	1.500	3.000	4.500	....	....	....	....	....	....	....	....	....	....	....	571	32
32	9×9	7×7	1-1/8	2432	0.484	0.969	1.935	2.895	3.720	....	....	....	....	....	....	....	....	....	....	827	32
32	10×10	8×8	1-1/4	3072	0.315	0.630	1.260	1.890	2.520	3.151	....	....	....	....	....	....	....	....	....	1148	32
32	11×11	9×9	1-3/8	3776	0.214	0.429	0.855	1.287	1.710	2.158	2.574	3.003	....	....	....	....	....	....	....	1543	32
32	12×12	10×10	1-1/2	4560	0.151	0.303	0.606	0.909	1.212	1.515	1.818	2.121	2.424	2.727	....	....	....	....	....	2020	32
32	13×13	11×11	1-5/8	5416	0.108	0.216	0.432	0.648	0.864	1.081	1.296	1.512	1.728	1.944	2.269	2.592	5	....	....	2586	32
32	14×14	12×12	1-3/4	6452	0.082	0.163	0.327	0.490	0.654	0.817	0.981	1.144	1.308	1.471	1.716	1.962	2.200	2.452	2.700	3250	32
32	15×15	13×13	1-7/8	7360	0.062	0.124	0.249	0.373	0.498	0.622	0.747	0.871	0.996	1.120	1.306	1.479	1.68	1.867	2.055	4018	32

33	5× 5	3× 3	9/16	619	5.670	....	....	....	....	....	....	....	....	....	....	....	....	....	....	100	33
33	6× 6	4× 4	5/8	957	2.725	....	....	....	....	....	....	....	....	....	....	....	....	....	....	153	33
33	7× 7	5× 5	3/4	1386	1.455	2.925	....	....	....	....	....	....	....	....	....	....	....	....	....	261	33
33	8× 8	6× 6	1	1914	0.825	1.665	....	....	....	....	....	....	....	....	....	....	....	....	....	549	33
33	9× 9	7× 7	1-1/8	2508	0.540	1.080	....	....	....	....	....	....	....	....	....	....	....	....	....	794	33
33	10×10	8× 8	1-1/4	3168	0.354	0.693	1.386	2.079	2.772	3.540	....	....	....	....	....	....	....	....	....	1078	33
33	11×11	9× 9	1-3/8	3894	0.243	0.486	0.970	1.456	1.941	2.428	2.763	3.390	....	....	....	....	....	....	....	1480	33
33	12×12	10×10	1-1/2	4702	0.169	0.339	0.678	1.017	1.356	1.704	2.034	2.373	2.712	3.051	....	....	....	....	....	1940	33
33	13×13	11×11	1-5/8	5577	0.121	0.243	0.486	0.729	0.972	1.215	1.458	1.701	1.943	2.178	2.547	2.916	....	....	....	2480	33
33	14×14	12×12	1-3/4	6550	0.093	0.184	0.369	0.553	0.738	0.924	1.107	1.291	1.476	1.660	1.939	2.220	2.490	2.775	....	3122	33
33	15×15	13×13	1-7/8	7590	0.070	0.141	0.297	0.420	0.561	0.702	0.840	0.981	1.122	1.260	1.470	1.680	1.890	2.100	2.310	3860	33
34	5× 5	3× 3	9/16	637	6.370	....	....	....	....	....	....	....	....	....	....	....	....	....	....	96	34
34	6× 6	4× 4	5/8	986	3.075	....	....	....	....	....	....	....	....	....	....	....	....	....	....	147	34
34	7× 7	5× 5	3/4	1428	1.560	3.120	....	....	....	....	....	....	....	....	....	....	....	....	....	255	34
34	8× 8	6× 6	1	1922	0.900	1.800	3.600	....	....	....	....	....	....	....	....	....	....	....	....	527	34
34	9× 9	7× 7	1-1/8	2584	0.600	1.200	2.400	3.600	....	....	....	....	....	....	....	....	....	....	....	764	34
34	10×10	8× 8	1-1/4	3264	0.397	0.795	1.590	2.385	3.180	3.970	....	....	....	....	....	....	....	....	....	1037	34
34	11×11	9× 9	1-3/8	4012	0.271	0.543	1.086	1.629	2.172	2.710	3.258	3.810	....	....	....	....	....	....	....	1427	34
34	12×12	10×10	1-1/2	4845	0.192	0.384	0.768	1.140	1.536	1.920	2.295	2.685	3.072	3.450	....	....	....	....	....	1868	34
34	13×13	11×11	1-5/8	5755	0.136	0.273	0.546	0.822	1.092	1.371	1.644	1.918	2.184	2.467	2.862	3.288	....	....	....	2400	34
34	14×14	12×12	1-3/4	6749	0.103	0.207	0.399	0.621	0.799	1.030	1.237	1.449	1.650	1.860	2.175	2.475	2.790	3.100	....	3005	34
34	15×15	13×13	1-7/8	7820	0.079	0.159	0.319	0.480	0.639	0.780	0.960	1.125	1.275	1.440	1.680	1.920	2.160	2.400	2.640	3715	34
35	5× 5	3× 3	9/16	656	7.120	....	....	....	....	....	....	....	....	....	....	....	....	....	....	93	35
35	6× 6	4× 4	5/8	1015	3.420	....	....	....	....	....	....	....	....	....	....	....	....	....	....	142	35
35	7× 7	5× 5	3/4	1470	1.845	3.690	....	....	....	....	....	....	....	....	....	....	....	....	....	245	35
35	8× 8	6× 6	1	2030	1.059	2.110	4.230	....	....	....	....	....	....	....	....	....	....	....	....	508	35
35	9× 9	7× 7	1-1/8	2660	0.684	1.365	2.730	4.110	....	....	....	....	....	....	....	....	....	....	....	736	35
35	10×10	8× 8	1-1/4	3360	0.445	0.885	1.770	2.655	3.540	4.455	....	....	....	....	....	....	....	....	....	999	35
35	11×11	9× 9	1-3/8	4130	0.304	0.610	1.230	1.845	2.460	3.060	3.690	4.305	....	....	....	....	....	....	....	1375	35
35	12×12	10×10	1-1/2	4988	0.213	0.420	0.840	1.260	1.680	2.145	2.520	2.940	3.360	3.780	....	....	....	....	....	1800	35
35	13×13	11×11	1-5/8	5924	0.151	0.300	0.600	0.900	1.200	1.530	1.800	2.100	2.400	2.700	3.180	....	....	....	....	2304	35
35	14×14	12×12	1-3/4	6948	0.115	0.210	0.420	0.630	0.840	1.155	1.260	1.470	1.680	1.890	2.430	2.520	3.225	....	....	2895	35
35	15×15	13×13	1-7/8	8050	0.087	0.180	0.360	0.540	0.720	0.885	1.080	1.260	1.440	1.620	1.845	2.160	2.370	2.490	2.730	3580	35

form throughout the length of the pole, but may be stepped off as the top of the pole is approached; a pole with uniform reinforcement will break at the ground line, while one with tapered reinforcement will break at some point above the ground, depending upon the

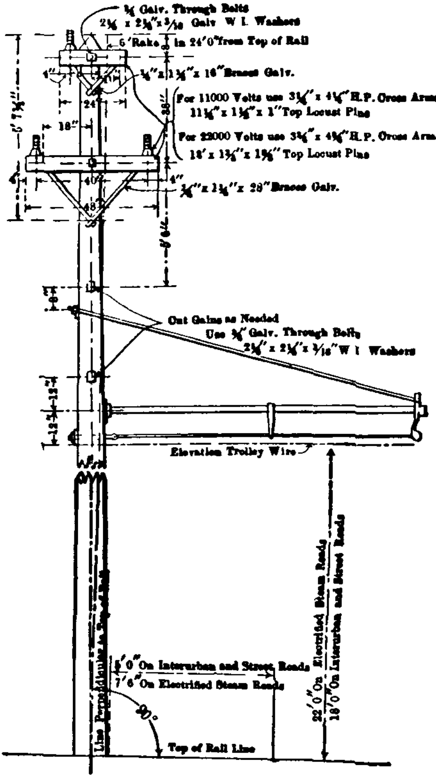


FIG 35 —Typical pole construction, trolley and one transmission line.

taper of the reinforcement; a concrete pole has an element of safety in that a failure of the pole will not in general allow it to fall to the ground; it is difficult at times to pull over a pole after failure, ever though it is inclined at a large angle from the vertical.

Typical Railway Transmission Line Construction. Figs. 35 to 38, inclusive, show typical 11,000-volt, and 22,000-volt pole line constructions from the practice of the Connecticut Company.

National Electrical Safety Code. The U S. Bureau of Standards has prepared and published a National Electrical Safety Code, consisting of a set of rules for the safe construction and mainte-

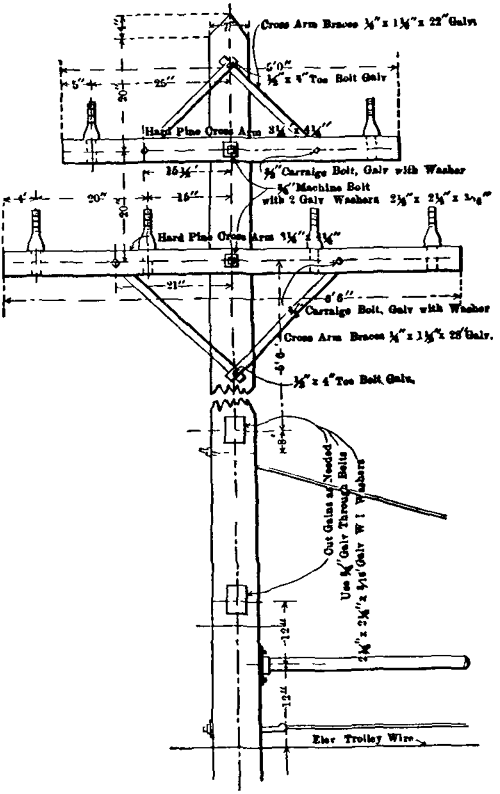


FIG. 36—Typical pole construction, trolley and two 11,000-volt transmission lines.

nance of electrical apparatus and lines. In preparation and revision of this code the Bureau had the cooperation and assistance of many State industrial and public service commissions, municipal electrical inspectors, engineers of operating and manufacturing companies, committees of engineering societies (including the Amer. Elec. Ry. Eng. Assn.) and representatives of the fire and casualty insurance interests, and of the electrical workers. The rules of the code are divided into four parts. Part 1 refers to the installation and operation of machinery, switchboards, etc., in central stations and substations. Part 2 contains rules for the installation and main-

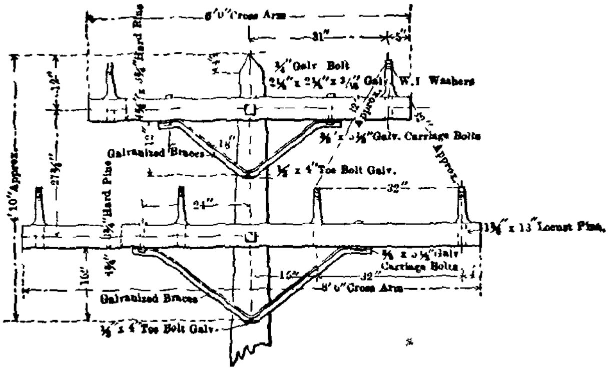


FIG. 37.—Typical pole-top construction, two 22,000-volt transmission lines.

tenance of electrical supply and signal lines, both overhead and underground. Part 3 contains rules for the installation and operation of electrical apparatus and wiring in factories, offices, residences, or other places where electric light and power are

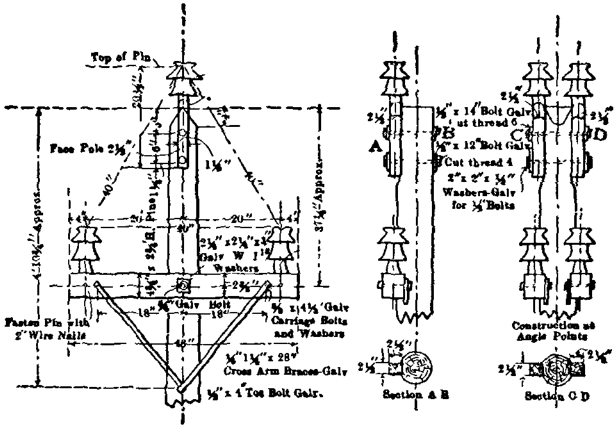


FIG. 38.—Typical pole-top construction, one 22,000-volt line.

utilized. Part 4 contains rules for safeguarding employees working about electric machinery or lines. A supplementary section gives rules for protective grounding of circuits and equipment.

This code, while prepared for the primary purpose of insuring safety, includes detailed data and descriptions of the best current practice in construction and maintenance. Part 2, referring to overhead and underground lines, is of especial value in including general rules for pole lines, clearances and separation of wires, calculations of stresses, loads and strengths in connection with poles, towers and lines, rules for crossings of overhead lines of various classes and quite complete tables of sags, stresses, tensions and loadings of copper, iron and steel wires, etc.

Joint Use of Wood Poles. A Committee with representatives from the Amer. Elec. Ry. Assn , Nat. Elec. Lt. Assn., Amer. Inst. Elec. Engrs., and conferring with representatives of the telephone companies, prepared in 1914, Specifications for the Joint Use of Wood Poles, which specifications may be found in the Manual of the Amer. Elec. Ry. Eng. Assn. Relative locations of wires of various classifications are specified, together with vertical and lateral spacings, length of spans, location for transformers and fixtures, guy arrangement, etc.

Electric Wire and Cable Terminology, A.E.R.E.A. Standard.

NOTE. Same as recommended by U. S. Bureau of Standards, circular No. 37.

1. *Wire.* A slender rod or filament of drawn metal.

The definition restricts the term to what would ordinarily be understood by the term "solid wire." In the definition, the word "slender" is used in the sense that the length is great in comparison with the diameter. If a wire is covered with insulation, it is properly called an insulated wire; while primarily the term "wire" refers to the metal, nevertheless when the context shows that the wire is insulated the term "wire" will be understood to include the insulation.

2. *Conductor.* A wire or combination of wires not insulated from one another, suitable for carrying a single electric current.

The term "conductor" is not to include a combination of conductors insulated from one another, which would be suitable for carrying several different electric currents. Rolled conductors (such as bus bars) are, of course, conductors, but are not considered under the terminology here given.

3. *Stranded Conductor.* A conductor composed of a group of wires or any combination of groups of wires.

The wires in a stranded conductor are usually twisted or braided together.

4. *Cable.* (1) A stranded conductor (single-conductor cable); or (2) a combination of conductors insulated from one another (multiple-conductor cable).

The component conductors of the second kind of cable may be either solid or stranded, and this kind of cable may or may not have a common insulating covering. The first kind of cable is a single conductor, while the second kind is a group of several conductors. The term "cable" is applied by some manufacturers to a solid wire heavily insulated and lead covered; this usage arises from the manner of the insulation, but such a conductor is not included under this definition of "cable." The term "cable" is a general one and in practice it is usually applied only to the larger sizes. A small cable is called a "stranded wire" or a "cord," both of which are defined above. Cables may be bare or insulated, and the latter may be armored with lead or with steel wires or bands.

5. *Strand.* One of the wires or groups of wires of any stranded conductor.

6. *Stranded Wire.* A group of small wires, used as a single wire.

A wire has been defined as a slender rod or filament of drawn metal If such a filament is subdivided into several smaller filaments or strands and is used as a single wire, it is called "stranded wire." There is no sharp dividing line of size between a "stranded wire" and a "cable" If used as a wire, for example, in winding inductance coils or magnets, it is called a stranded wire and not a cable. If it is substantially insulated, it is called a "cord," defined below.

7. *Cord.* A small cable, very flexible and substantially insulated to withstand wear.

There is no sharp dividing line in respect to size between a "cord" and a "cable," and likewise no sharp dividing line in respect to the character of insulation between a "cord" and a "stranded wire" Usually the insulation of a cord contains rubber.

8. *Concentric Strand.* A strand composed of a central core surrounded by one or more layers of helically laid wires or groups of wires.

9. *Concentric Lay Cable.* A single conductor cable composed of a central core surrounded by one or more layers of helically laid wires.

10. *Rope Lay Cable.* A single conductor cable composed of a central core surrounded by one or more layers of helically laid groups of wires.

This kind of cable differs from the preceding in that the main strands are themselves stranded.

11. *N-Conductor Cable.* A combination of N conductors insulated from one another.

It is not intended that the name as here given be actually used. One would instead speak of a "3-conductor cable," a "12-conductor cable," etc. In referring to the general case, one may speak of a "multiple-conductor cable" (as in definition No. 4 above).

12. *N-Conductor Concentric Cable.* A cable composed of an insulated central conducting core with $(N-1)$ tubular stranded conductors laid over it concentrically and separated by layers of insulation.

Usually only 2-conductor or 3-conductor. Such conductors are used in carrying alternating currents. The remark on the expression "N-conductor" given for the preceding definition applies here also.

13. *Duplex Cable.* Two insulated single-conductor cables twisted together.

They may or may not have a common insulating covering.

14. *Twin Cable.* Two insulated single-conductor cables laid parallel, having a common covering.

15. *Triplex Cable.* Three insulated single-conductor cables twisted together.

They may or may not have a common insulating covering.

16. *Twisted Pair.* Two small insulated conductors twisted together, without a common covering.

The two conductors of a "twisted pair" are usually substantially insulated, so that the combination is a special case of a "cord."

17. *Twin Wire.* Two small insulated conductors laid parallel, having a common covering.

Rubber Insulated Wire and Cable for Power Distribution

(A.E.R E.A. Standard)

Grades. Two grades of rubber insulation for distribution wires and cables are designated by the A.E.R.E A as "Grade A Rubber Insulation" and "Grade B Rubber Insulation." Grade A is a high grade insulation for use on circuits of working pressure not exceeding 15,000 volts. Grade B is for use where a cheaper insulation is desired and should not be used on circuits of working pressure in excess of 7000 volts.

Conductor Size and Stranding. The combined area of the wires when laid out straight and measured at right angles to their axes should not be less than the specified gage or circular mils. The stranding should be concentric and in accordance with the following table:

Size	Cables for aerial use	Cables for other than aerial use
2,000,000 c m	91	127
1,500,000 c.m	61	91
1,000,000 c.m	61	61
600,000 c m	37	61
500,000 c m	37	37
400,000 c.m.	19	37
0000 A.W.G	19	19
00 A W.G	7	19
2 A.W.G	7	7
7 and smaller		7

For intermediate sizes, use stranding for next larger size.

Conductor Material. The conductor should be of annealed copper, conforming to the standard specifications of the American Society for Testing Materials. It should be provided with a heavy uniform coating of commercially pure tin, without projections. The tinning may be tested as follows: Samples of the wire should be thoroughly cleaned with alcohol and immersed in hydrochloric acid of specific gravity 1.088 at 15.5 deg. C. (60 deg F.) for 1 minute. They should then be rinsed in pure water and immersed in an aqueous solution of sodium sulphide of specific gravity 1.142 for 30 seconds and again washed. This operation should be repeated three times. At the end of the fourth immersion in sodium sulphide the wire should show no sign of blackening. The sodium sulphide solution should contain an excess of sulphur and should have sufficient strength to thoroughly blacken a piece of untinned copper wire in 5 seconds.

Separator. The separator may consist of soft cotton yarn (which may be braided), or of paper or muslin tape. The separator should allow the insulation sufficient contact with the conductor to prevent the conductor sliding in the insulation.

Insulation Thickness. The thickness of insulation for potentials up to 7000 volts should be the minimum allowed by the National Board of Fire Underwriters, as summarized in the following table:

A.W.G. (B. & S.)	Working voltage					
	0–600	1500	2500	3500	5000	7000
	Thickness of insulation, 64ths of one inch					
14	3	4	6	8	12	16
12	3	4	6	8	12	16
10	3	4	6	8	12	16
8	3	4	6	8	12	16
6	4	5	6	8	12	16
4	4	5	6	8	12	16
2	4	5	6	8	12	16
1	5	6	7	8	12	16
0	5	6	7	8	12	16
00	5	6	7	8	12	16
000	5	6	7	8	12	16
0000	5	6	7	8	12	16
Cir. mils						
225,000	6	7	8	9	12	16
300,000	6	7	8	9	12	16
400,000	6	7	8	9	12	16
500,000	6	7	8	9	12	16
600,000	7	8	9	10	12	16
700,000	7	8	9	10	12	16
800,000	7	8	9	10	12	16
900,000	7	8	9	10	12	16
1,000,000	7	8	9	10	12	16
1,250,000	8	9	10	11	14	18
1,500,000	8	9	10	11	14	18
1,750,000	8	9	10	11	14	18
2,000,000	8	9	10	11	14	18

(For higher voltages, the thickness of insulation will depend upon the conditions of service.)

Insulation Repairs and Joints. If exigencies of manufacture require repairs or joints in the insulation, the work should be done in such a way as to leave the repaired part or the joint and all parts affected by it as strong and durable electrically as the remainder of the insulation. The repairs or joints should be properly vulcanized in a mold of approximately the same diameter as the remainder of the insulation.

Insulation rubber compound should be thoroughly and properly vulcanized. The vulcanized insulation should be homogeneous in character, tough and elastic and should be concentric about the conductors and fit tightly thereto. All laps or seams in the insulation should be as strong mechanically and electrically as the rest of the insulation.

"*Grade A Rubber Insulation.*" (1) A 30 per cent fine Para or smoked first latex Hevea rubber compound with mineral base should be furnished. It should contain only the following ingredients: Rubber, sulphur, inorganic mineral matter, refined solid paraffin or ceresine.

(2) It should not contain either red lead or carbon.

(3) The vulcanized compound should conform to the following requirements, when tested by the procedure of the Joint Rubber Insulation Committee (see Proc. A.I.E.E., Jan , 1914).

(a) Results to be expressed as percentages by weight of the whole sample.

	Maximum	Minimum
Rubber	33 0	30
Waxy hydrocarbons	4 0	.
Free sulphur	0.7	

(b) Results to be taken between the limits given in proportion to the percentage by weight of rubber found:

Limits allowed for 30 per cent rubber compound:

Saponifiable acetone extract	1 35	0.55
Unsaponifiable resins	0 45	
Chloroform extract	0 90	.
Alcoholic potash extract	0 55	
Specific gravity	.	1 75

Limits allowed for 33 per cent rubber compound:

Saponifiable acetone extract	1 50	0 60
Unsaponifiable resins	0 50	
Chloroform extract	1.00	
Alcoholic potash extract	0 60	
Specific gravity	. . .	1 67

(4) The acetone solution should not fluoresce.

(5) The acetone extract (60 c.c.) should not be darker than a light straw color.

(6) Hydrocarbons should be solid, waxy and not darker than a light brown.

(7) Chloroform extract (60 c.c.) should not be darker than a straw color.

(8) Failure to meet any requirement of this specification should be sufficient cause for rejection

(9) Contamination of the compound, such as by the use of impregnated tapes, should not excuse a contractor from conforming to this specification.

"*Grade B Rubber Insulation.*" The vulcanized rubber compound should show, when analyzed by the procedure hereinafter specified, not less than 27 per cent of rubber gum.

Five chemical tests should be made of the vulcanized rubber compound as follows. Acetone extract, alcoholic potash extract, chloroform extract, ash, total sulphur

The sum total of the results of these five tests should not exceed 73 per cent by weight of the total compound.

The ash test should be supplemented by tests to determine the quantity of substances other than vulcanized rubber which are combustible but not soluble in acetone, alcoholic potash, or chloroform, and any such substance, if any, should be counted as ash.

The tests should be made in accordance with the Underwriters Laboratories' specification.

Contamination of the compound, such as by the use of impregnated tapes, should not excuse a contractor from conforming to this specification.

Tensile Strength and Elongation of Rubber Insulation. A sample which may be of entire, segmental, or approximately rectangular cross-section, should be cut from the insulated conductor by means of a sharp knife. The sample should be bent in every direction to magnify and reveal any surface incision or imperfection which may exist. A portion of the sample without such defects and having a free length of not less than 2 in. should then be stretched at the rate of 12 in. per minute until it breaks; another sample should be stretched to three times its length and immediately released. Marks should be placed on the insulation before its removal from the conductor. The compound should conform to the following limits:

Property	Thickness			
	12/64 inches or less		Over 12/64 inches	
	Maximum	Minimum	Maximum	Minimum
Tensile strength, lb. per sq. in. (each sample)	....	1000	....	1000
Elongation in 2-in. length at break (times original length)	....	5	....	4
Elongation 5 seconds after release when stretched at the rate of 12 in. per minute to three times its length.	20%	....	20%	....

Electrical Tests. Each length of conductor insulated with the compound should be tested for dielectric strength and insulation resistance before the application of any outer covering other than tape or braid, after at least 12 consecutive hours' submersion in water and while still submersed. The conditions and conduct of these tests should conform to the Standardization Rules of the Amer. Inst. of Elec. Engrs.

Dielectric Strength. An alternating-current voltage of not less than twenty-five nor more than one hundred cycles and approximately as closely as possible to a sine wave should be applied between each conductor and the water for a period of 5 minutes. For a 30-minute test, 80 per cent of the specified voltage should be used. Lead sheathed cables after being finished should have the test voltage applied between their conductors and also between all of the conductors and the sheath. The voltage to be applied should be in accordance with the table on page 617. The conductor must not show any weakening of its insulation or any other injury under this test, which is to be made before the test for insulation resistance.

Insulation Resistance. The insulation resistance should be measured after the high voltage test and following a 1-minute

TEST VOLTAGES, KILOVOLTS FOR 5-MINUTE TESTS

Use 100 per cent of the following voltages for Grade A
Use 80 per cent of the following voltages for Grade B

Size of conductors	Thickness of insulation, 64ths. of one inch																
	2	3	4	5	6	7	8	10	12	14	16	18	20	22	24	26	28
2,000,000 cir. mils	....	....	....	....	....	....	5.0	10.5	19	22	24	26	28	30	31	33	34
1,750,000 cir. mils	....	....	....	....	....	....	6.5	11.5	19	22	24	26	28	30	31	33	34
1,500,000 cir. mils	....	....	....	....	....	....	7.0	12.0	19	22	24	26	28	30	31	33	34
1,250,000 cir. mils	....	....	....	....	....	....	7.5	12.5	19	22	24	26	28	30	31	33	34
1,000,000 cir. mils	....	....	....	....	....	5.5	8.0	13.0	19	22	24	26	28	30	31	33	34
750,000 cir. mils	....	....	....	....	....	6.5	9.0	14.0	19	22	24	26	28	30	31	33	34
500,000 cir. mils	....	....	....	2.5	5.0	7.5	10.0	14.5	19	22	24	26	28	30	31	33	34
250,000 cir. mils	....	....	....	4.0	6.5	9.0	11.0	15.5	19	22	24	26	28	30	31	33	34
0000 A.W.G.	....	....	....	4.5	7.0	9.0	11.5	15.5	18	21	23	25	27	28	30	31	32
000 A.W.G.	....	....	....	5.0	7.5	9.5	11.5	15.5	18	21	23	25	27	28	30	31	32
00 A.W.G.	....	....	....	5.0	7.5	9.5	11.5	15.5	18	21	23	25	27	28	30	31	32
0 A.W.G.	....	....	....	5.5	8.0	10.0	12.0	15.5	18	21	23	25	27	28	30	31	32
1 A.W.G.	....	....	4.0	6.0	8.0	10.0	12.0	15.5	18	21	23	25	27	28	30	31	32
2 A.W.G.	....	....	4.0	6.0	8.0	10.0	12.0	15.0	17	19	22	24	25	26	28	29	30
4 A.W.G.	....	....	4.5	6.5	8.5	10.0	11.5	14.5	17	19	22	24	25	26	28	29	30
6 A.W.G.	....	....	5.0	6.5	8.5	10.0	11.5	14.0	16	18	20	22	23	25	26	28	29
8 A.W.G.	....	3.0	5.0	7.0	8.0	9.5	11.0	13.0	....	....	....	....	....	....	....	....	....
10 A.W.G.	....	3.0	5.0	6.5	8.0	8.5	10.0	12.0	....	....	....	....	....	....	....	....	....
12 A.W.G.	....	2.5	5.0	6.0	7.5	8.5	9.5	11.5	....	....	....	....	....	....	....	....	....
14 A.W.G.	....	2.5	5.0	6.0	7.0	8.0	9.0	11.0	....	....	....	....	....	....	....	....	....
16 A.W.G.	1.0	2.5	4.5	5.5	6.5	7.5	8.5	10.5	....	....	....	....	....	....	....	....	....
18 A.W.G.	1.0	2.5	4.5	5.5	6.5	7.5	8.5	10.5	....	....	....	....	....	....	....	....	....

(For intermediate sizes use the voltages corresponding to the next larger size, having the same thickness of insulation.)
(For a 30-minute test use 80 per cent of value for 5-minute test.)

oughly saturated with a dense waterproof compound having the properties hereinafter specified. The impregnated braids should be uniformly covered with a continuous layer of suitable weatherproof compound which should adhere firmly to the braids. The outer surface should be thoroughly slicked down.

Tape. The tape should be of cotton thoroughly saturated with a waterproof rubber compound and of proper weight and width.

Weatherproof Compound. The weatherproof compound should be such as to have no injurious action upon the insulation of the braid and it should be insoluble in water; it should be sufficiently elastic between the temperature of 4.5 deg. C. (40 deg. F.) and 32 deg. C. (90 deg. F.) so that the completed wire can be bent to a radius of ten times its outside diameter without showing cracks in the finished surface of the wire or squeezing out of the weatherproof compound. The weatherproof compound should be of such nature that it will not crack when the finished wire is subjected to a temperature of −23.3 deg. C. (−10 deg. F.), and that it will meet the requirements when the finished wire is subjected to the following melting test.

Melting Test for Weatherproof Compound. Short samples of the finished conductor should be placed on a piece of clean white paper in an oven and should be subjected to a temperature of 52 deg. C. (125 deg. F.) for 1/2 hour. The compound should not become sufficiently fluid to form a ridge upon the paper perceptible to the fingers, or, in case the compound should be absorbed by the paper, to show a greasy or oily spot upon the paper, nor should the compound show a tendency to flow toward the bottom of the conductor, thus exposing the cotton fiber of the braid at the top.

Application of Braid and Tape. If the conductor is to be finished with a braid, all conductors smaller than No. 8 A.W.G. should be double braided, No. 8 and larger to be double or single braided as required. The outside braids should be not less than 1/32 in. thick and the inside braid not less than 1/64 in. thick. Single braid should be not less than 1/32 in. thick. When multiple conductor cables are specified, the rubber insulation on each conductor should be covered with a layer of tape overlapping not less than one-quarter of its width. In the case of two conductor cables the separate conductors thus finished should be laid side by side and an outer covering of single braid not less than 1/32 in. thick should be then applied. In the case of three or more conductors the taped conductors should be twisted together with a suitable lay of at least one complete twist in 24 in., the interstices being filled with sufficient jute to make the core of circular cross-section. Over this core should then be applied a tape with an overlap of at least 1/8 in. and the outer covering of braid not less than 1/32 in. should then be applied.

Lead Cable Sheath

Composition and Thickness of Lead Cable Sheath. If the wire or cable is to be furnished with a lead sheath instead of an outer covering of braid, as provided in the preceding paragraph, there should be tightly formed about the core a lead sheath of uniform thickness not less than that indicated in the following table:

Diameter of core, in.	Thickness of sheath
0–0.299	4/64 in.
0.300–0.699	5/64 in.
0.700–1.249	6/64 in.
1.250–1.999	7/64 in.
2.000–2.699	8/64 in.
2.700 and over	9/64 in.

A sheath having an internal diameter of less than 2 in. should consist of commercially pure lead; a sheath having an internal diameter of 2 in. or more should consist of an alloy containing not less than 98 per cent of commercially pure lead and not less than 1 per cent of commercially pure tin.

Cable Armor

Protection Under Armor. Rubber insulated cable covered with tape, braid, or other suitable protection, or the lead sheath should be run through a hot asphalt compound, served with a layer of jute yarn, run through hot asphalt again, and then laid with galvanized wire armor.

Size of Armor Wire. The proper size of armor wire will depend upon the conditions of service; the latitude allowed in the following table represents the difference arising from such difference in service conditions. The armor wire should be the minimum size.

Diameter of cable under jute bedding, inches	Armor wire, U. S. (steel) W.G.	Jute bedding under armor measured in finished cable
0.00–0.50	14–13	1/16 in. minimum
0.44–0.69	12	1/16 in. minimum
0.63–1.00	10	1/16 in. minimum
0.88–1.50	8	1/16 in. minimum
1.25–2.00	6	1/16 in. minimum
1.30–larger	4	1/16 in. minimum

Lay of Armor. The armor should be applied closely without appreciable space between adjacent wires. The lay should be from eight to twelve times the pitch diameter. Successive layers of jute or jute and armor should be laid in opposite directions. The direction of lay is the lateral direction in which the wires run over the top of the cable as they recede from an observer looking along the axis of the cable.

Finish of Armored Cable. If an outer covering is required, the armored cable should be run through hot asphalt compound, served with a layer of the best three-ply 14-lb. hard twisted jute yarn in a close short lay, run through hot asphalt compound, then served with a second layer of three-ply 14-lb. jute yarn, run through hot asphalt compound and finally run through some material to prevent sticking.

Material of Armor Wire. All armor wire should consist of galvanized mild steel wire of uniform diameter, free from all cracks, splits, or other flaws; it should be of such softness that it will easily take a permanent set and give great pliability to the cable, permitting it to be coiled evenly and smoothly and showing no tendency to rise or spring in the flakes.

MEGOHM-MILES AT 60 DEG F., BASED UPON THE CONSTANT 4000 1-MINUTE ELECTRIFICATION
Minimum shall be 100 per cent of following for Grade A and 80 per cent for Grade B

Size of conductors	Thickness of insulation, 64ths. of one inch																
	2	3	4	5	6	7	8	10	12	14	16	18	20	22	24	26	28
2,000,000 cir. mils.							200	275	325	425							
1,750,000 cir. mils							275	300	375	450							
1,500,000 cir. mils							300	325	400	475							
1,250,000 cir. mils.							325	375	425	500	575						
1,000,000 cir. mils.						300	325	400	475	550	625						
750,000 cir. mils.						325	400	475	525	625	700	775					
500,000 cir. mils.				300	375	400	475	575	675	750	850	900					
350,000 cir. mils.				350	425	475	525	675	775	850	950	1050	1125				
250,000 cir. mils.				400	475	575	625	725	875	1000	1100	1200	1275				
0000 A.W.G.				450	550	600	650	800	950	1050	1150	1250	1350	1450			
000 A.W.G.				500	600	650	750	850	1000	1150	1250	1350	1450	1550			
00 A.W.G.				550	650	750	850	950	1050	1250	1350	1500	1600	1700	1800		
0 A.W.G.				600	700	800	950	1050	1200	1350	1500	1600	1700	1800	1900	2000	
1 A.W.G.			600	650	750	850	1000	1150	1350	1450	1600	1750	1850	1950	2050	2150	2250
2 A.W.G.			650	750	850	950	1050	1250	1450	1600	1700	1850	1950	2100	2200	2300	2400
4 A.W.G.			750	850	1000	1150	1250	1450	1650	1850	2000	2150	2250	2400	2500	2600	2700
6 A.W.G.			850	1050	1200	1350	1450	1750	1950	2100	2300	2450	2550	2700	2800	2900	3000
8 A.W.G.		850	1050	1250	1450	1650	1750	2050	2200	2400	2600	2750	2900	3050	3150	3250	3350
10 A.W.G.		1150	1350	1600	1800	2000	2150	2400	2650	2900	3100	3250	3400	3550	3700	3800	3900
12 A.W.G.		1350	1600	1850	2020	2250	2400	2750	3000	3250	3400	3600	3750	3900	4050	4150	4300
14 A.W.G.		1550	1850	2150	2350	2550	2650	3050	3350	3550	3800	3950	4100	4300	4400	4550	4650
16 A.W.G.	1400	1800	2150	2400	2650	2850	3050	3400	3650	3950	4150	4350	4500	4660	4650	4900	5050
18 A.W.G.	1600	2050	2450	2800	3000	3200	3400	3750	4050	4300	4500	4700	4900	5000	5150	5300	5400

electrification with a continuous e m.f. of not less than 100 nor more than 500 volts, and the results corrected to the standard temperature at 15 5 deg C. (60 deg. F). All tests for insulation resistance should be made at a temperature within 10 deg C. (18 deg F) of this standard temperature. The insulation resistance of each conductor of multiple conductor cables should be the insulation resistance measured between each conductor and all the other conductors connected to the sheath or water. The insulation resistance should be not less than given in the table on page 618

The insulation resistance at any given temperature should be reduced to that at 15 5 deg C. (60 deg F) by multiplying by the coefficient in the following tables corresponding to that temperature for Grade A and Grade B insulation:

Temperature, deg C.	Coefficient	Temperature, deg. C.	Coefficient
7	0 67	16	1 02
8	0 70	17	1 07
9	0 74	18	1 12
10	0 77	19	1 18
11	0 81	20	1 23
12	0 85	21	1 30
13	0 89	22	1 35
14	0 93	23	1 42
15	0 98	24	1 49
15 5	1 00	25	1 55

Temperature, deg F	Coefficient	Temperature, deg F	Coefficient
46	0 69	60	1 00
47	0 71	61	1 03
48	0 73	62	1 05
49	0 75	63	1 08
50	0 77	64	1 11
51	0 79	65	1 14
52	0 81	66	1 17
53	0 83	67	1 20
54	0 85	68	1 23
55	0 88	69	1 26
56	0 91	70	1 30
57	0 92	71	1 33
58	0 95	72	1 37
59	0 97	73	1 40
60	1 00	74	1 44
		75	1 48

Weatherproof Braid

Weatherproof braid should consist of the required number of fine, smooth, closely woven braids, all of which must be thor-

Tensile Strength of Armor Wire. Armor wire should have a tensile strength of not less than 60,000 lb. per square inch of cross-section and an elongation of not less than 10 per cent in 10 in.

Flexibility of Armor Wire. The armor wire should admit of bending around a spindle of ten times the diameter of the wire and back again without developing cracks of the galvanizing which are visible to the naked eye.

Steel Tape Armor. If the cable is to be armored with steel tape the core covered with tape braid or other suitable protection should be run through a bath of hot asphalt compound, served with a layer of 14-lb. jute yarn, spun on with a close short lay, run through hot asphalt compound, armored with a steel tape; armored with second steel tape; run through hot asphalt compound, served with a layer of 100-lb. jute yarn with a close short lay, run through hot asphalt compound and finished by running through some material to prevent sticking. The layers of jute should be applied in the reverse directions. The space between adjacent turns of steel tape should not exceed one-tenth the width of the steel tape.

Thickness of Armor and Jute under Armor. The tape and jute under armor, after armoring, should conform to the following table:

Cable diameter before armoring, inches	Maximum width steel tape, inches	Minimum thickness each tape, inches	Minimum thickness under armor, inches
0 45	0 50	0 02	0 06
0 46–0 75	0 75	0 02	0 06
0 76–1 00	1 00	0 03	0 07
1 01–1 40	1 25	0 03	0 07
1 41–1 70	1 50	0 04	0 08
1 71–2 00	1 75	0 04	0 08
2 01–over	2 00	0.05	0 09

Single Conductor, Paper Insulated, Lead Covered Cable for 1200 Volts. The cable is composed of a single conductor, covered with paper tape to the required thickness. The paper, soaked in a suitable insulating compound, is covered with a lead sheath of the required thickness. The completed cable and the material of which it is made should conform to the requirements of the following.

Conductor. The conductor should consist of annealed copper wires, free from splints, flaws, or other defects, concentrically stranded together in reverse spiral layers, and having an aggregate cross-sectional area when measured at right angles to the axes of the individual wires at least equal to the area of the specified size, and in accordance with the following table:

Size of conductor	No. of wires
0 162 in.–0 204 in diameter	1
0 229 in.–0.258 in diameter	7
0.289 in.–0 460 in diameter	19
300,000–500,000 cir. mils	37
600,000–1,000,000 cir. mils	61
1,100,000–1,500,000 cir. mils	91
1 600,000–and larger cir. mils	127

Intermediate sizes take the stranding of the next larger listed size. Each of the individual wires should have a resistivity of not more than 888.55 ohms (mile, pound), at 20 deg C.

Insulation. The highest grade of pure manila rope paper tape should be applied helically and evenly to the conductor until the required thickness of paper is obtained. All moisture should be expelled from the paper by baking in suitable ovens; it should then be thoroughly saturated with the insulating compound of the required kind and quality. The paper tape must be applied to the conductor in such a manner, and the insulating compound must be of such a nature, that the cable will be capable of being bent to a radius of 12 in. when wound on reels and taken therefrom and put into place, at any temperature between 0 and 100 deg. F., without damage to the insulation or sheath. The insulating compound should contain no substance which will subject the paper or conductor to deterioration, and it should be of such composition as to maintain the insulation in a soft, plastic state at all seasons of the year.

Sheath. A sheath of uniform thickness, not less than 1/8 in. for cable under 1 1/2 in. core diameter, and not less than 9/64 in. for cable with core diameter 1 1/2 in. and larger, should be tightly formed about the core. A sheath having an internal diameter less than 1 1/2 in. should consist of commercially pure lead. A sheath having an internal diameter of 1 1/2 in. or more should consist of an alloy containing not less than 98 per cent of commercially pure lead and not less than 1 per cent of commercially pure tin.

Electrical Test. Each length of cable should be given the following dielectric strength and insulation resistance tests when mechanical operations of its construction are completed.

High Voltage Test. An alternating current of 5000 volts should be applied between the conductor and the lead sheath for a period of 5 minutes. The cable must not show any weakening of its insulation or any other injury under this test, which is to be made before the test for insulation resistance. The frequency of the test voltage should not exceed 100 cycles per second and should approximate as closely as possible to a sine wave. The initially applied voltage should not be greater than the working voltage, and the rate of increase should not be over 100 per cent in 10 seconds.

Insulation Resistance. Immediately after the dielectric test, a test should be made for insulation resistance. The measurement should be taken after 1 minute electrification, using an e.m.f. of not less than 100 volts, and should be not less than 50 megohms per mile at 60 deg. F

High Voltage, Three Conductor, Paper Insulated, Lead Covered Cable. The cable should be composed of three copper conductors, each covered with paper tape to the required thickness, and then twisted together with a suitable lay. The interstices should be rounded out with jute and the whole wrapped with a paper belt to the required thickness. The paper, soaked in a suitable insulating compound, should be covered with a lead sheath of the required thickness. The completed cable, and the materials of which it is made, should conform to the following.

Conductors. Each conductor should consist of annealed copper wire, free from splints, flaws, or other defects, concentrically stranded together in reverse spiral layers, and having an aggregate cross sectional area when measured at right angles to the axes of the individual wires at least equal to the area of the specified size and in accordance with the following table

Size of conductor	No. of wires
0.162 in.–0.204 in. diameter	1
0.229 in.–0.258 in. diameter.	7
0.289 in.–0.460 in. diameter	19

Each of the individual wires should have a resistivity of not more than 888.55 ohms (mile, pound), at 20 deg. C.

Insulation. The highest grade of pure manila rope paper tape should be applied helically and evenly to each of the conductors until the required thickness of paper is obtained. The conductors should then be twisted together, with a suitable lay of at least one complete twist in each 24 in. of length of cable. The interstices of the core so formed should be filled in with jute laterals, of the proper dimensions to form a true cylinder. The paper tape should then be applied helically and evenly over the core until the required thickness is obtained

All moisture should be expelled from the paper and jute by baking in suitable ovens, they should then be thoroughly saturated with the insulating compound. The paper tape should be applied to the conductors and the core in such a manner, and the insulating compound should be of such a nature, that the cable will be capable of being bent to a radius of 18 in. when wound on reels and taken therefrom and put into place, at any temperature between 0 and 100 deg. F., without damage to the insulation or sheath. The insulating compound should contain no substance which will subject the paper, jute or conductors to deterioration, and it should be of such composition as to maintain the insulation in a soft, plastic state at all seasons of the year.

Sheath. A sheath of uniform thickness, not less than that indicated in the following table, should be tightly formed about the core:

Diameter of core in mils	Thickness of sheath
Under 2000	1/8 in.
2000–2699.	9/64 in.
2700 and over	5/32 in.

A sheath having an internal diameter less than 2 in. should consist of commercially pure lead. A sheath having an internal diameter of 2 in. or more should consist of an alloy containing not less than 98 per cent of commercially pure lead, and not less than 1 per cent of commercially pure tin.

Electrical Test. Each length of cable should be given the following dielectric strength and insulation resistance tests when mechanical operations of its construction are completed. The conditions and conduct of the tests should conform to the recommendations of the Amer. Inst. of Elec. Engrs.

Dielectric Strength. An alternating current voltage should be applied between each conductor and all the others connected to the lead sheath, for a period of at least 5 minutes. The voltage to be applied should be that corresponding to the combined thickness of the conductor and belt insulation in accordance with the following table

Voltage	Thickness of insulation
5,000	9/64 in.
6,250	5/32 in.
7,500	3/16 in.
10,000	7/32 in.
12,500	1/4 in.
15,000	9/32 in.
20,000	5/16 in.
25,000	11/32 in.
30,000	3/8 in.
35,000	13/32 in.
40,000	7/16 in.
45,000	15/32 in.
50,000	1/2 in.
60,000	9/16 in.

The cable should not show any weakening of its insulation or any other injury under this test, which is to be made before the test for insulation resistance

Insulation Resistance Immediately after the dielectric test, a test should be made for insulation resistance, each conductor being measured against all the others connected to the lead sheath The measurement should be taken after 1 minute electrification, using an e m f. of not less than 100 volts, and should be not less than 50 megohms per mile at 60 deg.

Duct Conduit System

Tile Duct. (Fig. 39.) Each piece of tile duct should be of thoroughly vitrified and glazed tile, whole, sound, straight from end to end, with smooth interior surface, free from blisters, sharp corners and obstructions The bore should be not less than the nominal diameter ordered, and should be symmetrical with, and the ends should be cut off square with, the longitudinal axis of the duct. The wall of the duct should be not less than 3/8 in. thick at the thinnest place, but the duct should average 5/8 in. thick. The inner edges of the ends of each piece of duct should be chamfered off so that sharp edges will not be encountered when cable is drawn in. The completed duct should have the necessary mechanical strength and toughness to prevent chipping at the ends, and breakage in ordinary handling The outer surface of all duct should be scored in such manner as to give cement a hold on the surface of the duct.

Fiber Duct. Fiber duct is made of a finely divided wood pulp or fiber, thoroughly impregnated with bituminous insulating compound to give walls of hard, compact, homogenous material The duct should not be affected by acids, alkalies, or moisture and should be free from all substances which might corrode or injure the lead sheath or rubber compound of a cable. The duct material should not puncture when an alternating current of not less than 25 nor more than 100 cycles per second and approximating as closely as possible to a sine wave is applied between the inner and outer surfaces for a period of at least 5 min Previous to the application of this test, the duct to be tested may be immersed in water for not more than 48 nor less than 24 hr The voltage to be applied

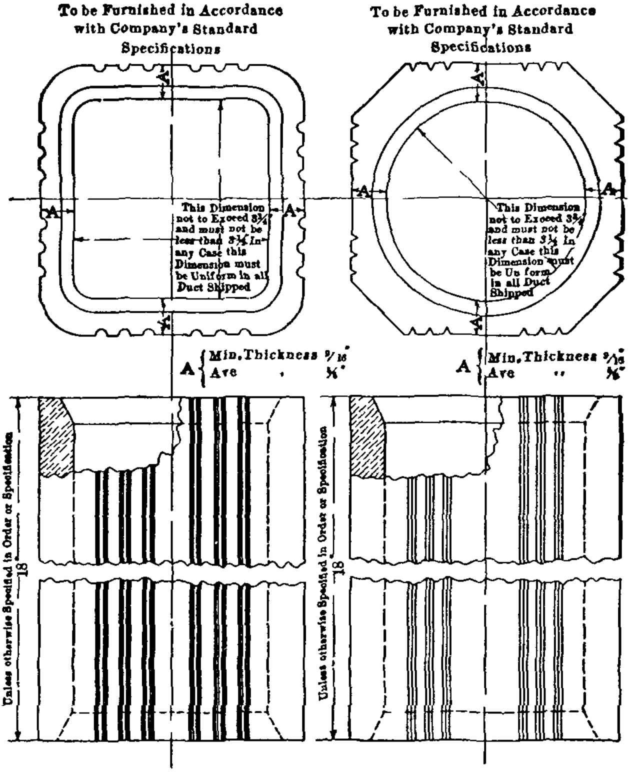


Fig 39 —Tile duct, square and round

shall be that corresponding to the nominal thickness of the duct wall in accordance with the following table

Nominal thickness of duct wall	Voltage
⅛ in	7 500
¼ in	15,000
5/16 in	18 800
⅜ in	22 500
½ in	30 000

The circumference, including the joints, should not vary at any point more than 1/16 in from a true circle at a temperature of 65 deg C (150 deg F) or less The thickness of the duct should not be more than 1/32 in less or 1/8 in greater, at any point, than the nominal thickness as follows

NOMINAL THICKNESS OF DUCT WALL FOR EACH TYPE INCHES

Nominal size of duct	Socket joint	Drive joint	Screw joint
1½ in	¼ in	¼ in	5/16 in
2 in	¼ in	¼ in	⅜ in
2½ in	¼ in	¼ in	⅜ in
3 in	¼ in	¼ in	7/16 in
3½ in	¼ in	¼ in	7/16 in
4 in	¼ in	¼ in	½ in

Each piece of duct should be capable of passing a mandrel 36 in long and of cross-section ⅛ in less than the nominal diameter of the duct A straight edge laid lengthwise on the concave side of a 5 foot section of duct should not show an offset of more than ¼ in Joints should be constructed as hereinafter specified

Socket joints should have a mortise or female joint on one end and tenon or male joint on the other end of each piece of duct The mortise and tenon should be machine cut to produce a snug fit ⅝ in long, slightly tapered and free from projecting surfaces which would prevent the joint from being properly assembled The thickness of the duct left after the mortise and tenon have been turned should not be less than 1/32 in less than one-half the nominal thickness of the pipe

Drive joints should have smooth machine cut tapers on each end of each piece of duct The taper should be four degrees on the diameter For each joint there should be furnished a sleeve, of the same material as specified for the duct, machine cut to an internal taper at each end, the taper being the same as that specified for the duct The minimum thickness of the sleeve should not be less than one half of the nominal thickness of the duct The tapers on the duct and the sleeve should be so cut that when the joint is made up the ends of the duct will neither touch nor be sepaated by more than ⅛ in

Screw joints should have machine cut threads on each end of each pipe For each joint there should be furnished a sleeve of the same material as specified for the duct, having machine cut thread to give an easy fit on the threads of the duct The minimum thickness of the sleeve should be not less than three-quarters of nominal thickness of the duct The threads should be so cut and the ends of the duct should be so faced that the ends of the duct will butt with a firm water tight joint when the joint is screwed up firmly by hand, using a joint compound furnished by the manufacturer The threads should be four to the inch Bends should have left hand threads and sleeves for bends should have one end with left hand thread All other threads should be right hand

Construction of Conduit. A bed of concrete to the depth of not less than 3 and preferably 4 in should be put in the bottom of the trench and the surface brought to grade The duct should be laid like brickwork in cement mortar in such a manner as to break joints both horizontally and vertically. Care should be taken in selecting cement for the mortar used in this work, on account of the possibility of seepage through the joints. A quick-setting cement is recommended, and the mortar should be as dry as practicable to obtain adhesion. After the top layer of ducts is in place, all exposed joints should be thoroughly covered by a coating of cement mortar, applied with a trowel. A 30 in. mandril should be drawn through each duct as it is laid; the mandril to be of a diameter ¼ in. less than the interior diameter of the duct. The mandril should be left in each duct until the next succeeding duct is laid. The space between ducts and sides of ditch, which space should in no case be less than 3 in., and for a depth of 3 or 4 in. above the duct should be filled with cement concrete. Each layer of concrete should be thoroughly rammed as it is put in. The top of the finished conduit should not be less than 2 ft. below the surface of the street.

Where fiber conduit is used, the ducts should be laid one tier at a time, using spacing combs about every 3 feet, and in such a manner as to break joints both horizontally and vertically. All joints should be so assembled and so sealed with approved compound that they will be water tight. No duct should be laid until the concrete base or concrete over the previous tier has set firmly. After each layer of duct is in place all space between ducts and all space between ducts and side boards, which latter in no case must be less than 3 inches, and the space for a distance of 1¼ inches above the top of all but the top tier of ducts is filled with concrete and thoroughly rammed so as to secure a complete concrete encasement for each separate duct of a minimum thickness at any point of 1 inch. After the concrete has partly set the combs are removed and the holes filled with concrete. In no case should a comb or spacer be left in the concrete.

Manholes. Manholes should be placed at street intersections or turns and wherever necessary to make cable joints. The distance between manholes should not exceed 500 ft. A manhole should be kept well ventilated, dry and clean. Fig. 40 shows a typical brick two-way manhole, from the Miscellaneous Methods of the American Electric Railway Engineering Association. The top of the manhole casting should be set so as to conform to the grade line of the street and the space between the bottom of the casting and the covering slab should be filled with brickwork laid in cement mortar. All bricks used should be of the best quality, whole, sound, perfect, and hard burned throughout. Every brick should be thoroughly wet by immersion in water previous to laying. In laying, each brick should have a full, close joint of cement mortar made at one operation on its bed, ends and sides. All joints should be thoroughly trowel struck. If the manhole is to be constructed of concrete, its walls, floor and ceiling should be of thickness not less than shown in Fig. 40.

Specifications for Electrical Conduit Construction. The Manual of the Amer. Elec. Ry. Eng. Assn. contains a complete recommended form of specifications for electrical conduit construction. Fig. 41 shows a typical conduit record sheet.

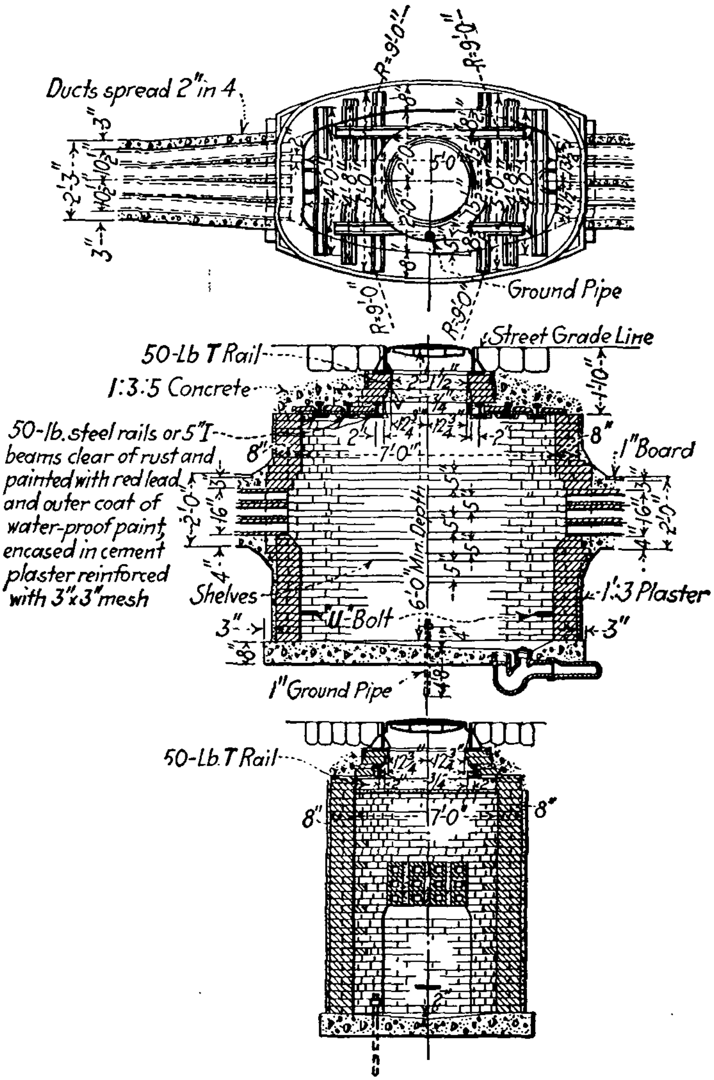


FIG. 40.—Typical design for 4 ft. × 7 ft. two-way manhole.

Limiting Clearance Lines for Third Rail Structures, A.E.R.E.A., A.R.E.A. and A.R.A. Standard. Fig 42 gives the standard locations of the limiting clearance lines for third rail and permanent way structures and rolling equipment. The space within the lines

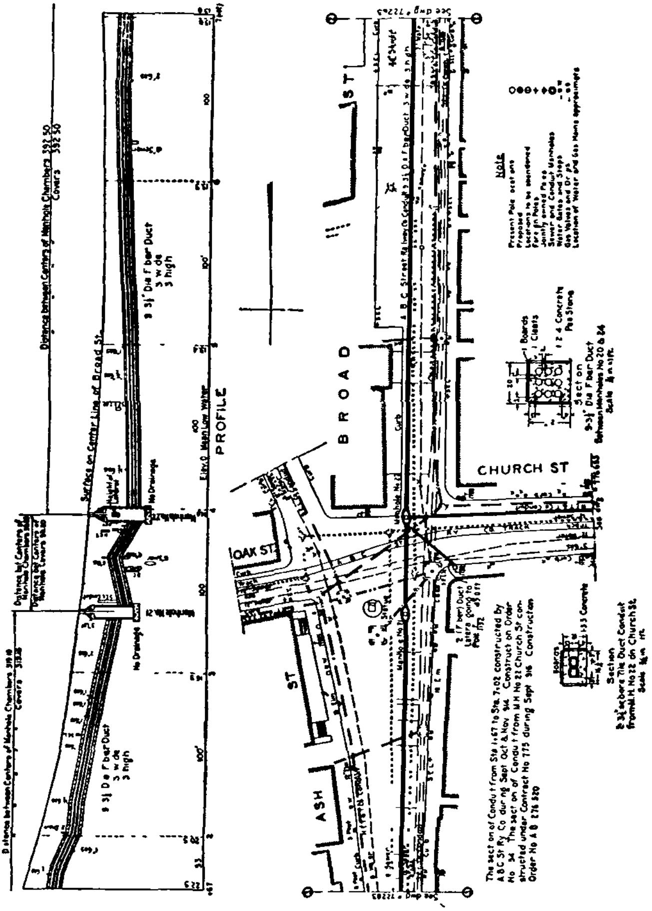


FIG. 41 —Typical conduit record sheet (reduced size).

AT, *BT*, *CT*, *DT*, *ET*, *FT*, and *AT*, *JT*, *KT*, *LT*, *MT* should be reserved for third rail structures on tangent track or curves of radius greater than 800 ft. (For curves of radius less than 800 ft see notes in Fig 42) Permanent way structures on tangent track or curves of radius greater than 800 ft. should not be nearer the third rail structure than lines *AS, JS, KS, LS, MS*. (For curves of radius less than 800 ft. see note in Fig. 42.) Rolling equipment under conditions of maximum wear and deflection on tangent track or curves of radius greater than 800 ft should not be nearer the third-rail structure than lines *AE, BE, CE, DE, EE, FE, GE*. (For curves of radius less than 800 ft see note in Fig 42.)

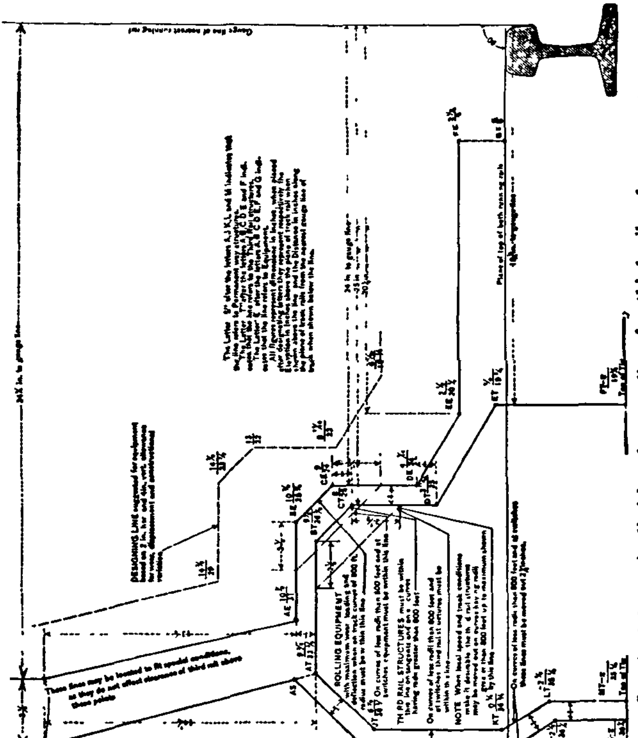


Fig. 42 —Standard design for **limiting clearance lines for third rail and permanent way structures and rolling equipment.**

Third Rail Gage and Elevation. The definitions of these terms, as shown on page 547, have also been adopted as standard by the Am El Ry. Eng. Assn. Standard dimensions, as adopted by the Amer. Inst of Elec. Engrs, are given on page 548

Composition and Resistance of Third Rail. The following tables and conclusions regarding the composition of steel and iron conductor rail are from a paper by J A Capp before the American Institute of Mining Engineers, 1903. The table on page 632 gives

Resistance and Composition of Steel and Iron*

Serial number	Specific resistance: Microhms per Cm/Cm³	Specific resistance: Temp. ° C.	Conductivity: Matthiessen standard	Resistance: Cu. = 1	Percentage composition: C	Mn	P	S	Si	Total not Fe	P + S + Si
1	22.72	19.0	7.58	13.20	0.33	1.27	0.09	0.05	0.05	1.79	0.19
2	20.90	20.0	8.27	12.12	0.17	1.09	0.09	0.05	0.004	1.404	0.144
3	21.29	25.0	8.27	12.09	1.40	0.222	0.01	0.020	0.082	1.734	0.112
4	19.87	19.0	8.65	11.55	0.20	0.95	0.10	0.08	0.05	1.38	0.23
5	19.80	19.0	8.68	11.51	0.43	0.77	0.10	0.04	0.066	1.406	0.206
6	19.80	19.0	8.69	11.51	0.36	0.80	0.10	0.04	0.047	1.347	0.187
7	19.81	20.0	8.69	11.51	0.22	1.08	0.10	0.05	0.06	1.510	0.210
8	19.69	19.0	8.73	11.44	0.74	0.58	0.043	0.036	0.20	1.599	0.279
9	18.95	25.0	9.29	10.76	1.61	0.147	0.015	0.018	0.092	1.882	0.125
10	18.17	19.0	9.46	10.56	0.41	0.72	0.039	0.041	0.11	1.32	0.190
11	17.27	19.0	9.96	10.04	0.36	0.87	0.08	0.09	0.04	1.44	0.21
12	17.10	19.0	10.06	9.94	0.37	0.73	0.09	0.04	0.06	1.29	0.19
13	17.10	19.5	10.06	9.94	0.23	0.80	0.046	0.033	0.016	1.095	0.065
14	16.96	19.0	10.14	9.86	0.30	0.95	0.063	0.01	0.01	1.333	0.083
15	16.95	19.5	10.14	9.86	0.29	0.99	0.084	0.01	0.01	1.384	0.104
16	16.32	19.0	10.55	9.48	0.23	0.89	0.053	0.01	0.005	1.193	0.073
17	16.25	19.5	10.59	9.44	0.26	0.83	0.053	0.01	0.004	1.157	0.067
18	16.21	20.0	10.62	9.42	0.28	0.65	0.083	0.06	0.05	1.123	0.193
19	16.09	19.0	10.60	9.36	0.22	0.68	0.077	0.07	0.05	1.097	0.197
20	16.09	19.0	10.69	9.36	0.16	0.66	0.074	0.030	0.014	0.938	0.118
21	15.32	19.0	11.24	8.90	0.33	0.49	0.068	0.05	0.02	0.958	0.138
22	14.57	19.5	11.82	8.46	0.31	0.45	0.10	0.04	0.026	0.926	0.166
23	14.49	20.0	11.88	8.42	0.25	0.41	0.10	0.04	0.03	0.83	0.17
24	14.73	23.5	11.88	8.42	0.144	0.46	0.09	0.08	tr.	0.774	0.17
25	14.62	23.5	11.96	8.36	0.188	0.48	0.09	0.08	tr.	0.83	0.17
26	14.15	19.0	12.17	8.22	0.22	0.56	0.024	0.34	tr.	0.838	0.058
27	14.03	19.0	12.26	8.16	0.192	0.57	0.024	0.34	tr.	0.82	0.058
28	13.86	19.0	12.41	8.06	0.16	0.48	0.091	0.04	0.01	0.781	0.144
29	13.83	19.5	12.44	8.04	0.10	0.55	0.08	0.05	0.024	0.804	0.154
30	13.80	19.0	12.57	8.02	0.14	0.41	0.11	0.05	0.009	0.719	0.169
31	13.67	19.0	12.58	7.95	0.23	0.48	0.024	0.01	0.023	0.767	0.057
32	13.64	19.0	12.61	7.93	0.24	0.57	0.029	0.01	0.003	0.850	0.042
33	13.90	24.0	12.63	7.92	0.10	0.25	0.04	0.02	0.05	0.46	0.11
34	13.31	19.0	12.92	7.74	0.25	0.37	0.04	0.03	0.018	0.708	0.088
35	13.30	19.5	12.94	7.73	0.23	0.49	0.024	tr.	0.004	0.748	0.028
36	13.27	19.0	12.97	7.71	0.19	0.37	0.09	0.05	0.01	0.71	0.15
37	13.25	19.0	12.99	7.70	0.27	0.41	0.024	0.01	0.001	0.715	0.035
38	13.18	19.0	13.05	7.66	0.28	0.28	0.027	0.034	0.04	0.661	0.111
39	13.18	19.0	13.05	7.66	0.07	0.40	0.08	0.07	0.013	0.633	0.163
40	13.07	19.0	13.16	7.60	0.28	0.42	0.022	0.04	0.008	0.770	0.070
41	12.87	20.0	13.27	7.48	0.16	0.38	0.08	0.04	0.009	0.669	0.129
42	12.73	20.0	13.52	7.40	0.15	0.45	0.011	0.033	tr.	0.644	0.044
43	12.69	19.0	13.55	7.38	0.19	0.21	0.025	0.04	0.034	0.499	0.099
44	12.53	19.0	13.74	7.28	0.215	0.22	0.051	0.113	.....	0.599	0.164
45	11.01	19.0	15.63	6.40	0.05	0.19	0.054	0.059	0.03	0.383	0.143
46	13.80	25.5	12.78	7.82	0.15	0.068	0.13	0.02	0.15	0.518	0.30
47	13.82	26.0	13.37	7.48	0.15	0.064	0.036	0.02	0.13	0.400	0.186
48	13.10	26.0	13.50	7.41	0.16	0.074	0.12	0.027	0.10	0.481	0.247
49	12.54	25.5	14.07	7.11	0.08	nil	0.13	0.008	0.024	0.242	0.162
50	11.92	25.5	14.80	6.76	0.17	0.027	0.074	0.022	0.077	0.370	0.173
51	10.82	24.0	16.21	6.17	0.058	0.10	0.014	tr.	0.012	0.184	0.026
52	10.80	25.5	16.34	6.12	0.16	0.018	0.049	0.011	0.015	0.252	0.075
LSS	11.40	17.0	15.00	6.68	0.050	0.180	0.013	0.011	0.02	0.274	0.044
B	11.00	17.0	15.57	6.44	0.030	0.036	0.065	0.016	0.14	0.287	0.221
SCI	10.35	17.0	16.55	6.06	0.028	tr.	0.004	0.005	0.07	0.107	0.079

* Serial numbers 1 to 45 inclusive are steel.
Serial numbers 45 to end of table are iron.

temperature and specific resistance at that temperature, conductivity and resistance, compared to that of copper, and the composition for each of several samples of steel and iron tested. The samples having serial numbers 1, 2, 4, 7, 11 and 12 were standard T-rails. Nos. 24 and 25 were cut from T-rail used for conductor on the Aurora, Elgin and Chicago R. R. Nos. 46, 47 and 48 were ordinary refined bar-iron; 49 and 50 were special refined bar-iron for staybolts and similar use, 51 and 52 were Swedish and Norway iron respectively.

The values in the tables showing the variation of resistance with manganese were selected from the table on page 632 in studying the influence of manganese on resistance, and indicate that the effect of manganese in increasing resistance gradually increases with the percentage of manganese present, within the limits represented by these samples. Messrs. Barrett, Brown and Hadfield (Trans. Royal Dublin Society, Vol. VII, series 2, Part IV) found the resistance to increase at first very rapidly, with constantly increasing percentage of manganese, then more and more slowly, until 7 per cent manganese, after which a further increase in the percentage of manganese produces little or no increase in resistance.

RESISTANCE OF STEEL. VARIATION WITH MANGANESE

(Carbon from 0.17 to 0.23 per cent)

Sample number	Manganese, per cent	Resistance	Carbon, per cent	P + S + Si per cent
2	1 09	12.12	0.17	0.144
4	0.95	11.55	0.20	0.23
7	1.08	11.51	0.22	0.210
13	0.80	9.94	0.23	0.065
16	0.89	9.48	0.23	0.073
19	0.68	9.36	0.22	0.197
25	0.48	8.36	0.188	0.17
26	0.56	8.22	0.22	0.058
27	0.57	8.16	0.192	0.058
31	0.48	7.95	0.23	0.057
35	0.49	7.73	0.23	0.028
36	0.37	7.71	0.19	0.15
43	0.21	7 38	0.19	0.099
44	0.22	7.28	0.215	0.164

RESISTANCE OF STEEL. VARIATION WITH MANGANESE

(Carbon from 0.27 to 0.33 per cent)

Sample number	Manganese, per cent	Resistance	Carbon, per cent	P + S + Si per cent
1	1.27	13.20	0.33	0.19
14	0.95	9.86	0.30	0.083
15	0.99	9.86	0.29	0.104
18	0.65	9.42	0.28	0.193
21	0.49	8.90	0.33	0.138
22	0.45	8.46	0.31	0.166
37	0.41	7.70	0.27	0.035
38	0.28	7.66	0.28	0.111
40	0.42	7.60	0.28	0.070

The values in the tables showing the variation of resistance with carbon were selected from the table on page 632 in studying the influence of carbon on resistance. With uniformly increasing carbon the resistance of unhardened steel at first rises very rapidly, but the rate of increase gradually drops until it reaches a straight line at about 0.2 per cent C., which continues up to limits of the carbon listed.

RESISTANCE OF STEEL. VARIATION WITH CARBON
(Manganese from 0.15 to 0.30 per cent)

Sample number	Carbon, per cent	Resistance	Manganese, per cent	P + S + Si per cent
3	1.40	12.09	0.222	0.112
9	1.61	10.76	0.147	0.125
33	0.10	7.92	0.25	0.11
38	0.28	7.66	0.28	0.111
43	0.19	7.38	0.21	0.099
44	0.215	7.28	0.22	0.164
45	0.05	6.40	0.19	0.143

RESISTANCE OF STEEL. VARIATION WITH CARBON
(Manganese from 0.4 to 0.5 per cent)

Sample number	Carbon, per cent	Resistance	Manganese, per cent	P + S + Si per cent
21	0.33	8.90	0.49	0.138
22	0.31	8.46	0.45	0.166
23	0.25	8.42	0.41	0.17
24	0.144	8.42	0.46	0.17
25	0.188	8.36	0.48	0.17
28	0.16	8.06	0.48	0.144
30	0.14	8.02	0.41	0.169
31	0.23	7.95	0.48	0.057
35	0.23	7.73	0.49	0.028
37	0.27	7.70	0.41	0.035
39	0.07	7.66	0.40	0.163
40	0.28	7.60	0.42	0.070
42	0.15	7.40	0.45	0.044

The following is also from Mr. Capp's paper: "The elements phosphorus, sulphur and silicon were not present in sufficient quantity in any of the samples tested to permit us to draw any curves showing their influence upon the resistance. . . . In commercial steels the percentages of all three of these elements is so small that their effect on resistance may generally be neglected." A study of the several tables given in the paper shows that manganese preponderates in influencing the resistance of steels, and that for lowest resistivity, this element must be present in very small quantity, much smaller than is usual in merchant or structural steels. While all the other elements must be present only in very small percentages, so great is the preponderance of the influence of manganese, that they may be tolerated in quantities which the steel makers would consider reasonable, without unduly increasing the resistance. For a satisfactory third rail the lowest possible resistance (from 6 times to 6.5 times that of copper?) is not necessary; and the great cost of making such extremely pure steel is not warranted. In fact, such extremely pure steels would probably be so soft that the frictional wear of the collecting shoe would be excessive and the life of the rail unduly short. Assuming, then, that a rail from steel having a resistance not greater than eight times that of copper (13.8 microhms at 20 deg. C.) would be desirable for conductor rails, the figures tabulated would seem to indicate that the following extreme composition would be permissible:

	Per cent
Carbon up to	0.2
Manganese up to	0.4
Phosphorus up to	0.06
Sulphur up to	0.06
Silicon up to	0.05

This composition, however, would be extreme, and any overstepping of bounds might result in too great resistance; therefore for resistance up to eight times that of copper, the specified analysis should be:

	Per cent
Carbon not to exceed	0.15
Manganese not to exceed	0.30
Phosphorus not to exceed	0.06
Sulphur not to exceed	0.06
Silicon not to exceed	0.05

The following suggested third rail composition is from the Standard Handbook for Electrical Engineers:

	Per cent
Carbon not to exceed	0.12
Manganese not to exceed	0.40
Sulphur not to exceed	0.05
Phosphorus not to exceed	0.10

The resistance of a third rail of the composition given above will be approximately 7.75 times that of an equivalent area of commercial copper.

Relative to conductor rails installed by the Underground Electric Rys. Co., London., S. B. Fortenbaugh (paper A.I.E.E., 1908), says: "The resistance of these rails was about 6.4 times that of an equivalent area of copper and the chemical composition substantially as follows:

	Per cent
Carbon	0.05
Manganese	0.19
Sulphur	0.06
Phosphorus	0.05
Silicon	0.03

It is interesting to note that the cost of this special conductor rail was no more than the standard track rail."

British Standard Method of Specifying Resistance of Steel Conductor Rails. The British Engineering Standard Committee has adopted (1914) the following: In specifying the resistance of a steel conductor rail the value shall be given in microhms (millionths of an ohm) and shall be expressed as the resistance in microhms at a temperature of 60 deg. F. (15.6 deg. C.) of a rail of the same material as the conductor rail in question, having a length of 1 yd. and a weight of 100 lb. It follows that if the resistance of a rail weighing 100 lb. to the yard be R_s microhms, that of a rail weighing W lb. to the yard will be $\frac{100R_s}{W}$ microhms. Conversely, if the resistance of a rail 1 yd. in length weighing W lb. to the yard be R_w microhms, that of the corresponding 100-lb. rail will be $\frac{WR_w}{100}$ microhms. In either case we have the relation $100R_s = WR_w$. For the purpose of reducing observations made at temperatures other than 60 deg. F. to the standard temperature, a mean coefficient of increase of resistance with temperature of 0.26 per cent per deg. F. (0.47 per cent per deg. C.) shall be employed unless otherwise specified. Where greater accuracy is required for a wide range of temperature, the coefficient of the actual piece of rail should be used.

Temperature Coefficient of Conductor Rails. Commenting on the above, the British Committee notes that the temperature coefficient of the conductor rails tested at the National Physical Laboratory for the purpose of the report varied from 0.28 per cent per deg. F. (0.51 per cent per deg. C.) for a rail of 15.9 microhms per 100-lb. yard to 0.24 per cent per deg. F. (0.43 per cent per deg. C.) for a rail of 20.5 microhms per 100-lb. yard. Taking the extreme case of a rail of 16 microhms per 100-lb. yard tested at 40 deg. F. (4.5 deg. C.) or 80 deg. F. (27 deg. C.), the use of the mean temperature coefficient may introduce an error of ± 0.6 per cent. For a smaller range of temperature the error would be proportionately less.

National Physical Laboratory Tests. The above-mentioned report of the British Engineering Standards Committee gives the following table showing the relation between the resistance, temperature coefficient and composition of some samples tested at the National Physical Laboratory for the purpose of the report. (Sample 1 is electrolytic iron; 2, 3, 4 and 5 are conductor rails; 6 is a piece of ordinary track rail.)

Sample	Microhms per 100-lb. yard	Relative resistivity		Temp. coeff. per cent per deg. C.	Carbon	Silicon	Sulphur	Phosphorus	Manganese	Arsenic	Copper	Nickel	Chromium
		Mass	Volume										
1	13.9	5.00	5.67	0.55	....	.....	....	....	....	....	....	....	.....
2	15.9	5.72	6.48	0.51	0.040	Trace	0.044	0.035	0.165	0.030	0.025	0.047	Trace
3	16.9	6.08	6.89	0.47	0.035	0.011	0.076	0.017	0.100	0.028	0.005	0.330	Trace
4	18.35	6.59	7.47	0.45	0.092	Trace	0.097	0.072	0.362	0.065	0.046	0.094	Trace
5	20.5	7.38	8.37	0.43	0.403	Trace	0.060	0.027	0.365	0.012	0.048	0.081	Trace
6	25.3	9.1	10.3	0.35	0.561	0.068	0.028	0.066	0.605	0.021	....	....	.....

Third Rail Sections. Both T-rails and rails of specially rolled sections are used for third rail. The section should be such as to offer ample surface for current collection, and it should have an area of cross-section proportional to the conductivity desired. The following table gives the weight per yard of the T-rail used for third rail in several installations:

Road	Weight of third rail, lb. per yd.
Englewood Elevated, Chicago	48
Metropolitan West Side Elevated, Chicago	48
Northwestern Elevated, Chicago	48
Grand Rapids, Grand Haven & Muskegon	60
Michigan United Railways	60
Interborough Rapid Transit Co., New York	75
Interborough Rapid Transit Co. (Westchester branch)	75
Lackawanna & Wyoming Valley	75
Wilkes-Barre & Hazelton	80
Boston Elevated	85
Aurora, Elgin & Chicago	100
Interborough Rapid Transit (elevated division)	100
Long Island R.R.	100
Puget Sound Elec. Ry.	100
Scioto Valley Traction Co.	100
Seattle-Tacoma Interurban	100
Pennsylvania Tunnel & Terminal R. R	150

The section shown in Fig. 43 was used in the New York Central & Hudson River R. R. electrification. Fig. 44 shows an inverted channel section devised by S. G. Redman and C. H. Merz, London. This channel is of irregular form, as the non-contacting flange is used to secure more cross-sectional conductivity and to keep the contact-making flange in place

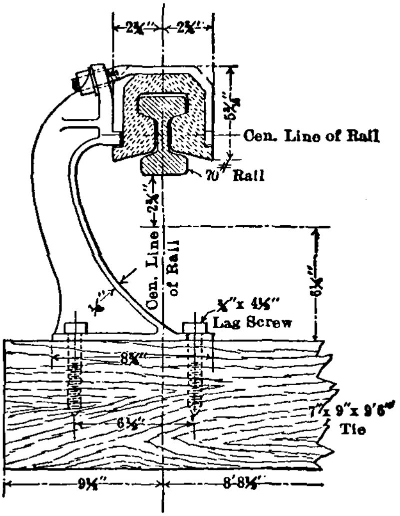


Fig. 43.—N. Y. C. type underrunning third rail.

Third Rail Support. *The 1908 report of the* Committee on Power Distribution, A.E.R.E. A. states "The spacing of third rail supports varies without respect to the type of rail. The spacing most used is that of 10 ft., with a maximum in some instances of 11 ft. and a minimum of from 5 ft. to 6 ft. These locations are apparently governed by the lengths of third rail and the standard tie spacing in use. The weights of brackets vary between 9 lb. with the over running rail and 13 lb. to 20 lb. for the underrunning rail. The committee can see no reason for such great diversity in spacing of supports, and would recommend that a spacing of 10 ft. be used on all third rail construction where 30-ft. conductor rail is used, and 11 ft. where 33-ft. conductor rail is used. In all cases but one (reported), the support for the bracket is an extended tie, which is also a part of the track structure. In the one case referred to

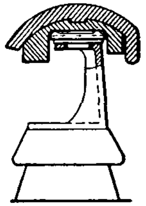


FIG. 44.—Inverted channel third rail.

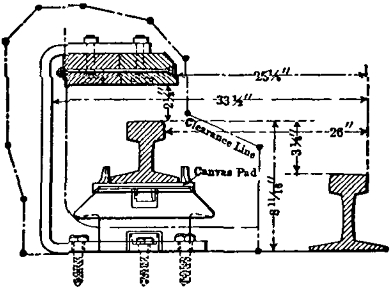


FIG. 45.—Third rail and support, Phila. & Western.

(subway of the Philadelphia Rapid Transit Co.), it is entirely independent of the track structure. It might be well in this connection to note that the road referred to reports an absolute lack of insulator breakage, undoubtedly accounted for by the above condition of supports independent of track structure." Fig. 45 shows the method of supporting the overrunning T-section third rail on the Philadelphia & Western Ry. Fig. 46 shows the method of supporting the underrunning T-section third rail of the 1200-volt interurban division of the Central California Traction Co. Fig. 43 shows the method of supporting the underrunning third rail on the New York Central & Hudson River R. R. Fig. 44 shows the method of supporting the Redman-Merz underrunning third rail. The rectangular base of this channel is supported on a flat insulator, an intermediate bracket and a foundation insulator. The capping of the conductor channel may be of fiber, stoneware, or other material keyed into the conductor as indicated.

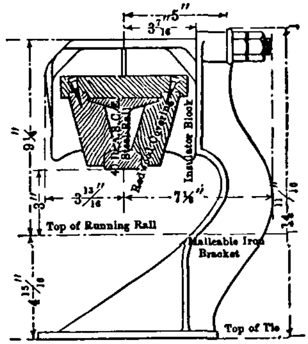


FIG. 46.—Underrunning (T) third rail, California.

Third Rail Insulation. The 1908 report of the Committee on Power Distribution, A.E.R.E.A. states "Three kinds of insulation are used with the overrunning type of rail: wood, reconstructed granite and composition. Four kinds of insulation are used in the underrunning type: wood, porcelain, semi-porcelain and composition."

A third rail insulator should have sufficient mechanical strength to stand up under conditions of severe mechanical shocks and vibration. It must also support the rail in such a manner that expansion and contraction of the rail can take place without tipping or damaging the insulator. The insulator should be made up of a small number of pieces and there should be no bolts used which can loosen from vibration, shrinkage of insulation, etc. The insulator must provide adequate electrical insulation for the voltage on which it is used. It should be so designed that a liberal water drip is provided so that there will not be a continuous leakage path from the conductor rail to the ground during times of rain and moisture.

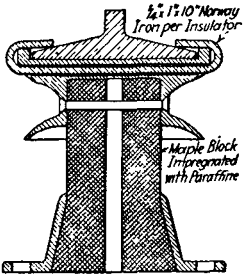


FIG. 47.—Third rail insulator, Chicago Elevated Railroads.

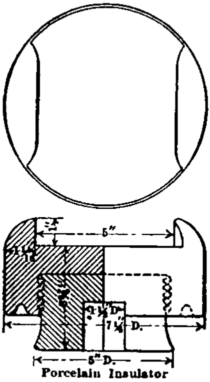


FIG. 48.—Third rail insulator, Lackawanna & Wyoming Valley.

The insulating material should be such that it will stand up under vibration and shocks. The experience of the Chicago Elevated Railroads indicates that maple wood blocks, impregnated in paraffin for a period of several hours, meets the electrical requirements for 600 volt service in a very satisfactory manner. The top and bottom castings of the insulator illustrated in Fig. 47 are one piece affairs, easily assembled, are ample in thickness so they do not rust out readily, and the means of holding the rail down in the chair is simple; the clips can be removed by breaking off and the insulator re-used by inserting a new clip. Two types of insulators are used in the installation shown in Fig. 45. The one-piece insulator is dry-process porcelain and the two-piece insulator is wet-process porcelain. A cast-iron cup with lag screw holds the insulator in place on the tie and a cast-iron cap on top of the insulator holds the third rail in position. A canvas pad between this cap and the insulator, together with provision for vertical movement, reduces the vibration which is so destructive to the porcelain. In the installation shown in Fig. 46 a porcelain block is used at each bracket. In the installation shown in Fig. 43 the third rail is loosely held in each bracket by non-charring, moisture-proof insulator blocks. Fig. 48 shows a two-piece porcelain insulator used on the Lackawanna & Wyoming Valley R. R. The two pieces of porcelain are cemented together as shown and then mounted on a wooden pin.

Third Rail Protection. Methods of protecting overrunning and underrunning T- and U-section third rail are shown by Figs. 43 to

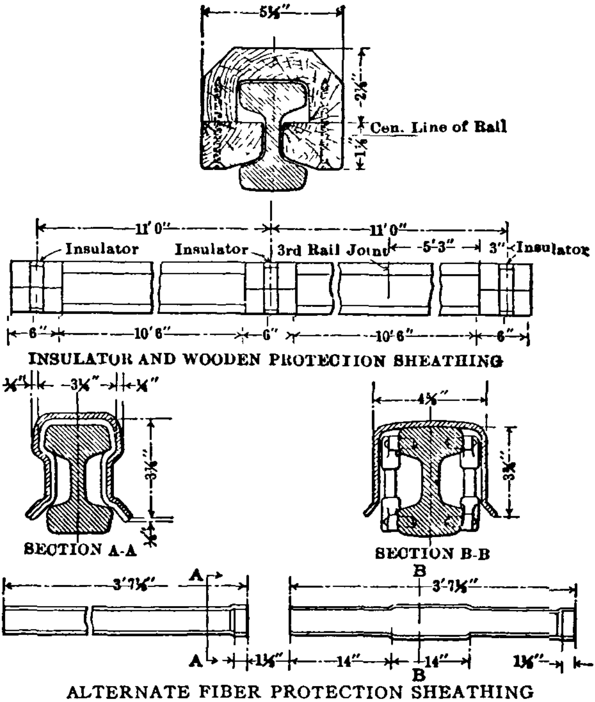


FIG. 49.—Protection for N. Y. C.-type third rail.

45, respectively. The details of two types of protection used on the New York Central type third rail (Fig. 43) are given in Fig. 49. In this type a sheathing of wood or fiber reaching from bracket to bracket embraces the rail head, reaches nearly to the running face of the rail and extends outward from the web, thus forming a petticoat.

Operation of Third Rail in Snow. In Feb., 1906, tests of the operation of third rail in snow were made on the New York Central & Hudson River R. R. After medium heavy snow had fallen, without drifting, to an average depth of 17 in., a flanger was sent over the track. This packed the snow against the third rail. The test was made by observing the operation during the passage of an electric locomotive on several trips. During the first passage over the overrunning unprotected third rail there was very little flashing or trouble. Due to the formation of ice and the ironing out of snow, operation grew worse on succeeding passages until it became almost impossible. Results with the overrunning protected third rail were about the same as with the overrunning unprotected third rail. The first passage scooped the snow away for 2½ or 3 in. under the contact surface of the underrunning protected third rail. Each succeeding passage tended to clean the contact surface and operation was practically free from trouble.

Operation of Third Rail in Sleet. In March, 1906, tests of the operation of third rail after a sleet formation were made on the New York Central & Hudson River R. R. The sleet formation continued for a day in a wind having a velocity of about 2 miles an hour and temperature ranging from 28 to 33 deg. F. The thickness of the sleet averaged about ¼ in. The effect of the sleet on the overrunning unprotected third rail was so bad that running was impossible after 3 hours. In places where the wind caused the sleet to distribute entirely over the running surface of the rail the contact shoes arced badly and the operation was unsatisfactory. *There was no sleet formation on the running surface of* the underrunning rail and there was no sign of arcing.

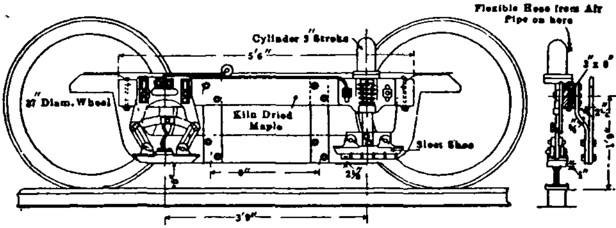


FIG. 50.—Third rail sleet shoe, Michigan United.

Mechanical Third Rail Sleet Remover. A type of third rail sleet shoe designed by W. Silms, and which is giving satisfaction on the Michigan United Railways, is shown in Fig. 50. It is of the same design as the regular third rail shoe, but has four steel cutters set diagonally in its face and cast integral with the body of the shoe. When these cutting edges become worn or damaged they are chipped off, and the shoe is kept in regular service until it is worn out. The sleet-cutting shoe is mounted on a vertical iron shaft which passes through guides and is attached to the piston of an air cylinder which has a 3-in. stroke. A spiral spring around this shaft holds the shoe off the rail except when air pressure is put on the cylinder to press the cutters against the rail. The air supply is taken from the train line through a ⅜-in. three-way valve and pipes extending under the car to a point convenient for connecting with the cylinder which

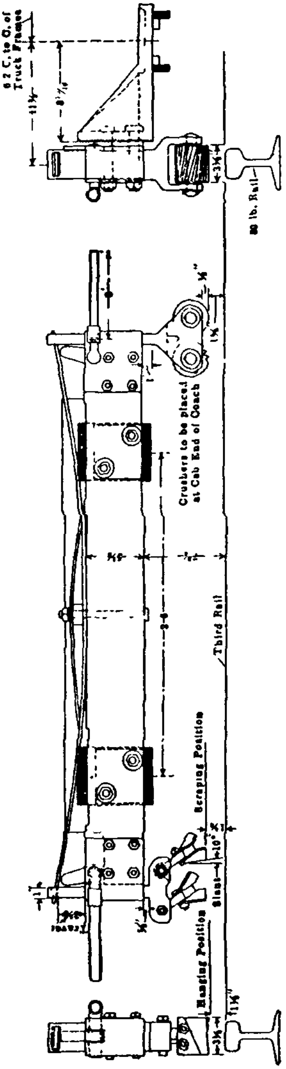


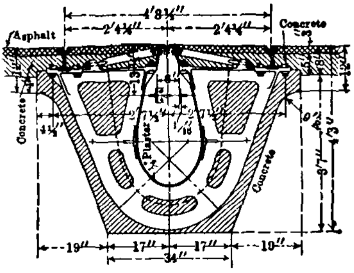
FIG. 51 —Sleet-removing device—Metropolitan West-side Elevated.

operates the shoe. This connection between the pipes and the cylinder is made with 2½-ft lengths of ⅜-in air hose, secured at each end with hose clamps. An ordinary straight valve is placed in each supply pipe to enable the motorman to use one or both shoes as occasion demands.

The third rail sleet-removing device shown in Fig. 51, developed by the Metropolitan West Side Elevated R. R. Co., was found by that company to give satisfactory service under all operating and weather conditions. The device consists of a piece of oak 2⅝ in. × 5½ in. × 4 ft. 1¾ in., at one end of which crusher rolls are placed and at the other end are scrapers. The crushing rolls, 3¼ × 3¼ in. in size, are right and left spiral toothed, made of cast crucible steel, hardened, but not machined. These rollers are carried on 1 × 6-in. steel pins, which are held in place in the steel slide-rod casting by cotter pins at each end. At each end of the oak board is a cast iron cam and wooden handle, which is used in raising or lowering the rollers or scrapers from or to the working position. The scraper blades are 3¼ × 3¾ × 2⅜ in., and are made of tool steel. There are four of these blades bolted to cast steel pivots so that they may take the scraping position for either a backward or forward movement of the car. The scrapers are supported on a slide rod provided with the elevating cam similar to the crushers. Immediately inside the guide plates which support the crusher and scraper slide rods are two cast iron corrugated plates which are used in adjusting the sleet remover for different truck heights and wheel wear. Bolted to the top of the oak insulating timber is an adjusting leaf spring, the ends of which are fitted into slots provided at the top of each slide rod. At first this spring provided 100 lb. pressure at both the crusher and scraper, later the crusher end pressure was increased to 200 lb. by adding another leaf to the crusher end of the spring. This device may be either bolted to the third rail shoe beam or held in position by means of a casting which is fitted to the car spring seats. A ½-in. flashboard made of ash and running the full length of the sleet remover is inserted between adjusting plates and the oak timber to provide additional insulation.

Calcium Chloride Third Rail Sleet Removal. On the Aurora, Elgin & Chicago Ry. a solution of calcium chloride made by dissolving caustic chloride of calcium in warm water in the proportion of 5 lb. of caustic chloride of calcium to 1 gal. of water has been used in the removal of sleet from the third rail and the prevention of the formation of a coat of sleet on the rail. The solution was kept in a 40-gal. tank in the motorman's cab and was led through a rubber tube to a ¼-in. pipe from which it was squirted upon the third rail a few inches in front of a steel sleet brush. The flow of liquid was controlled by a globe valve. The pipe was grounded to the truck frame. There were four sleet brushes per car and each was provided with a pipe. In operation only the forward pipe on the third rail side was used. The flow of liquid was regulated according to the speed of the train and about 1 gal. per mile was required. The sleet brush immediately behind the pipe spread the liquid uniformly over the surface of the rail. A thin sleet formation was dissolved and direct contact was made with the third rail. A thicker sleet formation was not melted at once, but was so loosened that it was scraped off the rail by the passage of one or two sleet brushes. The formation of new ice was prevented as long as the liquid remained on the rail, but after about 2 hours the liquid was removed to such an extent that another application was necessary. This process removed the sleet and kept the trains in operation, but it had a bad effect on the insulation of the third rail and car wiring. This led to the development of a side contact shoe having a cross-section similar to an inverted V. By riding on the edges of the third rail this shoe either removes the sleet or makes contact where the sleet is thinnest.

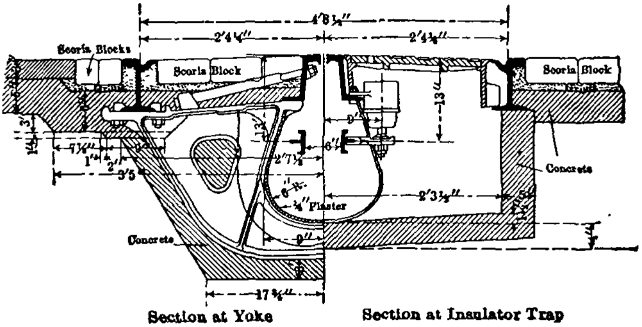
Conduit System. Figure 52 shows a section of cable conduit as rebuilt for electrical operation, and Fig. 53 shows a standard section of conduit, both as used in Washington, D. C. These are from a paper by J. H. Hanna, *Aera,* 1913. The essential elements of this conduit system consist of cast iron yokes supporting both wheel and slot rails and two steel "T"-shaped conductor bars supported from the bottom flange of the slot rail by insulators. The wheel rails are fastened with four hook bolts at each yoke seat with liners driven back of the bolts to allow accurate lining and gaging of the rails after the yokes have been concreted. The conduit proper is of concrete with concrete manholes at each insulator, spaced 15 ft. apart. Insulators consist of malleable caps with lugs by which the insulator is bolted to the slot rail porcelain insulators and forged studs cemented together, the latter supporting the conductor rail by means of adjustable malleable clips or brackets. The slot rail weight is 67 lb., and conductor bar, 22.4 lb. per yard. Yokes are spaced 5 ft. apart and weigh about 350 lb. each. The conductor bars are connected with underground feeder cables at intervals varying from 800 ft. for lines having dense traffic to much greater intervals in outlying districts. The distribution system is identical for positive and negative sides and is controlled at substations by double-pole, double-throw switches, which make it possible to reverse the polarity if necessary on account of grounds on different sides of different circuits.



Section at Yoke

FIG. 52.—Cable conduit rebuilt for electric operation.

The conduit system is a constant source of trouble to both the railway company and the traveling public. It is by far the least reliable of all the systems. This is because of the difficulties of ordinary operation and the excessive number of accidents which are possible with this system and against which it is impossible to provide. Neglecting accidents, for the moment, this system is the most difficult to keep in ordinary operation. The ordinary diffi-



Section at Yoke Section at Insulator Trap

FIG. 53.—Slot conduit, Washington.

culties reach a maximum with rain and snow storms. Snow tends to fill the conduit and must be removed by pushing it to manholes by scrapers. These manholes must be kept clean. The removal of snow interferes with traffic. To return to the purely accidental, various pieces of metal are washed or accidentally or maliciously dropped through the slot and these foul the plow and cause short circuits. Among the most common articles which cause this trouble may be mentioned automobile chains, hoops, rods, pipes and cables. Defective switches or plow guides in the slot often make the car and the plow start down different tracks. The results are bent plow bars, broken yokes, bent and grounded plows. Locating a ground in this system is often a slow process during which traffic is tied up.

Track Bonding

The following consideration of bonding deals with the application of electrical conductor to maintain good conductivity from rail to rail at the joints of the track circuit. This conductor usually serves no mechanical purpose at the joint. It should be noted that in city work the practice of joining the rails together by welding is increasing and that the use of additional electrical conductor is unnecessary where this is the practice. (See pp. 49–51.)

Classification of Bonds. Bonds may be classified according to the materials of which they are made, whether of solid copper, copper ribbon, or wire stranded copper; according to place of application to the rail, whether to the head, web or base, and if attached to web, whether "concealed" between joint plate and rail or "exposed" by being run over the outside of the joint plate; according to the method of attaching the terminal to the rail, whether compressed or expanded, brazed, soldered or amalgamated.

Compressed and Expanded Terminals. The bond terminal is forced into intimate contact with the walls of a hole drilled in the rail. The two general types of this terminal are commonly known as the "compressed (or stud) terminal" and the "pin terminal," respectively. The terminal of the compressed terminal bond is forced into contact with the wall of a hole in the rail by a screw or hydraulic press. The pin terminal is tubular and this tube is expanded into contact with the wall of a hole in the rail by driving a tapered steel pin into the bore of the tube. The following from the results of tests by the Chicago Board of Supervising Engineers, 1911, gives briefly some of the important points relative to the installation of these types of bonds:

First. Averaging the thirty-two tests on each type, the hydraulic compressed bond shows the least resistance, viz., about 96.65 per cent of the pin terminal type. Stated in terms of conductivity, the pin terminal bond is 96.65 per cent and the hand-compressed bond 98.64 per cent of the hydraulic compressed bond considered as the standard, 100 per cent.

Second. The best form of terminal is that in which the flow of copper is into and not out of the bore, that is, one in which the flow is restricted. In this respect the compressed terminal is superior, although not so easy to apply as the pin terminal, and it also insures a much better mechanical attachment.

Third. A certain pressure between copper and steel is essential to good electrical contact. Conductivity improves up to about 35,000 lb. per square inch. With the pin terminal, however, it is found that around 20,000 lb. the copper begins to flow out of the bore and no higher contact pressure is possible on this account. This maximum is somewhat dependent upon the texture of the copper and the lubricant used, and upon the manner in which the pin is driven into place.

Fourth. General precautions to be taken in bonding as largely developed from these tests are as follows: Use only accurately ground drills, entered at right angles to the web, and finish smooth. The surface of the rail web should be cleaned for 1/2 in. around the bore, and the latter should be thoroughly cleaned and dried before bonding; thus the head of the bond will properly abut the web surface. In driving the pin a heavy lubricant should be used, first with a drift pin of the proper size (7/16 in. for a 3/4-in. terminal), care being taken to avoid striking the bond itself.

Brazed Terminals. "Brazed" bond terminals which are generally flat are brazed to a spot on the rail which has been prepared by grinding. The heat is supplied by a blow torch or heavy electric current. The brass used to make the brazed joint is used in a solder-like strip. A bond brazed electrically is sometimes called an electrically welded bond. The current is supplied at from 1 to 6 or 8 volts from a transformer which, on a direct current road, is supplied by a motor-generator set operated on trolley voltage.

Soldered Terminals. Flat terminals are soldered to the rail or the end of a compressed or expanded terminal bond is soldered to the rail after the terminal has been compressed or expanded into place. The area over which the solder is to adhere to the rail is prepared by grinding. Heat is supplied by a blow torch or, in the case of the compressed terminal bond, the bond may be soldered by the Thermo bonding process, in which the necessary heat for soldering is obtained from the heat of reaction of a special compound which is ignited in a graphite cup held against the rail on the side opposite that to which the solder is to be applied.

Electric Weld Terminals. The demand for something better than a soldered contact and a substitute for the purely mechanical contact has been responsible for the wide adoption of the electric weld bond within the past few years. Although it has been used for the most part as a head bond it is coming into general use for concealed application to the web of the rail. There seems to be practically no question regarding the permanency of the contact that these bonds make with the rail, but some criticism has been directed against some of their other features, particularly to the breaking of the ribbons and to the inconvenience of using the bonding car on tracks over which traffic must be maintained. The question also has been raised as to what effect if any the welding heat has on the steel of the rail. Regarding injury to the rail by heat there seems to be very little definite information. A number of engineers have expressed a question or fear regarding this point, but with possibly one or two exceptions no company has definitely reported any broken rails from this cause. In the absence of more definite complaint on this score it is safe to say that the injury to rails resulting from the heat generated in welding rail bonds is so small as to be negligible and can be practically disregarded in the selection of a type of bond.

Oxy-acetylene Weld Terminals. This type of bonding is accomplished by welding bonds with forged or cast terminals to the rail, usually the head, and employing pure or fluxed copper to build up the terminal of the bond along the rail head. So far, it has not found a very wide application, although it has been used in Minneapolis and St. Paul for a number of years with marked success. It possesses practically all of the advantages of the electric weld bond and is free from the objection of requiring an expensive equipment and of interfering with traffic. Although it does not make quite as strong a contact with the rail as does the electric weld bond, the contact is permanent and does not deteriorate with time. No injuries to rails have been reported from the use of the flame which apparently does not produce a higher temperature in the rail than is generated by the electric weld.

Bonding Manganese Rail. As manganese steel is too hard to drill, its use has led to some difficulties in track bonding. Such material is used almost wholly in special work. Some companies depend wholly on the cable connections around the special work, as described on page 657, with no bonding of the separate pieces except the mechanical joints. Others report that soldered, brazed or welded bonds are satisfactorily attached to the manganese steel. Other companies specify the insertion of soft steel plugs in each manganese steel piece, into which holes may be drilled for compressed terminal bonds; in some cases only one soft steel plug is inserted in each piece, and a copper connection is run from this place to the jumper cable.

Ideal Bond. The material, form and structure of the ideal bond are briefly presented by C. W. Ricker, A.I.E.E., 1905, as follows: To get the necessary conductivity, bonds are nearly always of copper, about the only exception being those of tin amalgam. To reduce cost and resistance, they must be as short as practicable, and the manufacturing cost must be kept low, to preserve the scrap value as near the first cost as possible. For durability they must be flexible enough so as not to break or lose contact by the allowable relative motion of the rails. They must be formed so they may be applied to the types of rails in ordinary use, in such position as to be protected from accidental damage and from theft. They should be readily accessible for inspection and repair. The cost of application must be kept low, and to this end it is very important that the process shall be so simple and easy that no highly skilled labor or extraordinary care is required to install them with certain and uniform results.

Failure of Bonds in Service. Failures of bonds may be placed in three classes, namely, (1) breakage of bonds, (2) disintegration of bonds, (3) impairment of contacts. The following discussion of these cases is from the above noted paper by C. W. Ricker:

Breakage of Bonds. Breakage may occur because of defects in manufacture, as in copper bonds with welded terminals the strands may be weakened by overheating where they enter the terminal; and a slight but continuous motion of the joint will cause them to break, one by one, at this place. Long-continued jar and repeated flexures will produce fatigue in the metal. Such breakage in the case of either welded or solid bonds is of course most frequent where the flexure of the bond due to rail movements is too great for its flexibility, which means ill-selected bonds or badly-kept track. A less common manner of breakage occurs in laminated concealed bonds which are too large for the space between the joint plate and the rail, the bond shank is pinched and the working of the joint under passing wheels tears off the outer strands by a kind of ratchet effect, working them into the narrowing space at top or bottom of the rail web and sometimes squeezing them out of the joint in thin ribbons. Bonds secured under the base of the rail may be frozen in the ballast and torn off by the movement of the rail.

Disintegration. The surfaces at the imperfect welds in composite bonds corrode, increasing the resistance greatly and loosening and weakening the bonds so that they may be pulled apart. Tin amalgam used at contacts or in masses, hardens and shrinks, losing flexibility and contact with the bonded surfaces. In the case of amalgam plugs enclosed in cork boxes, the cork sometimes breaks, allowing the soft amalgam to run out.

Impairment of Contact. By far the most important cause of impaired contact is oxidation. This is greatly facilitated by the presence of moisture, so that the slightest crevice into which moisture may penetrate and lodge is dangerous. Soft-soldered contacts underground are not to be trusted, especially on track laid in streets, which is sure to be wet with dirty water, though there is no apparent reason why soldered contact entirely above ground should not be durable, if all traces of corrosive flux are removed. Amalgamated steel surfaces are not durable and soon rust in track exposed to dampness. Expanded or compressed terminal bonds, which have not been properly applied, may be loosened by the movement of the rail, and well-soldered, brazed or welded bonds may be loosened or torn off by the same means if they are too rigid. No mention has been made of accidental breakage of bonds. Of course, bonds which are improperly located may be knocked off by rolling stock, and various local external conditions may operate to destroy any kind of bond.

Length of Bond. A short rail bond is more economical of copper than a long one, but is more likely to damage from vibration of a loose joint. A short concealed bond should not be used where the rail joint is not rigidly supported. The length of a bond to be concealed between joint plate and rail is, under certain conditions, somewhat dependent upon the spacing of joint bolts. The length of bonds in use varies from 6 to 10 and more inches. The 10-in. bond is in very common use. Electrified steam roads are using bonds 16 to 24 in. long. Long bonds for spanning splice bars on small rails should be about 5 in. longer (formed) than the splice bar, and about 6 in. longer than the splice bar on large rails.

Resistance of Bonded Rail Joint. Neglecting the very unreliable conductivity between rail and rail by way of mechanical joint plates, the resistance of a newly bonded joint may be approximated as follows:

Brazed, Soldered or Welded Bond. Add 1 in. to the length of the bond shank conductor between terminals. The resistance of the bonded joint is approximately equal to the resistance of such a length of the shank conductor.

RESISTANCE IN INTERNATIONAL OHMS OF A CONTINUOUS STEEL RAIL AT 20° C. OR 68° F., NO JOINTS

Weight of rail, pounds per yard	Ratio of resistance of steel to that of copper									
	8		10		11		12		13	
	1000 ft.	Mile	1000 ft.	Mile	1000 ft.	Mile	1000 ft.	Mile	1000 ft.	Mile
50	0.013243	0.069923	0.016605	0.087674	0.018266	0.096444	0.019925	0.105204	0.021587	0.113979
60	0.011071	0.058455	0.013838	0.073065	0.015221	0.080367	0.016606	0.087680	0.017989	0.094982
70	0.009489	0.050097	0.011861	0.062621	0.013047	0.068888	0.014233	0.075152	0.015419	0.081412
80	0.008303	0.043834	0.010378	0.054796	0.011416	0.060276	0.012454	0.065757	0.013480	0.071174
90	0.007380	0.038966	0.009225	0.048708	0.010147	0.053576	0.011070	0.058449	0.011992	0.063318
100	0.006642	0.035070	0.008302	0.043835	0.009133	0.048222	0.009963	0.052604	0.010794	0.056992
110	0.006039	0.031886	0.007548	0.039853	0.008302	0.043834	0.009057	0.047821	0.009812	0.051807
120	0.005535	0.029224	0.006919	0.036532	0.007611	0.040186	0.008303	0.043839	0.008994	0.047488

AREA OF COPPER IN CIRCULAR MILS EQUIVALENT TO RAILWAY STEEL IN CONDUCTIVITY

Weight of rail, pounds per yard	Actual area in		Ratio of resistance of steel to that of copper				
	Square inches	Circular mils	8	10	11	12	13
50	4.90	6,238,800	779,850	623,880	567,164	519,900	479,908
60	5.88	7,486,600	935,825	748,660	680,600	623,883	575,890
70	6.86	8,734,400	1,091,800	873,440	794,036	727,866	671,876
80	7.84	9,982,100	1,247,763	998,210	907,463	831,841	767,853
90	8.82	11,229,900	1,403,737	1,122,990	1,020,900	935,825	863,838
100	9.80	12,477,700	1,559,712	1,247,770	1,134,336	1,039,812	959,823
110	10.78	13,725,400	1,715,675	1,372,540	1,247,763	1,143,783	1,055,800
120	11.76	14,973,200	1,871,650	1,497,320	1,361,200	1,247,766	1,151,784

Compressed or Pin Terminal Bond. To the resistance of a length of shank conductor equal to the length from center to center of terminals measured along the shank conductor add twice the contact resistance of one terminal.

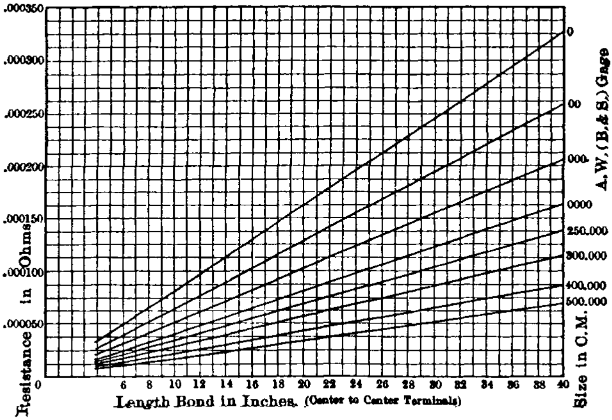


FIG. 54 —Resistance annealed copper rail bonds.

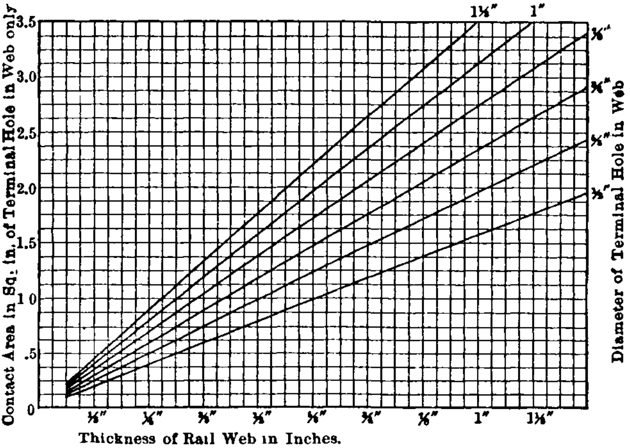


FIG. 55 —Contact area compressed or pin terminal rail bonds.

The resistance of the shank conductor may be determined from Fig. 54. Knowing the thickness of the rail web and diameter of terminal hole in web, the contact area (considering only area of hole wall) may be determined from Fig. 55. For thickness of web see rail dimensions on pp. 36–39. Common outside diameters of bond plug of compressed or expanded terminal bonds are as follows:

Size of bond, A.W G. (B & S.) and cir mils	Diameter of plug, inches
0	½, ⅝
00	½, ⅝, ¾, ⅞
000	¾, ⅞
0000	¾, ⅞
300,000	1
500,000	1

Tests made by Messrs. Hall, Smith and Starbird at the Worcester Polytechnic Institute indicate that the contact resistance between *soft steel and pure copper at pressures from 20,000 to 30,000 lb. per square inch* has values from ½ to 1 microhm per square inch.

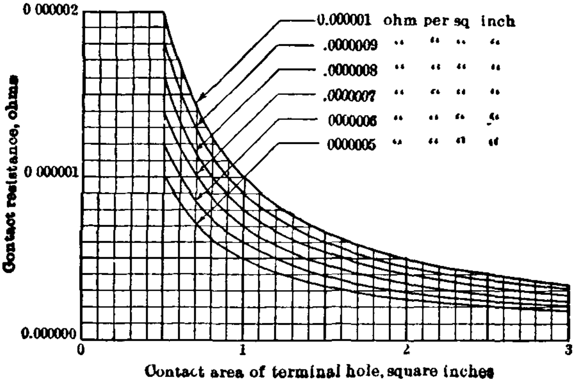


FIG. 56 —Contact resistance

These values are plotted in Fig. 56. The contact resistances of bond terminals vary from each other, depending upon the many factors in the process of installation. The value, 0 8 microhm (0 0000008 ohm), may be used in the practical calculation of the resistance of track return where bonds are carefully installed.

Resistance of Track Return. The resistance of a given section of track using both rails as return conductors is equal to one-half the resistance of the sum of all the joints plus one-half the resistance of the rail between joints contained in one of the rails of that section. *This may be determined by the following method: From one rail* length (ordinarily 30, 33, or 60 ft) subtract the length of one joint (distance between centers of the terminals of one bond installed), and by the table, page 649, find the resistance of the part of rail remaining (in the absence of more definite information the ratio of resistance of steel to that of copper may be taken as 11 for ordinary running rail) To this, add the resistance of one bonded joint (see Resistance of Bonded Rail Joint, page 648). The resistance of the given section of track will be equal to one-half this sum multiplied by the quotient obtained by dividing the length of the given section by the length of one rail. This may be conveniently expressed as follows:

$$R = \frac{r(l - l_1) + r_1}{2} \times \frac{L}{l}$$

in which R = resistance of section of track, ohms
L = length of section, feet
l = one rail length, feet
l_1 = length of one joint, feet
r = resistance of rail, ohms per foot. See page 649.
r_1 = resistance of one bonded joint, ohms.

The allowable value of the resistance, R, depends upon the current flowing and the allowable voltage drop. Its value is determined as follows:

$$R = \frac{e}{I}$$

in which R = the allowable resistance of the given section of track, ohms
e = the allowable voltage drop from one end to the other of the given section, volts
I = current flowing in track return, amperes.

The allowable voltage drop may depend upon the satisfactory operation of apparatus or it may depend upon local regulations or upon the economical amount of power loss. The power loss may be found as follows:

$$p = \frac{eI}{1000}$$

in which p = power lost, kilowatts.

(Significance of remaining symbols as above.)

Size of Bond. A common allowance for the area of cross-section of an ordinary joint bond is 500 cir. mils per ampere carried by the bond. Short bonds may be operated at a much greater current density. The latter is of value in handling rush loads of short duration.

The proper size and number of rail bonds per joint appears to be a subject regarding which very little definite knowledge exists, and the present practices of operating companies seem to be based upon arbitrary rules rather than upon theoretical considerations. W. A. Del Mar, in a letter published in the *Electrical World*, 1909, commented upon the absence of information and standard practice upon this point, and attempted to determine the proper capacity of bonds by the following three more logical considerations: (1) Make the bond large enough to conform to proper mechanical conditions. (2) Determine the magnitude of the continuous and maximum currents in the rail and make the bond large enough to prevent undue heat. (3) Make the bond large enough to give at least 90 per cent efficiency to the return circuit; that is, make the conductance of the bonded rails at least 90 per cent of the conductance of a theoretically continuous rail. The following equation is given by Mr. Del Mar for the efficiency of the return circuit:

$$Eff. = \frac{L_1}{L_1 + \frac{L_2}{K}(1 - K)}$$

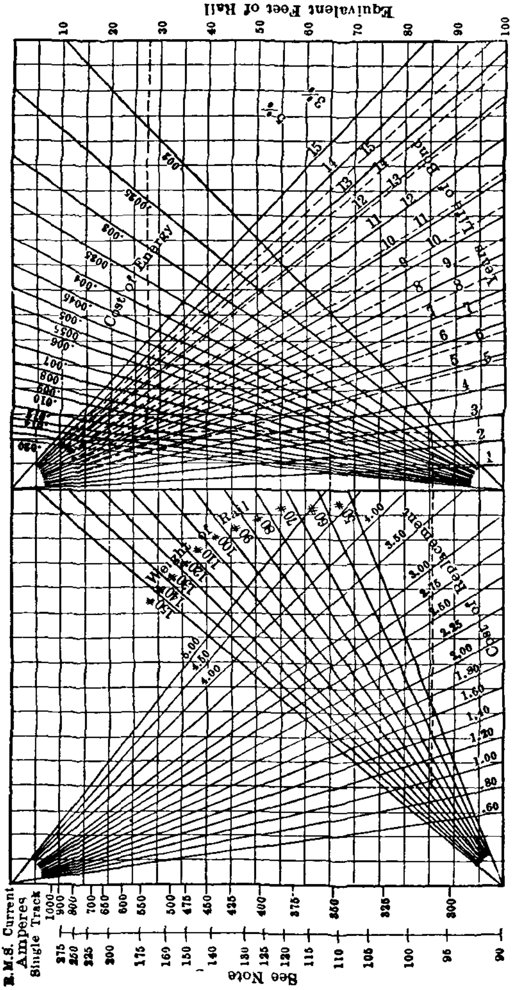
where K = average efficiency of bond, or the ratio of the conductance of the bond to the conductance of an equal length of rail

L_1 = length of rail

L_2 = length of bond between terminals.

That a short copper bond attached to heavy masses of cold steel cannot attain a dangerously high temperature is obvious even when carrying currents of the magnitude found on heavily loaded tracks. The contact resistance of one terminal of a mechanically applied bond in good condition is in the order of 0.000005 ohm. When carrying a current of 500 amp., which is greatly in excess of currents ordinarily found in rails, there would be a dissipation of only 1¼ watts, and with 1000 amp. a dissipation of 5 watts per terminal. This is, of course, in addition to the heat generated in the copper of the bond, but as far as the contact is concerned there seems to be no practical limit to its capacity to carry current. Parshall in England, in an article on "Earth returns for electric tramways," published in the *Journal of the Institution of Electrical Engineers*, 1898, stated that experience with pressure contacts in central station work had demonstrated that 100 amperes per square inch was the safe limit, but suggests that 50 and even 25 amp. per square inch would be found more advisable for rail bonds. As these figures are exceeded in practically all installations in this country they cannot be regarded in any way as a practical limit for current density. A number of companies, including some in large cities, install only single bonds, while some employ double bonding only when necessary to reduce the potential gradient on the tracks. This would indicate that double bonding is not considered necessary solely from the standpoint of capacity, but that its adoption is demanded by other considerations. One of the usual reasons for its use is to insure safety, and not place reliance on a single bond in a permanent track where the repair of a bond would mean tearing up the pavement. Under extreme conditions double bonding is justifiable solely from the economic standpoint, and the factors which determine these conditions may also be used to determine the economic replacement of deteriorating bonds. (See below.) These conditions can be determined when the constants of a given system are known and the cost of power and the average current in the rails are obtainable.

Economic Replacement of Bonds. The length of life of a rail bond in place may be anything up to the length of life of the rails it connects, depending upon the degree of perfection of the installation and the condition and care of track. The life of a bond may terminate at actual breakage or at the point at which its resistance becomes such that the voltage drop becomes too great or the saving of energy brought about by its renewal will justify the total cost of such renewal. The chart, Fig. 57, gives a method of determining the resistance of a bond in terms of feet of adjacent rail at which a bond should be renewed, considering financial economy only.



NOTE.—If amperes are taken in left-hand column, multiply ft. of rail by ten.

FIG. 57.—Economic replacement resistance of rail bonds.

It shows the increase in resistance over that of a well-bonded joint at which the energy saving amortizes the cost of replacement in a given number of years. The use of the chart is explained by the following example, solution of which is shown by the dotted lines on Fig. 57: R.m.s. current in track, 450 amp.; weight of rail, 90 lb. per yard; cost of energy saved, $0.0025 per kw.-hr.; cost of bond renewal, $2.00 per bond; estimated life of bond, 7 years; interest 5 per cent. Chart shows that joints in this case should be rebonded when resistance is (approximately) 27.6 ft. of rail *more* than resistance of newly bonded joint.

The U. S. Bureau of Standards, in its Technological Paper No. 62, gives a similar method of calculation as follows:

Let W = *weight of rail per yard*
I = root-mean-square current in rail over a 24-hour period
p = cost of power per kilowatt-hour in dollars
P = cost of installing a bond
n = number of years bond will last
L = reduction in joint resistance resulting from installation of bond, expressed in feet of adjacent rail
r = rate of interest paid on invested capital;

then the resistance of the rail is very close to $0.001/W$ ohm per foot and the annual saving of energy in dollars due to installing a bond would be

$$I^2 \times \frac{0.001L}{W} \times \frac{p}{1000} \times 24 \times 365 = \frac{0.00876I^2Lp}{W}$$

The annuity required to retire the investment, P, on a new bond, at the end of n years, its period of usefulness, is

$$P\left(\frac{R-1}{R^n-1}\right) \text{ where } R = 1 + \frac{r}{100}$$

When the annual power loss without the new bond exceeds this annuity, it is obvious that the installation of a new bond would be a matter of economy. The limiting condition would be obtained when the energy charge is equal to the annuity. Equating these two we get: $0.00876\frac{I^2Lp}{W} = P\left(\frac{R-1}{R^n-1}\right)$, from which any quantity may be obtained provided the others are known. If we employ the equation to determine at what current double bonding becomes economical, we have

$$I^2 = \frac{WP}{0.00876Lp} \times \frac{(R-1)}{(R^n-1)}$$

In order to apply this equation with six independent variable factors it is necessary to assume a set of values which would obtain under normal conditions. Let it be required to determine whether one or two 4/0, 10-inch compressed terminal bonds should be used on 100 lb. rails being newly installed under the following conditions:

p = cost of energy = \$0 01 per kw.-hr.
P = cost of installing bond = \$0 60
n = life of bond = *12 years*
r = rate of interest = 5 per cent

The resistance of one 10-inch 4/0 bond, including contact resistance, is very close to 0 000055 ohm, and would probably average more. Two such bonds in parallel would have approximately one-half of this, or 0 0000275 ohm, which is also the decrease in the resistance of the joint resulting from the installation of the second bond. As a 100 lb. rail has a resistance very close to 0 00001 ohm per foot, L would be 2 75 ft. Substituting these values in the above equation we find that I^2 is equal to 15,660 or I = 125 amp.

The Bureau of Standards has examined numerous railway load curves and has found that the ratio of the root-mean-square current to the all day average ranges from 1 25 to 1 4 If we use the lower of these values, which is more applicable for the heavily loaded lines with which we are concerned in this discussion, we get 100 amperes as the all-day average value of the current. With a load factor of 40 per cent this would give 250 amp. per rail as the value at the peak period. While the values here assumed are normal in every respect, they are undoubtedly on the side tending to make the limiting current small. A lower cost of energy, a shorter life of the bond, and a higher cost of installing a bond will tend to give a larger current as the point double bonding becomes economical As the values assumed for these three variables are obviously near the limit in the other direction, it is difficult to conceive how the economy point would be reached at any all-day average current value much less than 100 amp.

The value of the resistance which a deteriorating bond must reach before economy will justify its replacement can be determined from the above equation in which the reduction in the joint resistance resulting from the installation of the new bond becomes the unknown quantity. Transforming the equation for this purpose we find that

$$L = \frac{WP}{0\ 00876I^2p} \times \frac{(R - 1)}{(R^n - 1)}$$

The following concrete example will illustrate the application of this formula. Let us assume:

W, the weight of rail per yard = 100 lb.
P, the cost of replacing a bond = \$0 80
p, the cost of energy = 0 005 per kw.-hr.
n, the life of the new bond = 10 years
r, the rate of interest = 5 per cent

Let us also assume that the current is of such a value as to give a drop of 0 8 volt per 1000 ft as an all-day average on a perfectly bonded rail This is not ordinarily exceeded, even in regions where the problem of electrolysis does not exist The resistance of a 100 lb. rail is 0 01 ohm per 1000 ft which would limit the average current to 80 amp. Taking 1.3 as the ratio between the root-mean-square and the all day average current we get for our equation $I = 104$ amp. The replacement cost of a bond is usually greater than the cost on new work and is therefore taken at 80 cents. One-half cent per kilowatt hour for power may seem low, but as it is assumed to be the cost of energy which will be saved by the application of the bond, it would not be logical to load it with fixed charges and operating costs other than the fuel. Upon this basis it is high rather than low. A replacement bond is often installed on old track which is partially worn out and which may be entirely replaced within a few years. Ten years, therefore, is considered as a liberal life for the new bond. The scrap value of bonds is small and is neglected in this discussion. Substituting these values in the above equation we find that $L = 13.4$ ft. As the new bond itself will test equal to from 3 to 6 ft. of rail this means that the old bond will test equal to about 18 ft. of rail before a new one can be installed with economy. Electrolysis and voltage conditions ordinarily, therefore, demand a better return circuit than economy itself can dictate. In fact, it is doubtful if economy alone in many circumstances will justify any but the cheapest and simplest type of bonding.

Cross Bonding. In order to maintain the current in the two or four rails of the track return circuit as nearly equal as possible, and to care for a possible open bond, the rails should be bonded together by cross bonds. These make contact at the web of the rail or at the outside of the rail head. The spacing of cross bonds depends upon the density of traffic. In city work it is customary to place them about three per 1000 ft. They are spaced from this to 1500 ft., but commonly 1000 ft. in interurban work. Where track circuit signal systems are used the matter of cross bonding must be considered in its relation to the signal system. Cross bonding may be entirely prohibited or, in the case of alternating current signal systems, it may be provided by inductive bonds (see page 773).

Bonding at Special Work. Special work must be removed comparatively frequently, its joints are subject to great vibration and offer construction difficulties to bonding, it may be of steel having low conductivity, and it contains many joints. Thus, the bonding of joints through special work in a manner similar to that on ordinary track is unreliable. It has been found satisfactory to bond around special

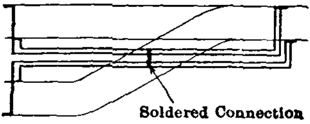


FIG. 58.—Bonding around switch.

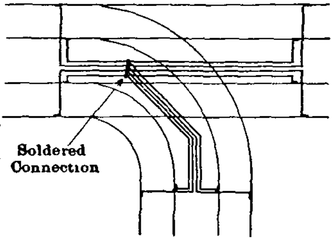


FIG. 59.—Bonding around branch-off.

work by means of bare stranded copper (500,000 to 1,000,000 cir. mil cable is commonly used for this work). Figs. 58 to 62, inclusive, give schemes for bonding around several typical pieces of special work. It is the practice of many roads to bond through as well as around special work. This is necessary in city work and is done by connecting each piece of rail in the special work to the cable which bonds around the special work, or by bonding the joints in the special work as in straightaway track construction. As in the case of cross bonding, where track signal systems are used, bonding at special work should be considered in its relation to the signal system. Much care should be taken to insure good contact at the terminals of cables used in bonding around special work. The following briefly outlines the tests and results of the Chicago Board of Supervising Engineers, 1911, in the determination of the proper method to use in attaching such auxiliary cables:

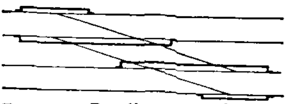


FIG. 60.—Bonding around crossover.

The tests comprised five different types of terminals.

First. Welded connection (Fig. 63) in which the weld is made under current of 15,000 amp. to 25,000 amp. applied with flux and spelter and a final compression of about 15 tons.

Second. The same type of joint with a considerable reduction in the final pressure applied.

Third. Welded connection in which the cable is enclosed by a mold containing an overflow chamber and clamped against the freshly cleaned web of the rail. Especially prepared copper is poured into this mold.

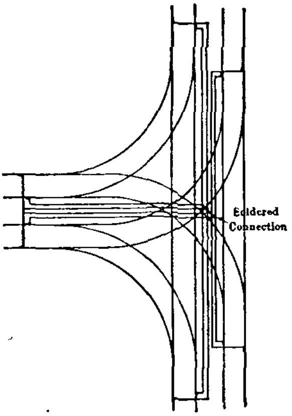


FIG. 61.—Bonding around Y.

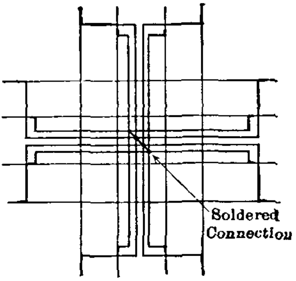


FIG. 62.—Bonding around crossing.

Fourth. Bolted type terminal (Fig. 63) constructed especially for this experimental work and so designed that the thickness of copper and the length of iron bolt are inversely proportional to the expansion coefficients of copper and iron, so that the changes in temperature will not tend to loosen the connection. This type was tested under varying compressions with both increasing and decreasing loads and plain as well as amalgamated surface.

Fifth. A multiple-pin type terminal, which was made up from standard bond terminals.

An analysis of the general results of the tests indicates:

First. A very large improvement in the conductivity—over 50 per cent—of Type 1, by reducing the final compression as noted under the description of Type 2.

Second. Under a pressure of about 75.000 lb. or 50,000 lb. per square inch the improvement in conductivity practically ceases.

Third. Bolted type terminals show ten to fifteen times greater unit conductivity than either of the welded type terminals and at about one-half the cost per terminal.

Fourth. The cable weld (Fig. 63) showed the greatest mechanical strength—13,200 lb. shear per square inch of contact.

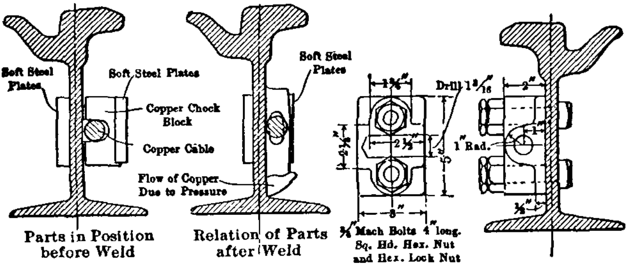


FIG. 63.—Welded and bolted cable terminals.

Fifth. A bolted type terminal can be developed which will be suitable for installation in limited numbers, while for a large number of connections the cable weld is most desirable on account of its higher mechanical strength and conductivity at lower cost.

As a result of these tests it was decided to adopt the electrically welded terminal as the standard. This standard is now in force on the Chicago Surface Lines.

Code of Bonding Precautions. To insure proper attention to details essential to proper bonding the bonding department should be provided with a code of bonding precautions. The following is an example of such a code by Howard H. George, *Electric Railway Journal*, 1914, for the installation of pin terminal bonds:

1. Every roadmaster and foreman should see that one or more men in each gang are taught the proper way of installing bonds, and should be sure that any bonding done thereafter is performed by these men.

2. When renewing rail or joint plates on single track in operation, care should be taken not to open or disconnect both rails at the same time, as this would open the return circuit by which the current returns from the cars to the power house. When it is absolutely necessary to open both rails, a long copper jumper should be installed to connect the open ends so that the path of the return circuit shall not be interrupted. This applies more particularly to road ends and interurban lines.

3. Whenever any track is opened up and any ground wires for electric lights, lightning arresters, or other electrical apparatus which should be connected to the rail, are found disconnected, they should be reported at

once to the bond inspector or distribution department, so that they may be repaired before the track is closed up. This is very important and should receive careful attention.

4. No bond holes should be drilled until just before the bonds are ready to be put in. There are, of course, times when it is desirous to have the holes drilled before the rail is placed on the ties. When this occurs, it is necessary to place a tight-fitting plug in the hole as soon as it is drilled to avoid any possible introduction of moisture. To drill a hole a day or two before and not protect it from moisture means a film of rust in the hole, which will greatly increase the resistance of the joint

5. Old bonds should never be used again because they become battered up in driving them out. Then, when they are put in again they will not make good contact with the rail, which means a poor bond. Where a bond is removed from the rail it is not advisable to use the same hole in putting in a new bond, unless some precautionary methods are used. The proper way is to drill a new hole, but as this is not allowable in some types of rails ream out the old hole and use a bond with a special large-sized terminal.

6. Great care should be taken with the drills used in making bond holes. If an improperly ground drill is used the hole will be irregular and oval shaped, thus giving a poor contact between the terminal and the rail. All dull and broken drills should be carefully boxed, labeled and sent to the shop to be reground, where the company has installed a special machine for the purpose to do the work perfectly and at much less expense than could possibly be done by hand.

7. In drilling bond holes never use oil to lubricate the drills. It is better not to use anything, but where it is absolutely necessary to use a lubricant, nothing more than a soda solution should be employed.

8. Holes, after being drilled, should be carefully cleaned of any chips, and wiped dry of any solution that may have been used to lubricate the drills. The holes must have a smooth and dry surface so that the bond terminal will make a good contact all around.

9. With a proper-sized hole, the bond terminal will make a very snug fit, not small enough to have to be driven with a heavy maul nor large enough to be put in easily with the hands. It should require a couple of taps with a hammer weighing about 3 lb. With a heavy hammer or spike maul, the head of the bond terminal is very likely to be battered, and the taper punch struck on the slant, causing it to split and bend the terminal.

10. After the bond terminals are in position, always drive the long steel taper punch entirely through the terminal, taking care to strike the punch squarely on the head. The small end of this punch should be dipped in some kind of heavy grease, such as track grease, just before it is driven through each terminal. The grease will lubricate the sides of the punch, thereby expanding the terminals and not drawing the copper with the punch.

11. Drive into each of the expanded terminals one of the short drift pins, thus expanding the copper a little more. This pin should be driven in until it is just flush with the head of the bond terminal.

12. The bond should then be shaped by straightening out the bond conductors, and forming them so that they will not be cut by either the track bolts or the splice bars. If it is a 36-in. bond, it should be so shaped that it will in no way interfere with the removal of the splice bar.

13. The bond, and particularly the bond terminals on both sides of the rail, are to be painted with some good weatherproof paint, care being taken to see that the paint fills the space back of the terminal heads.

Bond Testing. An inspection to determine the condition of the joint between bond terminal and shank may be made by sectioning the terminals in various planes, polishing the cut surfaces with a smooth file, emery, and crocus to remove all burrs, and then etching with a mixture of strong sulphuric and nitric acids, when the defective welds will show as fine black lines, and the actual welds and the form of the various component parts of the bond at that surface can be traced by the different colors of the metal after etching.

Bonded Joint Testing. Joints should be tested immediately after installation of bonds. At this time compressed or expanded terminal bonds should be tested for electrical resistance, and soldered, brazed and welded bonds should be subjected to both hammer and electrical resistance tests. Throughout the life of the bonds, joints should be kept under inspection to a degree depending upon the condition and character of the roadbed, the intensity of traffic, the cost of energy, danger from electrolysis, and the voltage drop allowable for good operation or permitted by local ordinance. There are two general methods of testing bonds: one, an aggregate test, is a test to determine the resistance of a whole section of track, thereby giving a general idea of its condition; the other is an individual test of each joint in the track.

Aggregate Test of Return Circuit. (See "Determination of Potential Drop in Rails," page 695.)

Individual Joint Bond Testing. An individual test is necessary to determine the exact condition of the individual bonds. Many methods and various types of apparatus have been devised for such testing. The resistance of the bonded joint is generally compared with the resistance of adjacent rail by comparing the voltage drops across this joint and in the rail, accompanying the flow of current through joint and rail

General Classification of Bond Testing Apparatus:

- Hand instruments
  - One-man instruments
    - Sight
      - Double millivoltmeters
      - Roller direct reading
        - Standard for returns with 40–50 amperes
        - High sensibility type using battery impressed 10–12 amperes
    - Sound
      - Conant
      - Crown
  - Two-man instruments
    - Sight
      - Differential millivoltmeter, balanced by measuring adjacent rail with steel tape
      - Double millivoltmeter, balanced in same manner
- Test Cars
  - Two classes of propulsion
    - Self-propelled
    - Trailer
  - Two classes current through joints
    - Circulation from low-voltage motor generator set
    - Resistance and propelling current
  - Four classes indication
    - Automatic recording and marking
    - Automatic signaling and marking
    - Automatic signaling and hand marking
    - Indicating and hand marking.

Typical Hand Bond Testing Instruments. The instrument consists essentially of two parts: first, a folding framework fitted with three steel rail contacts 3 ft. apart as shown in Fig. 64; and second, a standard differential voltmeter controlled by a two-point switch (in the circuit through the unjointed rail). As shown in Fig. 64, the frame is so placed on the rail that the joint is included beween two of the terminals with an equal length of solid stock rail between the center and the other outer terminal. If the voltmeter reading indicates a joint conductivity of 100 per cent or over, this is the only observation required. If less than 100 per cent, the voltmeter is connected up with a detachable prod by which contact is made at successive points along the rail until a balance appears on the differential voltmeter. The conductivity

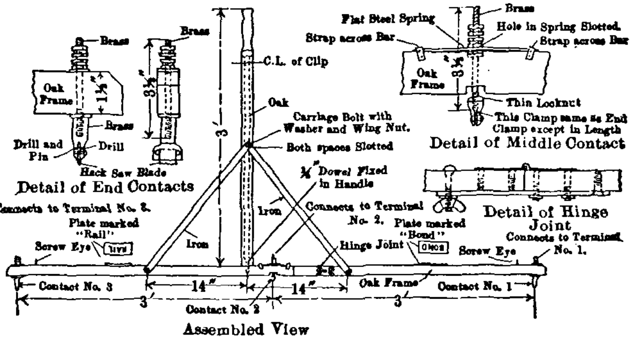


FIG. 64.—Contact frame for hand bond testing.

of the joint may be then computed from the relative rail distances included between points. An average of about 200 joints may be tested in a day of 8 hours on a fairly free track, which corresponds to slightly over a mile of single track a day per man.

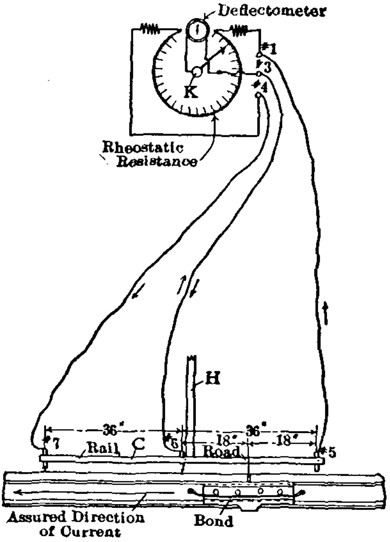


FIG. 65.—Hand bond testing circuits.

The scheme of connections of another typical hand bond tester (the Roller Smith) is shown in Fig. 65. It consists of a T-shaped bar, *C*, similar to that shown in Fig. 64, upon which three contacts are mounted, and connected with the electrical mechanism as shown. The contacts are made of short pieces of hack-saw blades held in clamps. By rocking the handle *H* of the bar at right angles to the rails the blades saw through the dirt and scale on the rails and insure good contacts. The principle of operation is that of the Wheatstone bridge. By turning the handle *K* the resistance of the two legs of the circuit is varied. The indicator attached to the handle K moves around a scale which is so graduated as to read directly the resistance of the section containing the bond in terms of the resistance of unbroken rail when the needle deflection is zero.

Bond Test Car. A test car makes possible a rapid and complete report of the entire track by making a continuous record of track condition and location as the car proceeds at a speed of 8 or 10 miles per hour. As a motor-generator set may be carried, its operation may be independent of the current in the track due to ordinary traffic and it may be operated where traffic conditions are such as to make hand bond testing highly difficult or impossible. Tests may be made by it with great rapidity, thus affording knowledge of the true condition and location of bonds of a whole system in a very short time.

Electrolysis

The electrolytic corrosion of earthed metallic conductors proceeds at a rate dependent upon the rate of flow of the current which leaves the metallic conductor for the earth.

Theoretical Rate. With an oxidizing anode the weight of anode corroded by 1 ampere in 1 second is equal to the electro-chemical equivalent of the metal of the anode. The amount of metal (pounds) corroded by 1 ampere in 1 year is equal to the electro-chemical equivalent of the metal (grams) multiplied by $60 \times 60 \times 24 \times 365 \times 0.0022046$, the latter figure being the conversion factor between grams and pounds. The product of these factors is 69,524, which may be termed the conversion factor between the electro-chemical equivalent of a metal as usually expressed in grams, and the theoretical amount of such metal, in pounds, which would be corroded by the electrolytic action of 1 ampere in 1 year. The values are as follows:

Metal	Electro-chemical equivalent, grams	Corroded by one ampere per year, pounds
Iron (ferrous)	0.0002894	20.1
Iron (ferric) *	0.0001929	13.4
Lead	0.0010718	74.5
Copper	0.0003293	22.9
Aluminum	0.0000935	6.5

* Less commonly met with than ferrous.

Variations from Theoretical Rate. McCollum and Logan show results of some very extended studies by the U. S. Bureau of Standards on the variations from the above theoretical rate. Some of their conclusions are as follows, their term "efficiency of corrosion" being based on the above theoretical rate as 100 per cent. They state that the current density has a marked effect on the corrosion of iron in soils, the efficiency of corrosion being in general greater as the current density is lower. In saturated soil the corrosion may vary between 20 and 140 per cent for a range of current density varying from about 5 to 0.05 milli-amperes per square centimeter. Moisture content also has a marked effect on efficiency of corrosion, it being in general greater with increased moisture content up to saturation of soil. Temperature changes within the limits commonly encountered in general practice have no marked effect on corrosion efficiency, neither has the depth of burial of pipes, other conditions remaining constant. The amount of oxygen present has no appreciable effect on the efficiency of corrosion in the case of iron immersed in liquid electrolyte, but it has a marked effect on the end products of corrosion. If the corrosion is rapid and supply of oxygen small, there will be a preponderance of magnetic oxide, while if the rate of corrosion is low and the supply of oxygen abundant, the ferric oxide will predominate. Owing to the fact that the supply of oxygen around pipes buried in earth is always more or less limited, the character of the oxides formed gives some indication as to the rate of corrosion, and thus indirectly as to the cause of the corrosion if local conditions are properly considered. The efficiency of corrosion was found not to be a function of the voltage, except in so far as the current density may be affected. Voltages as low as 0.1 to 0.6 volt showed practically the same efficiency of corrosion as 5 to 10 volts or higher. Corrosion tests on a large number of different kinds of soil from widely different sources with average moisture content and current density indicated that corrrosion efficiency between 50 and 110 per cent may usually be expected under most practical conditions.

Resistance of Soils. With a given potential difference between two earthed metallic structures, the amount of current which will flow between them is dependent not only on the contact resistance but also upon the resistance of the soil. Relative to the latter, McCollum and Logan state that the resistance of soils varies throughout a very wide range with variations in moisture content, the resistance of comparatively dry soil being of the order of several hundred times the resistance of the same soil at about saturation. Above saturation increase in moisture content has but little effect on the resistance of the soil. The resistance of the soil varies greatly with temperature within the ordinary range encountered in practice. In the case of the soils tested the resistance at 18 deg. below zero C. was over two hundred times as great as at 18 deg. above zero C. Even at about freezing temperature the resistance will be several times that at summer temperatures. This has an important bearing on the magnitude of the electrolysis trouble that may occur at different seasons and also indicates that where practicable voltage surveys should not be made when extremely low temperatures prevail. These authors give a table of the specific resistance of soils as found in Philadelphia, Washington, St. Louis and elsewhere, showing it to vary between the extremes of 400 ohms per cubic centimeter for a St. Louis blue clay with 26 per cent moisture, and 2,340,000 ohms per cubic centimeter for Washington air dry red clay with 4 per cent moisture. The majority of soils tested show resistivities of between 1000 and 5000 ohms per cubic centimeter.

Contact Resistance. Polarization and film resistances at the surface of the pipes may be an important factor in current flow. As soon as an electromotive force is applied to a buried pipe the current flow drops off rapidly with time, especially during the first few minutes, due to the setting up of counter electromotive forces and the formation of film resistances. McCollum and Logan show the effective resistance as a function of time after the application of about 6 volts between two short lengths of cast iron pipe buried about 12 ft. apart. The initial resistance of about 18 ohms practically doubled within half an hour after the voltage was applied and after that the resistance remained practically constant. In this case the effect of polarization and film resistance was practically as great as the total soil resistance between the pipes.

The Character of the Electric Railway Roadbed is an important factor in determining the extent of leakage of stray current into the earth. A well drained rock or concrete roadbed may in general be expected to offer much higher resistance to the leakage of current than one in which the construction is such that a large amount of moisture is retained.

The U. S. Bureau of Standards presents the following conclusions on this subject. Roadbeds constructed with solid concrete ballast and vitrified brick or other nonporous pavements have a low leakage resistance to earth, which is affected only moderately by seasonal and weather changes. There is little difference between wood and steel ties in their effects on the resistance of roadbeds of this kind. Insulating layers of bituminous materials are not of practical value in reducing leakage currents from such roadbeds. The resistance of 1000 feet of single roadbed of this type is from 0.2 to 0.5 ohm under ordinary conditions, but may be double or treble these values when the ballast is frozen to a depth of 1 foot or more. For double roadbed of this type the resistance is approximately 70 per cent of that for single roadbed, or the leakage from double track would be about 40 to 50 per cent greater than from single track. Roadbeds constructed with a foundation of clean, crushed stone under a concrete paving base have about three times the resistance of those with a solid concrete ballast. Roadbeds with a full crushed stone ballast and a Tarvia finish have a very high leakage resistance, of the order of 2 to 5 ohms for 1000 ft. of single track. The leakage from a double roadbed of this and other high resistance types is from 80 to 100 per cent greater than from single roadbeds. The resistance of earth roadbeds in which the ties are embedded and therefore kept in a moist condition, is much lower than that of open construction roadbeds, being from 1 to 1.5 ohms for 1000 ft. of single track under normal conditions and considerably more when the ground is frozen. The resistance of roadbeds of open construction is subject to wide variations, depending on the condition of the ties and ballast. In very dry weather with good ballast the resistance will be 10 to 15 ohms and even more for 1000 feet of single track, but when wet will drop to from 3 to 5 ohms. Cinders, gravel and particularly crushed stone, when used as ballast in open track construction, will produce very high resistance roadbeds. Earth has a tendency to keep the ties moist and therefore increases the leakage. Open construction track is often considered as being insulated from the earth, but this is not strictly true, even though the leakage may not be in harmful amounts when compared with other types of construction. Assuming a potential difference between a track and the earth of 5 volts and a leakage resistance of 10 ohms per 1000 ft. of single roadbed, the total leakage per mile of track would be 2.64 amp. This small leakage current would not ordinarily be harmful to underground structures in the vicinity of the track. Zinc chloride and other chemical salts used as preservatives render ties highly conducting and greatly increase leakage currents from tracks. Unless combined with some other material, such as creosote, these salts gradually leach out, particularly in damp climates, and eventually their influence on the resistance of roadbeds disappears. Creosote has very little effect upon the resistance of wood ties, but a treating material consisting of 75 per cent gas oil and 25 per cent creosote appears to increase their resistance materially. No direct comparison, however, was made between these two treatments. Open track construction on which ties treated in this manner were employed had a leakage resistance about twice as great as similar roadbeds with untreated ties and about four times as great as roadbeds with chemically treated ties.

The Bureau of Standards suggests that electric railway companies can do much toward reducing leakage currents from their tracks by observing the following regarding roadbed construction: Solid concrete ballast should be abandoned, and clean crushed stone should be used as a foundation under ties. This type of construction is approved by the Am. El. Ry. Eng. Assn., as it gives greater resiliency to the track and is cheaper than the full concrete ballast. Where crushed stone or gravel is used it should be kept clean by proper coverings or pavements. If earth, sand, or street dirt is permitted to filter into ballast of this character, its function as an insulating material is greatly impaired. Salt, which is often used to prevent frogs and switches from freezing, will greatly reduce the resistance of roadbeds and should be avoided if possible. In open construction rails should be kept out of contact with the earth. The roadbed should be well drained to prevent fine material from washing into the ballast and to keep the ties as dry as possible. Vegetation should be kept down, as this tends to make the roadbed moist and to fill the ballast with foreign material. Zinc chloride and similar chemical preservatives should be avoided where the escape of stray currents is objectionable or where block signals are used. A treating mixture of creosote and gas oil improves the insulating properties of wood ties.

Self-corrosion is very generally regarded as being due primarily to the presence of local galvanic currents at the surface of the corroded metal, due either to physical differences between adjacent points on the surface of the metal or to foreign conducting substances in the soil. For example, if a piece of carbon in the form of coke be embedded in the soil in contact with the pipe surface at one point there will be a difference of potential between the coke and the pipe, and a current will flow from the pipe through the moist soil to the coke and return to the pipe through the contact point between the two. The action here is exactly analogous to the action in a primary battery in which a piece of zinc and a piece of carbon are immersed in the electrolyte and connected together through an external circuit. A current flows through the electrolyte from the zinc to the carbon, thereby corroding the zinc. The electromotive force given by iron or steel when embedded in ordinary soils in contact with coke is usually about 0.6 volt, and this is sufficient to give rise to very rapid corrosion and pitting of the iron surface. Action of this kind is frequently encountered where pipes are embedded in soils containing cinders in which particles of coke may be found. Galvanic action of this kind may take place as a secondary action following electrolysis from stray currents. If stray currents are discharged from a pipe, producing initial corrosion, the corroded iron thus carried into solution in the soil comes in contact with oxygen dissolved in soil water which results in the precipitation of iron oxide on the surface of the pipe. This iron oxide is a conductor of electricity, and it also exhibits an electromotive force against the iron, as in the case of the piece of coke. Hence, where iron oxide is thus deposited on the surface of the iron at any point, corrosion may continue to some extent even though the stray currents which initiated the trouble have been removed. Self-corrosion may also occur from galvanic action due to physical differences between different points of the surface of the iron itself, or due to differences in the electrolyte in the soil near to adjacent portions of the iron surface. Unfortunately, self-corrosion generally manifests itself in a manner very similar to that caused by stray currents. Pipe corroded under conditions such that no stray currents could exist exhibits pitting very similar to that caused by electrolytic corrosion. Because of this similarity in appearance it is not possible in general to determine by inspection of a corroded pipe whether or not the corrosion was caused by stray currents or by local galvanic action. Owing to this fact it not infrequently happens that cases of pipe corrosion are charged to the railway companies when the damage was actually due to local causes arising from the nature of the soil or of the pipe, or both.

The only sure way of determining whether or not stray currents are causing corrosion in any particular case is by making proper electrical tests to determine whether or not the pipes are actually discharging current into the earth. In a case where serious corrosion has been caused by stray currents and the cause of these stray currents later removed, the only certain way of determining whether or not the previous corrosion was caused by stray currents or by local influences is by making actual corrosion tests in the soil under the same average conditions of moisture and using the same kind of iron as was previously found corroded. In the absence of a test of this kind it is not possible to fix with certainty the cause of the damage.

Electrolysis in Concrete. Following an extended investigation in this matter, the U. S. Bureau of Standards presents the following conclusions: 1. The observations of previous investigators that the passage of current from an iron anode into normal wet concrete caused the destruction of the test specimen by cracking the concrete were only partly confirmed. This effect was found not to occur in most of the specimens tested when the potential gradient was less than about 15 volts through a distance of 3 in., or about 60 volts per foot. These figures must be considered as but roughly approximate as they depend much on conditions. 2. Of the numerous theories that have been advanced for the cracking of reinforced concrete due to electric current that one which attributes it to oxidation of the iron anode following electrolytic corrosion has been fully established. The oxides formed occupy 2.2 times as great a volume as the original iron, and the pressure resulting from this increase of volume causes the block to crack open. 3. Metals which do not form insoluble end products of corrosion and all noncorrodable anodes never cause cracking of the concrete as a result of the passage of an electric current. 4. The mechanical pressure developed at the iron anode surface by corrosion of the iron has been measured in a number of cases and has been found to reach values as high as 4700 lb. per square inch, a value more than sufficient to account for the phenomena of cracking that have been observed. 5. Suggestions of some engineers that copper-clad steel or aluminum be used as reinforcing material have been shown to be impracticable, since the copper coating is readily destroyed and the aluminum is attacked by the alkali in the concrete. 6. Corrosion of iron anodes even in wet concrete is very slight at temperatures below about 45 deg. C. (113 deg. F.). 7. For any fixed temperature the amount of corrosion for a given number of ampere-hours is independent of the current strength. 8. The lack of corrosion of the iron at temperatures below 45 deg. C. is due to the inhibiting effect of the $Ca(OH)_2$ and possibly other alkalies in the concrete. 9. The rapid destruction of anode specimens of moist concrete at high voltages (60 to 100 volts or more) is made possible mainly by the heating effect of the current, which raises the temperature above the limit mentioned above. If the specimen be artificially cooled no appreciable corrosion occurs, and no cracking results. 10. The potential gradient necessary to produce a temperature rise to 45 deg. C. with consequent corrosion, in the specimens used, was about 60 volts per foot. For air-dried concrete it is much higher. This shows that under actual conditions corrosion from stray currents may be expected only under special or extreme conditions as noted below. These figures are but roughly approximate since they will vary greatly with the conditions, such as the size, form, and composition of the specimen, but they serve to show the order of magnitude of the voltage required to produce trouble. 11. Since the passivity of iron in concrete is due chiefly to the $Ca(OH)_2$ present it appears probable that old structures in which the $Ca(OH)_2$ has been largely converted into carbonate will be more susceptible to the effects of electric currents than comparatively new concrete with which the foregoing experiments have been made. The increase in the efficiency of corrosion would, however, be at least partly offset by the increase in the resistance of the concrete which would accompany the change. 12. The addition of a small amount of salt (a fraction of 1 per cent) to concrete (as is frequently done to prevent freezing while setting) has a twofold effect, viz., it greatly increases the initial conductivity of the wet concrete, thus allowing more current to flow, and it also destroys the passive condition of the iron at ordinary temperatures, thus multiplying by many hundreds of times the rate of corrosion and consequent tendency of the concrete to crack. Salt, therefore, should never be used in structures that may be subjected to electrolytic action. Further, reinforced concrete structures built in contact with sea water, or in salt marshes, are more susceptible to electrolysis troubles than concrete not subjected to such influences. 13. Specimens of normal wet concrete carrying currents increase their resistance a hundredfold or more in the course of a few weeks. 14. The rise of electrical resistance is probably due to a number of causes among which are the precipitation of $CaCO_3$ within the pores of the concrete, thus plugging them up. A slight amount of salt tends to prevent this precipitation and interferes with the rise of resistance, thus still further emphasizing the detrimental effect of salt. 15. Contrary to the observations of previous investigators there was a distinct softening of the concrete near the cathode. This begins at the cathode surface and slowly spreads outward, in some cases as far as one-fourth inch or more. After exposure to the air this softened layer becomes very hard again, but remains brittle and friable. 16. The softening effect at the cathode noted above caused, under the conditions of the experiments, practically complete destruction of the bond between reinforcing material and the concrete, reducing it to a few per cent of its normal value. 17. Unlike the anode effect which becomes serious in normal concrete only on comparatively high voltages, the cathode effect develops at all voltages used in the experiments, the rate being roughly proportional to the voltage in a given specimen. 18. In general the cathode effect occurs under conditions which may not infrequently occur in practice and is therefore probably a more serious matter practically than the anode effect about which so much has been written. This trouble is unlikely to be serious, however, except where the concrete is wet and the potential differences rather large. 19. The softening of the concrete at the cathode is due chiefly to the gradual concentration of Na and K near the cathode by the passage of electric current. In time the alkali becomes so strong as to attack the cement. 20. Softening at the cathode is increased by increasing the Na and K content of the cement, and reduced by diminishing this content, at least within the range below 10 per cent of the total salts. 21. The softening of the concrete has never been observed except very close to the cathode, the main body of the concrete remaining perfectly sound. Numerous tests show conclusively that the crushing strength of the main body of the concrete is not reduced even when the potential gradient is maintained at 175 volts per foot for over a year. 22. Because of the cathode effect noted above, the proposal to protect reinforced concrete buildings by maintaining the reinforcing material cathode as a by battery or booster would be much more dangerous than no protection at all. 23. Aside from slight heating, which is usually negligible, the only effect which an electric current has on unreinforced concrete is to cause a migration of the water soluble elements. Consequently, in the absence of electrodes, the ultimate effect of current flow on the physical properties of the concrete is not materially different from that of slow seepage, which also removes the water soluble elements. Nonreinforced concrete buildings are therefore immune from trouble due to stray earth currents. They might, however, be injured by the grounding of power wires within the structure since these or the inclosing conduits would then act as electrodes. 24. Conditions arise in practice which give rise to damage due to stray currents, but the danger from this source has been greatly overestimated. While precautions are necessary under certain conditions, there is no cause for serious alarm. 25. If reinforced concrete could be thoroughly waterproofed, it would greatly increase its resistance and diminish accordingly the danger from either the anode or cathode effects. It should be emphasized, however, that waterproofing to prevent electrolysis is a much more difficult matter than waterproofing to maintain a moderate degree of dryness, because of the much higher degree of waterproofing required in the former case. It has been found that practically all of the waterproofing agents now on the market that are intended to be mixed with the concrete are of little value as preventives of electrolysis. Waterproofing membranes, etc., applied to the surface can be made more effective and when properly applied may have considerable effect in preventing the entry of earth currents into the concrete. 26. Painting or otherwise coating iron with an alkali resisting metal preservative before mbedding it in concrete may serve to minimize the dangers of electrolysis, but no such coating has been found that does not prevent the proper formation of the bond between the concrete and iron when the concrete sets. 27. In order to insure safety of reinforced concrete from electrolysis the investigation shows that potential gradients must be kept much lower in structures exposed to the action of salt waters, pickling baths, and all solutions which tend to destroy the passive state of iron. 28. All direct current electric power circuits within the concrete building should be kept free from grounds. If the power supply comes from a central station the local circuits should be periodically disconnected and tested for grounds and incipient defects in the insulation. In the case of isolated plants ground detectors should be installed and the system kept free from grounds at all times. 29. All pipe lines entering concrete buildings should, if possible, be provided with insulating joints outside the building. If a pipe line passes through a building and continues beyond, one or more insulating joints should be placed on each side of the building. If the potential drop around the isolated section is large, say, 8 or 10 volts or more, the isolated portion should be shunted by means of a copper cable. 30. Lead covered cables entering such buildings should be isolated from the concrete. Wooden or other nonmetallic supports which prevent actual contact between the cable and the concrete will give sufficient isolation for this purpose. Such isolation of the lead covered cable is desirable for the protection of the cable as well as the building. 31. The interconnection of all metal work within a building is an advantage where practicable, provided that all pipe lines entering the building are equipped with insulating joints and lead cables are taken care of as indicated in the preceding paragraph, but the grounding of such interconnected metal work or any part of it to ground plates or to pipe lines outside of the insulating joints is to be strictly avoided. 32. In making a diagnosis of the cause of damage in any particular case, the fact that a fairly large voltage reading may be obtained somewhere about the structure should not be taken as sufficient evidence that the trouble is due to electrolysis. The distance between the points, and particularly the character of the intervening medium, are of much greater importance than the mere magnitude of the voltage reading. As a precautionary measure, however, all potential readings about a reinforced concrete structure should be kept as low as practicable.

Electrolysis Tests

Usual Polarities. With the common arrangement of connecting the positive terminal of the railway generator to the trolley and the negative to the rails, the general path of stray currents is from the rails through earth to pipes or cable sheaths at places distant from the power station, through the pipes or cable sheaths and from these through the earth back to the rails or other return conductors in the vicinity of the power station. Where current tends to flow from the rail to a buried pipe or cable sheath the rail has a positive potential with reference to the buried conductor which is negative. This is generally known as the negative district. Where current tends to flow from a buried conductor to the rail the buried conductor has a positive potential with reference to the rail, and this is usually known as the positive district. The intermediate district where the potentials may shift from positive to negative is sometimes called the neutral district.

Potential Survey. As the polarity of the earthed conductor is indicative of the tendency of current to flow, and electrolytic corrosion only takes place where current leaves the current conductor for earth, the first set of tests is generally a set of potential readings and is called potential survey. It is permissible and the usual practice to use hydrants or service connections for a contact to underground piping system. These readings usually are made with a low-reading voltmeter, preferably with the zero indication in the center of the scale, and readings are taken in a large number of places throughout the system between the rails and the buried conductors (pipes or cable sheaths). The voltmeter used should preferably have a high resistance to minimize the effect of accidental poor contact. A Weston high resistance zero center voltmeter with ranges of 1.5, 15 and 150 volts is a very satisfactory instrument. Readings should be taken at intervals of a few seconds for several minutes at each point and notation made of the location and of the maximum, minimum and average readings. It is desirable to plot these potential readings graphically on a map similar to Fig. 66, where the differences of potential between city water mains and street railway tracks are shown graphically plotted with the latter as a base line. It is also desirable to show on this map the size and location of the various pipe systems. A more detailed map of this character is shown in Fig. 67 which is an actual potential survey in the Grand Avenue substation district of the Chicago Railways Company. This figure is reproduced from the Fourth Annual Report of the Board of Supervising Engineers, Chicago Traction.

Potential Readings Merely Indicative. It should be remembered always that the potential difference between pipes and rails, even if large, is not conclusive evidence of stray currents, but is only an indication of the points at which current may be flowing from rails to pipes or from pipes to rails or between other conductors. In fact, a high potential reading is generally an indication of high earth resistance and consequently a small current flow rather than of a large current flow.

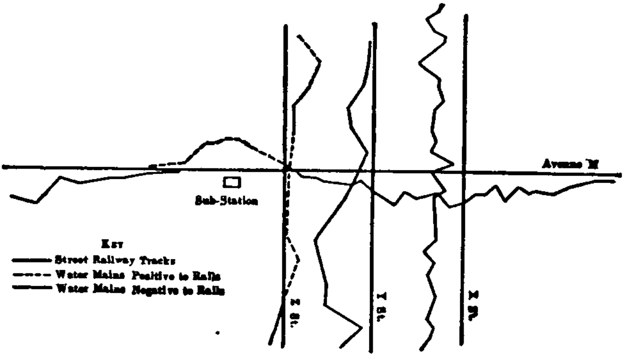


FIG. 66.—Sample potential survey.

E. R. Shepard (*Elec. Ry. Jour.*, 1923) points out that (1) lateral, as against longitudinal, potential gradients in the earth are directly responsible for stray current electrolysis. (2) The function of underground structures in the longitudinal transmission of stray currents is small and almost negligible in comparison with the part played by the earth, unless such structures are electrically drained to the railway return circuit. (3) Piping systems in general assume the average potential of the earth in which they are buried and limited portions of such systems should be considered as equipotential bodies and not subject to the same potential gradients that exist in the earth. (4) Potential gradients in the earth are in general very much smaller in a direction parallel to street railway tracks than at right angles to them. (5) The familiar over-all voltage curves showing the potentials of tracks, earth and pipes are misleading, as such potential profiles cannot be made to apply to the earth as a mass nor to the structures as a network without rigid qualifications. Such profiles are applicable only to line conductors such as a wire or an electric railway line.

Mr. Shepard says that in some cases water mains have been found to be collecting current on one side and discharging on the opposite

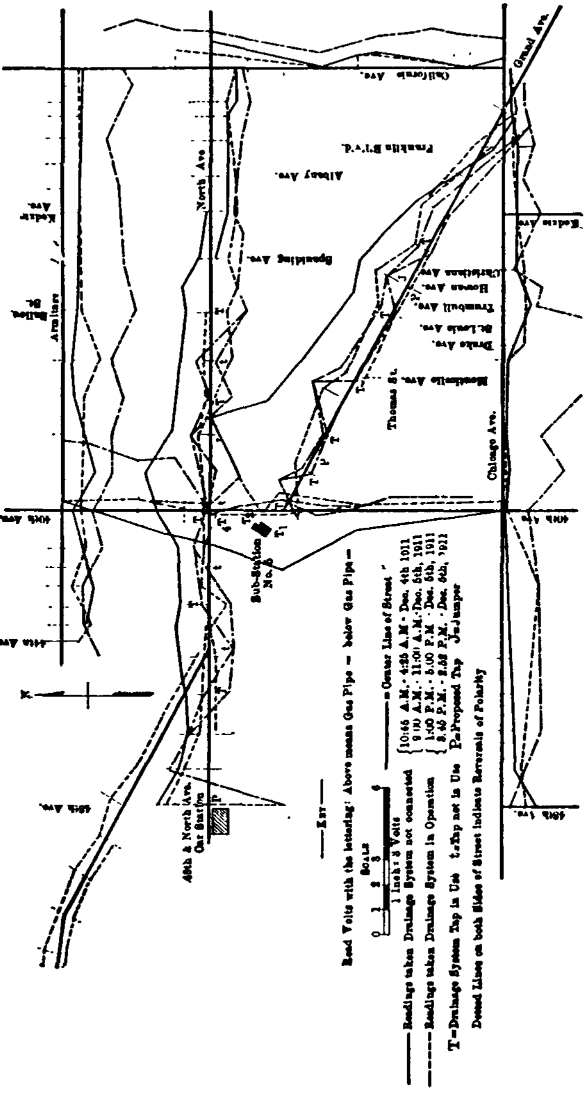


FIG. 67.—Sample potential survey.

side, although potential difference measurements show them to be strongly positive to the track. An example of this kind was observed in New Orleans in 1922, and is illustrated in Fig. 68. A 6-in. cast iron main 3 ft. deep and paralleling an electric railway track at a distance of about 4 ft. was 1.65 volts positive to the track. The main was uncovered and the end of the trench squared up. A polar diagram of current discharge was obtained by taking readings at 15 deg. intervals with the non-polarizing electrodes and the earth current meter. The main was found to be collecting current on one side and discharging it toward the track on the other side, the current density at different points being shown in the diagram in milliamperes per square foot of pipe surface. This condition is typical of many other observations made on pipes paralleling electric lines.

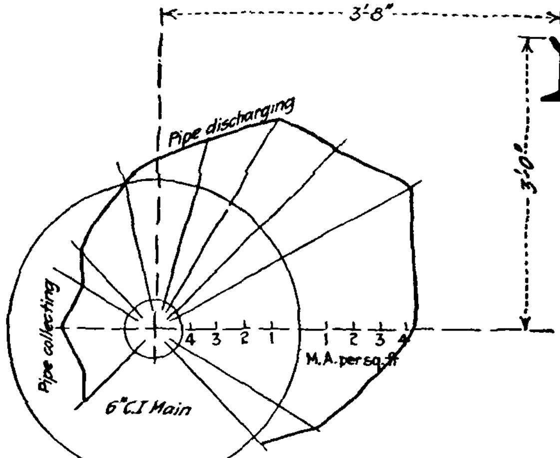


Fig. 68.—Polar earth current diagram showing transverse electrolysis.

Direction and Relative Magnitude of Current Flow in Underground Conductors. Tests to determine the direction of current flow in underground conductors may be made by measuring potential differences between two points in the underground conductor. A zero center Weston milli-voltmeter with scales of 10 and 100 millivolts is a satisfactory instrument for this test and connections may be made to the cable sheath in two adjacent manholes or on the piping system between hydrants or service connections 100 or 200 ft. apart. These readings which are clearly indicative of the direction of current flow may be used in the calculation of the amount of current flow in the case of cable sheaths where the resistance per foot can be quite accurately known and is not seriously affected by joint resistances. In the case of pipes, however, such readings can only be used as an indication of the relative magnitude of currents because, especially in the case of the ball and spigot joint usually used in cast iron pipes, the resistances vary so greatly that it is not possible to make any accurate assumption as to the resistance of a considerable length of pipe, including the joints.

Current Flow in Pipes. In only one way can an accurate determination be made of the current flow in underground piping systems, this method with its modifications being a determination of the potential drop in a continuous length of pipe between joints, and the application of Ohm's law, knowing the resistance of the pipe between the points of contact. Knowing the weight of the pipe per foot, exclusive of hubs or joints, the resistance per foot may be calculated by dividing the resistance in ohms per pound-foot by the weight per foot. The following is a table of resistances in ohms per pound-foot of various pipe materials:

Cast iron	0.00144 ohm per pound-foot
Wrought iron	0.000181 ohm per pound-foot
Steel	0.00021 ohm per pound-foot
Lead	0.00048 ohm per pound-foot

A table showing the resistance per foot of various sizes and weights of American Water Works Association cast-iron pipe, American Gas Institute standard cast-iron pipe, standard wrought-iron pipe and standard pipe steel is shown (pp. 677 et seq.) in order to simplify these calculations. In making these current measurements, it is necessary to have a perfect metallic contact between the pipe and the milli-voltmeter leads, and it is therefore necessary to expose the pipe where current measurements are to be made. The best contact is obtained by soldering the leads directly to the pipe, especially where readings are to be taken over a considerable period of time. It is sometimes convenient to carry these milli-voltmeter leads as rubber-covered wires through a conduit to a box at the curb so that later readings may be taken conveniently. In such cases it is necessary to have a calibration of the milli-voltmeter including such leads.

Where measurements are to be made in a pipe, the weight or resistance of which is not definitely known, and it is required that the determination of current be made with accuracy, a determination of the resistance and current may be made by one of the methods described by Dr. Carl Hering (Trans. A.I.E.E., 1912) as follows:

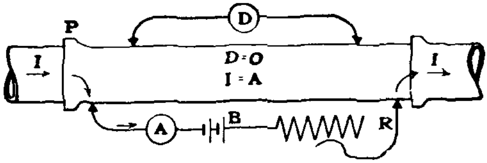


FIG. 69.—Determination of current flow in pipes.

The fundamental principle is as follows: Let *P*, Fig. 69, be a part of an underground pipe which has been uncovered and through which an unknown current *I* is flowing as shown; at first let it be supposed that this current is steady, and of course a direct current. Let *D* be a sensitive galvanometer, milli-voltmeter or any other form of detector of small differences of potential, connected as shown; there should preferably be no variable resistance like an unbonded pipe joint between the two contact points. Let *A* be an ammeter, *B* a few cells of storage battery and *R* an adjustable resistance; the shunt circuit containing them is connected as shown anywhere outside of the points of application of the voltage detector, the farther away the better—they may even be on the other side of a joint. To find the current flowing in the pipe adjust the resistance *R* until *D* reads zero; then there will no longer be any current flowing in the shunted part of the pipe, hence the reading of the ammeter will give the current *I* in the pipe. The current may be said to have been sucked out of the pipe by the battery, and made to flow through the ammeter, where it can be measured; as far as the current in that short section is concerned, the pipe circuit has in effect been electrically cut in two as though an insulating joint had been introduced. If *D* is a galvanometer with proportionate deflections, instead of a mere detector, then by taking a deflection immediately after the shunt circuit has been opened a reading proportionate to the drop of voltage for that current will be obtained. The instrument *D* is thereby calibrated to read the pipe currents directly and can be used for this purpose thereafter; the test with the battery current is therefore merely of the nature of a preliminary calibration and need be carried out only once for each station. If in addition this voltage instrument is calibrated to read directly in volts (usually in terms of milli- or microvolts as for instance a milli-voltmeter) then if the deflection reduced to millivolts is divided by the current it will give the resistance of the pipe in milliohms between the two points of application of the voltmeter, hence it enables the true resistance of the pipe to be measured.

In this method shown in Fig. 69 there should be no current leaving or entering the pipe between the points of application of the shunt, hence the hole should be free from water and there should be no moist earth in contact with that part of the pipe. It is assumed in this method that the short length of pipe between the shunt contacts is too small a fraction of the whole circuit of the pipe line currents to alter these currents when that section is practically cut out of circuit, as it is when the current flows through the shunt. This assumption is probably absolutely safe in all cases in practice. As this method is not in any way concerned with the nature of the circuit beyond the single pipe length under test, nor with the rest of the path of the current, it can be applied to the most complex network of pipes, even when the pipes are interconnected elsewhere or bonded to the track. In measuring the pipe resistance by this method, or in general when the voltage drop is measured, it is of course assumed that the current is constant while the two successive readings are taken, hence it is recommended to repeat the two readings a number of times.

TABLE FOR DETERMINATION OF CURRENT FLOW ON PIPING FROM MILLIVOLT DROP ALONG CONTINUOUS LENGTH OF PIPE BETWEEN JOINTS

L = Distance between contacts in feet
E = Instrument reading in millivolts
K = Constant from table
$\frac{KE}{L}$ = Current flow in amperes.

STANDARD CAST IRON PIPE
(Based on Resistance of 0.00144 ohm per lb.-ft.)

Classification					Actual dimensions		Weight per ft. exclusive of hub—lb.	K-current for one millivolt drop per foot of continuous pipe—amperes
Nominal diameter, inches	*Association standard	Class letter †	Head, feet	Pressure, lb. per sq. in.	Outside diameter, inches	Inside diameter, inches		
4	N	A	....	....	4.80	4.12	14.9	10.3
4	N	C	....	....	4.80	4.08	15.7	10.9
4	N	E	....	....	4.80	4.02	16.9	11.7
4	G	....	....	....	4.80	4.00	17.2	12.0
4	W	A	100	43	4.80	3.96	18.0	12.5
4	N	G	....	....	5.00	4.16	18.9	13.1
4	N	I	....	....	5.00	4.10	20.0	13.9
4	W	B	200	86	5.00	4.10	20.0	13.9
4	N	K	....	....	5.00	4.04	21.3	14.8
4	W	C	300	130	5.00	4.04	21.3	14.8
4	W	D	400	173	5.00	3.96	22.8	15.8
6	N	A	....	....	6.90	6.14	24.3	16.9
6	N	C	....	....	6.90	6.06	26.7	18.5
6	G	....	....	....	6.90	6.04	27.2	18.9
6	W	A	100	43	6.90	6.02	27.8	19.3
6	N	E	....	....	6.90	5.98	29.1	20.2
6	W	B	200	86	7.10	6.14	31.1	21.6
6	N	G	....	....	7.10	6.10	32.4	22.5
6	W	C	300	130	7.10	6.08	32.9	22.8
6	N	I	....	....	7.10	6.02	34.8	24.2
6	W	D	400	173	7.10	6.00	35.3	24.5
6	W	E	500	217	7.22	6.06	37.7	26.2
6	W	F	600	260	7.22	6.00	39.6	27.4
6	W	G	700	304	7.38	6.08	42.8	29.7
6	W	H	800	347	7.38	6.00	45.2	31.4
8	N	A	....	....	9.05	8.21	35.5	24.7
8	G	....	....	....	9.05	8.15	37.9	26.3
8	W	A	100	43	9.05	8.13	38.7	26.9
8	N	C	....	....	9.05	8.09	40.3	28.0
8	W	B	200	86	9.05	8.03	42.7	29.6
8	N	E	....	....	9.05	7.99	44.3	30.7

* W = American Water Works Association Standard
N = New England Water Works Association Standard
G = American Gas Institute Standard.
† As used by the American Water Works Association and the New England Water Works Association.

STANDARD CAST IRON PIPE—(*Continued*)
(Based on Resistance of 0 00144 ohm per lb -ft)

Classification					Actual dimensions		Weight per ft exclusive of hub—lb	K-current for one millivolt drop per foot of continuous pipe—amperes
Nominal diameter, inches	*Association standard	Class letter †	Head, feet	Pressure, lb per sq in.	Outside diameter, inches	Inside diameter, inches		
8	W	C	300	130	9 30	8 18	47 9	33 3
8	N	G			9 30	8 14	49 6	34 5
8	W	D	400	173	9 30	8 10	51 2	35 5
8	N	I			9 30	8 04	53 6	37 2
8	W	E	500	217	9 42	8 10	56 7	39 4
8	W	F	600	260	9 42	8 00	60 6	42 1
8	W	G	700	304	9 60	8 10	65 0	45 1
8	W	H	800	347	9 60	8 00	69 0	48 0
10	N	A			11 10	10 16	49 0	34 0
10	G				11 10	10 12	51 0	35 4
10	N	B			11 10	10 10	51 9	36 1
10	W	A	100	43	11 10	10 10	51 9	36 1
10	N	C			11 10	10 04	54 9	38 1
10	N	D			11 10	9 98	57 9	40 2
10	W	B	200	86	11 10	9 96	58 9	40 9
10	N	E			11 40	10 20	63 6	44 1
10	W	C	300	130	11 40	10 16	65 5	45 5
10	N	F			11 40	10 14	66 5	46 2
10	N	G		....	11 40	10 06	70 5	49 0
10	W	D	400	173	11 40	10 04	71 5	49 7
10	N	H			11 40	10 00	73 5	51 1
10	W	E	500	217	11 60	10 12	78 7	54 6
10	W	F	600	260	11 60	10 00	84 6	58 8
10	W	G	700	304	11 84	10 12	92 4	64 1
10	W	H	800	347	11 84	10 00	98 5	68 4
12	N	A			13 20	12 22	61 1	42 5
12	N	B			13 20	12 14	65 9	45 7
12	G	....			13 20	12 12	67 0	46 5
12	W	A	100	43	13 20	12 12	67 0	46 5
12	N	C			13 20	12 06	70 6	49 0
12	N	D			13 20	11 98	75 3	52 3
12	W	B	200	86	13 20	11 96	76 4	53 0
12	N	E			13 50	12 20	81 9	56 8
12	W	C	300	130	13 50	12 14	85 5	59 4
12	N	F			13 50	12 12	86 6	60 2
12	N	G			13 50	12 04	91 5	63 6
12	W	D	400	173	13 50	12 00	93 8	65 1
12	N	H			13 50	11 96	96 2	66 8
12	W	E	500	217	13 78	12 14	104 0	72 3
12	W	F	600	260	13 78	12 00	112 0	77 9

See footnotes, page 677.

STANDARD CAST IRON PIPE—(*Continued*)

(Based on Resistance of 0 00144 ohm per lb.-ft)

Classification					Actual dimensions		Weight per ft exclusive of hub—lb.	K-current for one millivolt drop per foot of continuous pipe—amperes
Nominal diameter, inches	*Association standard	Class letter †	Head, feet	Pressure, lb per *sq. in.*	Outside diameter, inches	Inside diameter, inches		
12	W	G	700	304	14 08	12 14	125	86 7
12	W	H	800	347	14 08	12 00	133	92 4
14	N	A			15 30	14 24	76 8	53 4
14	N	B			15 30	14 16	82 3	57 1
14	W	A	100	43	15 30	14 16	82 3	57 1
14	N	C			15 30	14 08	87 9	61 0
14	N	D			15 30	13 98	94 8	65 8
14	W	B	200	86	15 30	13 98	94 8	65 8
14	N	E			15 65	14 25	103	71 4
14	W	C	300	130	15 65	14 17	108	75 0
14	N	F			15 65	14 15	109	76 2
14	N	G			15 65	14 07	115	80 0
14	W	D	400	173	15 65	14 01	119	82 8
14	N	H			15 65	13 99	121	83 9
14	W	E	500	217	15 98	14 18	133	92 4
14	W	F	600	260	15 98	14 00	145	101
14	W	G	700	304	16 32	14 18	160	111
14	W	H	800	347	16 32	14 00	172	120.
16	N	A			17 40	16 30	90 9	63 1
16	N	B			17 40	16 20	98 9	68 6
16	W	A	100	43	17 40	16 20	98 9	68 6
16	G				17 40	16 16	102.	70 7
16	N	C			17 40	16 10	107	74 1
16	N	D			17 40	16 00	115	79 6
16	W	B	200	86	17 40	16 00	115	79 6
16	N	E			17 80	16 30	125	87 1
16	N	F			17 80	16 20	133	92 6
16	W	C	300	130	17 80	16 20	133	92 6
16	N	G			17 80	16 10	141	98 2
16	W	D	400	173	17 80	16 02	147	102 3
16	N	H			17 80	16 00	149	103 5
16	W	E	500	217	18 16	16 20	165	114 5
16	W	F	600	260	18 16	16 00	181	125 5
16	W	G	700	304	18 54	16 18	201	139 5
16	W	H	800	347	18 54	16 00	215	149 0
18	N	A			19 25	18 11	104	72 5
18	N	B			19 25	17 99	115	79 8
18	W	A	100	43	19 50	18 22	118	82 2
18	N	C			19 50	18 12	127	88 5
18	N	D			19 50	18 00	138	95 8
18	W	B	200	86	19 50	18 00	138	95 8

See footnotes, page 677.

STANDARD CAST IRON PIPE—(*Continued*)

(Based on Resistance of 0.00144 ohm per lb.-ft.)

Classification					Actual dimensions		Weight per ft. exclusive of hub—lb.	K-current for one millivolt drop per foot of continuous pipe—amperes
Nominal diameter, inches	*Association standard	Class letter †	Head, feet	Pressure, lb. per sq. in.	Outside diameter, inches	Inside diameter, inches		
18	N	E	......	....	19.70	18.10	148.	103.
18	N	F	......	....	19.70	17.98	159.	110.4
18	W	C	300	130	19.92	18.18	162.	113.
18	W	D	400	173	19.92	18.00	178.	123.8
18	W	E	500	217	20.34	18.20	202.	140.5
18	W	F	600	260	20.34	18.00	220.	152.6
18	W	G	700	304	20.78	18.22	245.	170.
18	W	H	800	347	20.78	18.00	264.	183.3
20	N	A	......	....	21.30	20.10	122.	84.6
20	N	B	......	....	21.30	19.98	134.	93
20	W	A	100	43	21.60	20.26	137.	95.4
20	G	......	......	....	21.60	20.24	140.	97.0
20	N	C	......	....	21.60	20.16	147.	102.5
20	N	D	......	....	21.60	20.02	161.	112.
20	W	B	200	86	21.60	20.00	163.	113.
20	N	E	......	....	21.90	20.20	175.	122.
20	N	F	......	....	21.90	20.06	189.	131.
20	W	C	300	130	22.06	20.22	191.	132.
20	W	D	400	173	22.06	20.00	212.	148.
20	W	E	500	217	22.54	20.24	241.	167.
20	W	F	600	260	22.54	20.00	265.	184.
20	W	G	700	304	23.02	20.24	295.	205.
20	W	H	800	347	23.02	20.00	319.	221.
24	N	A	.....	....	25.40	24.12	156.	108.
24	N	B	......	....	25.40	23.96	174.	121.
24	G	......	......	....	25.80	24.28	187.	130.
24	W	A	100	43	25.80	24.28	187.	130.
24	N	C	......	....	25.80	24.20	196.	136.
24	N	D	......	....	25.80	24.04	215.	149.
24	W	B	200	86	25.80	24.02	217.	151.
24	N	E	......	....	26.10	24.20	234.	163.
24	N	F	......	....	26.10	24.04	253.	176.
24	W	C	300	130	26.32	24.24	258.	179.
24	W	D	400	173	26.32	24.00	286.	198.
24	W	E	500	217	26.90	24.28	328.	228.
24	W	F	600	260	26.90	24.00	362.	251.
30	N	A	......	....	31.60	30.18	215.	149.
30	N	B	......	....	31.60	29.98	245.	170.
30	G	......	......	....	31.74	30.04	257.	179.

See footnotes, page 677.

STANDARD CAST IRON PIPE—(*Continued*)

(Based on Resistance of 0.00144 ohm per lb.-ft.)

Classification					Actual dimensions		Weight per ft. exclusive of hub—lb.	K-current for one millivolt drop per foot of continuous pipe—amperes
Nominal diameter, inches	*Association standard	Class letter †	Head, feet	Pressure, lb. per sq. in.	Outside diameter, inches	Inside diameter, inches		
30	W	A	100	43	31.74	29.98	266.	185.
30	N	C	.....	....	32.00	30.18	277.	192.
30	N	D	.....	....	32.00	29.98	306.	213.
30	W	B	200	86	32.00	29.94	312.	217.
30	N	E	.....	....	32.40	30.20	337.	234.
30	N	F	.....	....	32.40	30.00	367.	255.
30	W	C	300	130	32.40	30.00	367.	255.
30	W	D	400	173	32.74	30.00	422.	292.
30	W	E	500	217	33.10	30.00	479.	333.
30	W	F	600	260	33.46	30.00	537.	373.
36	N	A	.....	....	37.80	36.22	287.	199.
36	N	B	.....	....	37.80	36.00	326.	226.
36	G	.....	.....	....	37.96	36.06	345.	239.
36	W	A	100	43	37.96	35.98	358.	248.
36	N	C	.....	....	38.30	36.26	373.	259.
36	N	D	.....	....	38.30	36.04	412.	286.
36	W	B	200	86	38.30	36.00	418.	290.
36	N	E	.....	....	38.70	36.20	459.	319.
36	W	C	300	130	38.70	35.98	497.	346.
36	N	F	.....	....	38.70	35.96	502.	349.
36	W	D	400	173	39.16	36.00	581.	404.
36	W	E	500	217	39.60	36.00	666.	463.
36	W	F	600	260	40.04	36.00	753.	523.
42	N	A	.....	....	44.00	42.26	368.	256.
42	N	B	.....	....	44.00	42.00	422.	293.
42	G	.....	.....	....	44.20	42.06	452.	314.
42	W	A	100	43	44.20	42.00	465.	323.
42	N	C	.....	....	44.50	42.24	480.	333.
42	N	D	.....	....	44.50	41.96	538.	374.
42	W	B	200	86	44.50	41.94	542.	376.
42	N	E	.....	....	45.10	42.30	600.	416.
42	N	F	.....	....	45.10	42.04	654.	454.
42	W	C	300	130	45.10	42.02	657.	456.
42	W	D	400	173	45.58	42.02	763.	530.
48	N	A	.....	....	50.20	48.30	459.	319.
48	N	B	.....	....	50.20	48.00	529.	367.
48	N	C	.....	....	50.80	48.30	608.	422.
48	G	.....	.....	....	50.50	47.98	608.	422.
48	W	A	100	43	50.50	47.98	608.	422.
48	N	D	.....	....	50.80	48.00	678.	471.

See footnotes, page 677.

STANDARD CAST IRON PIPE—(*Concluded*)

(Based on Resistance of 0.00144 ohm per lb.-ft.)

Classification					Actual dimensions		Weight per ft. exclusive of hub—lb.	K-current for one millivolt drop per foot of continuous pipe—amperes
Nominal diameter, inches	*Association standard	Class letter †	Head, feet	Pressure, lb. per sq. in.	Outside diameter, inches	Inside diameter, inches		
48	W	B	200	86	50.80	47.96	686.	477.
48	N	E	......	....	51.40	48.30	757.	526.
48	N	F	......	....	51.40	48.00	828.	575.
48	W	C	300	130	51.40	47.98	832.	578.
48	W	D	400	173	51.98	48.06	961.	667.
54	N	A	......	....	56.40	54.34	559.	388.
54	N	B	......	....	56.40	54.00	650.	452.
54	W	A	100	43	56.66	53.96	731.	508.
54	N	C	......	....	57.10	54.36	750.	521.
54	N	D	......	....	57.10	54.02	840.	583.
54	W	B	200	86	57.10	54.00	845.	586.
54	N	E	......	....	57.80	54.26	946.	657.
54	N	F	......	....	57.80	54.00	1041.	723.
54	W	C	300	130	57.80	54.00	1041.	723.
54	W	D	400	173	58.40	53.94	1230.	854.
60	N	A	......	....	62.60	60.40	664.	460.
60	N	B	......	....	62.60	60.00	782.	543.
60	W	A	100	43	62.80	60.02	836.	581.
60	N	C	......	....	63.40	60.40	910.	632.
60	W	B	200	86	63.40	60.06	1010.	701.
60	N	D	......	....	63.40	60.00	1028.	714.
60	N	E	......	....	64.20	60.40	1160.	806.
60	W	C	300	130	64.20	60.20	1220.	848.
60	N	F	......	....	64.20	60.00	1280.	889.
60	W	D	400	173	64.82	60.06	1455.	1010.
72	W	A	100	43	75.34	72.08	1178.	819.
72	W	B	200	86	76.00	72.10	1415.	983.
72	W	C	300	130	76.88	72.10	1745.	1212.
84	W	A	100	43	87.54	84.10	1445.	1005.
84	W	B	200	86	88.54	84.10	1878.	1304.

See footnotes, page 677.

Standard Steel (or Wrought Iron) Pipe

(Based on resistance of steel—0.00021 ohm per lb.-ft. Based on resistance of wrought iron—0.000181 ohm per lb.-ft.)

Nominal diameter—inches	*Classification	Actual dimensions		Weight per foot. Plain ends—steel—lb.	K-current for one millivolt drop per foot of continuous pipe—amperes	
		Outside diameter—inches	Inside diameter—inches		Steel	Wrought iron
1/8	S	0.405	0.269	0.244	1.16	1.32
1/8	X	0.405	0.215	0.314	1.50	1.70
1/4	S	0.540	0.364	0.424	2.02	2.30
1/4	X	0.540	0.302	0.535	2.55	2.90
3/8	S	0.675	0.493	0.567	2.70	3.07
3/8	X	0.675	0.423	0.738	3.51	4.00
1/2	S	0.840	0.622	0.850	4.05	4.60
1/2	X	0.840	0.546	1.09	5.18	5.88
1/2	XX	0.840	0.252	1.71	8.16	9.28
3/4	S	1.050	0.824	1.13	5.38	6.11
3/4	X	1.050	0.742	1.47	7.03	7.98
3/4	XX	1.050	0.434	2.44	11.6	13.2
1	S	1.315	1.049	1.68	7.99	9.09
1	X	1.315	0.957	2.17	10.3	11.8
1	XX	1.315	0.599	3.66	17.4	19.8
1 1/4	S	1.660	1.380	2.27	10.8	12.3
1 1/4	X	1.660	1.278	3.00	14.3	16.2
1 1/4	XX	1.660	0.896	5.21	24.8	28.2
1 1/2	S	1.900	1.610	2.72	12.9	14.7
1 1/2	X	1.900	1.500	3.63	17.3	19.6
1 1/2	XX	1.900	1.100	6.41	30.5	34.7
2	S	2.375	2.067	3.65	17.4	19.8
2	X	2.375	1.939	5.02	23.9	27.2
2	XX	2.375	1.503	9.03	43.0	48.8
2 1/2	S	2.875	2.469	5.79	27.6	31.4
2 1/2	X	2.875	2.323	7.66	36.5	41.5
2 1/2	XX	2.875	1.771	13.69	65.2	74.2
3	S	3.500	3.068	7.57	36.0	41.0
3	X	3.500	2.900	10.2	48.8	55.6
3	XX	3.500	2.300	18.6	88.5	101.
3 1/2	S	4.000	3.548	9.11	43.4	49.3
3 1/2	X	4.000	3.364	12.5	59.6	67.8
3 1/2	XX	4.000	2.728	22.8	109.	124.
4	S	4.500	4.026	10.8	51.4	58.4
4	X	4.500	3.826	15.0	71.3	81.1
4	XX	4.500	3.152	27.5	131.	149.

*S = Standard pipe.
X = Extra strong pipe.
XX = Double extra strong pipe.

STANDARD STEEL (OR WROUGHT IRON) PIPE—(*Concluded*)

(Based on resistance of steel—0.00021 ohm per lb.-ft. Based on resistance of wrought iron—0.000181 ohm per lb.-ft.)

Nominal *diameter*—inches	*Classification	Actual dimensions		Weight per foot. Plain ends—steel—lb.	K-current for *one millivolt* drop per foot of continuous pipe—amperes	
		Outside diameter—inches	Inside diameter—inches		Steel	Wrought iron
4½	S	5.000	4.506	12.5	59.8	67.9
4½	X	5.000	4.290	17.6	83.9	95.3
4½	XX	5.000	3.580	32.5	155.	176.
5	S	5.563	5.047	14.6	69.7	79.2
5	X	5.563	4.813	20.8	98.9	112.
5	XX	5.563	4.063	38.5	183.	209.
6	S	6.625	6.065	19.0	90.3	103.
6	X	6.625	5.761	28.6	136.	155.
6	XX	6.625	4.897	53.2	253.	288.
7	S	7.625	7.023	23.5	112.	127.
7	X	7.625	6.625	38.0	181.	206.
7	XX	7.625	5.875	63.1	300.	342.
8	S	8.625	8.071	24.7	118.	134.
8	S	8.625	7.981	28.5	136.	155.
8	X	8.625	7.625	43.4	206.	235.
8	XX	8.625	6.875	72.4	345.	392.
9	S	9.625	8.941	33.9	161.	184.
9	X	9.625	8.625	48.7	232.	264.
10	S	10.750	10.192	31.2	149.	169.
10	S	10.750	10.136	34.2	163.	185.
10	S	10.750	10.020	40.5	192.	219.
10	X	10.750	9.750	54.7	261.	297.
11	S	11.750	11.000	45.6	217.	247.
11	X	11.750	10.750	60.1	286.	326.
12	S	12.750	12.090	43.8	208.	237.
12	S	12.750	12.000	49.6	236.	269.
12	X	12.750	11.750	65.4	311.	354.
....	S	14.000	13.250	54.6	260.	296.
....	X	14.000	13.000	72.1	343.	391.
....	S	15.000	14.250	58.6	279.	317.
....	X	15.000	14.000	77.4	369.	420.
....	S	16.000	15.250	62.6	298.	339.
....	X	16.000	15 000	82.8	394.	449.

*S = Standard pipe.
X = Extra strong pipe.
XX = Double extra strong pipe.

C. W. Kinney, *Electric Journal*, 1909, describes an approximate method of determining the amount of current flow in a pipe, the resistance of which was not known. The determination was made as follows: The water pipe was exposed for about 6 ft. of its length. Three places on the pipe, 2½ ft. apart, were filed bright and the drop in voltage, as indicated by the voltmeter, between the first and second points and then between the second and third points, was noted. After these readings had been taken, an ammeter reading was taken between the rail and the middle point on the water pipe. While the ammeter was still connected, voltmeter readings were taken as before. The voltage drop between points 1 and 2 showed less than one-half of 1 per cent of the original value. Thus, the voltage drop between points 2 and 3 could be assumed to be due entirely to the current flowing through the ammeter. The normal flow of current in the pipe was then calculated by a simple proportion; that is, the current flow during the first test bore the same relation to the ammeter reading that the voltmeter deflection between points 2 and 3 during the first test bore to the deflection of the voltmeter needle when the ammeter was connected, since the voltmeter was of uniform scale type. It was, therefore, not necessary to determine the actual value of the divisions on the voltmeter scale in order to find the amount of current flowing in the pipe.

Resistance of and Current in Lead and Lead Alloy Pipes and Sheaths. The resistance of "lead" pipe and "lead" cable sheath will depend upon the alloy that is used in its construction. Practical results indicate that the resistance of pure lead is about 480 microhms (0.00048 ohm) per pound-foot, and the resistance of "lead" cable sheathing containing 3 per cent tin is about 530 microhms (0.00053 ohms) per pound-foot. The curve sheet, Fig. 70, is arranged for the rapid approximation of resistance and current per millivolt drop per foot for lead and lead-tin alloy pipes. The following example will illustrate the method of using these curves:

What is the resistance per foot of sheath and the current per millivolt drop per foot along a cable sheath of a lead-tin alloy containing 3 per cent of tin, the resistance of the sheath being 0.00053 ohms per pound-foot? The outside diameter of the sheath is 2.28 in. and the thickness of the wall is 5/32 in.

Starting at the "2.28" point on the "Outside Diameter—Inches" scale (Fig. 70), follow the vertical to intersection with the oblique 0.156 (= decimal equivalent of 5/32) "Thickness—Inches" line. From this point follow the horizontal to the intersection with the "Weight per foot—Lead—3 per cent Tin-pound" scale at the 5.07 lb. point. Now starting with the 5.07 lb. point on the "Weight per foot, pure lead, pounds," follow the horizontal to the left to the intersection with the 0.0053 "ohm per pound-foot" oblique. From this point drop a perpendicular. The intersections of this perpendicular with the horizontal scale indicate that the resistance of the sheath is about 104 microhms per foot and the current is about 9.6 amperes per millivolt drop per foot along the sheath. NOTE: If the cable sheath or pipe is of pure lead, the horizontal

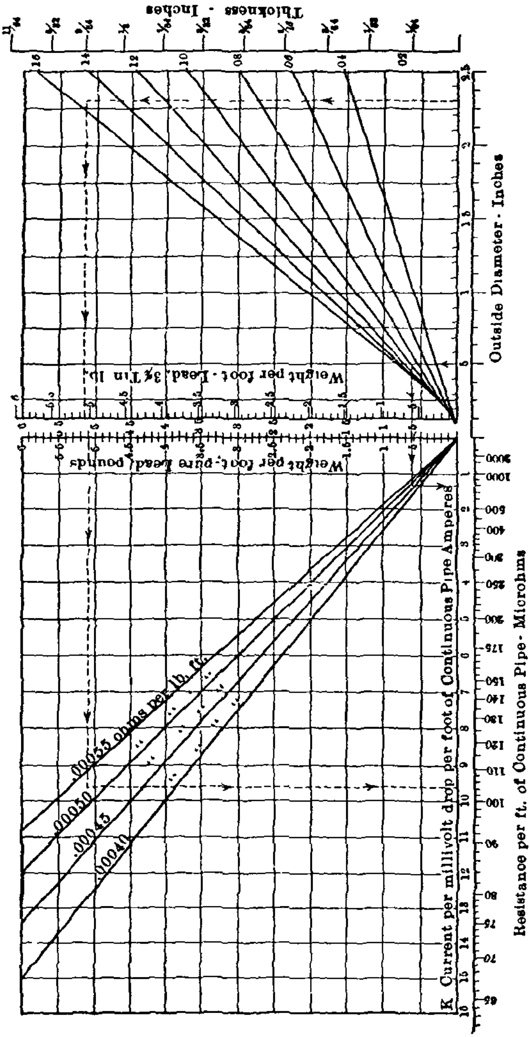


FIG. 70.—Current and resistance, lead and lead alloy cable sheath and pipe.

should be carried straight through, instead of making the correction on the vertical scale. Such a case is shown by the arrows starting at 0.5 in. outside diameter, Fig. 70.

Determination of Amount and Distribution of Current Leaving Underground Metallic Conductors. The generally accepted method consists in determining the amount of current flow (by one of the methods outlined above) in two locations on the same line of underground pipe. The difference in the amount of current on the pipe between these two locations will then be the amount of current leaving (or entering) the pipe between the two locations.

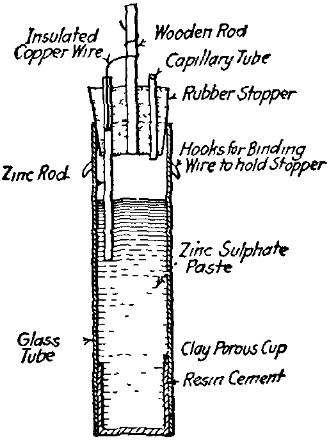


Fig. 71.—Haber normal electrode. Cross section.

Normal Electrode. The Haber normal electrode, also called non-polarizing electrode, consists of a rod of zinc which is enveloped in a wet paste of zinc sulphate contained in a glass tube which has had cemented to it at the bottom a porous clay cell. The other end of the tube is closed with a stopper from which the zinc rod is supported; an insulated wire is led from the end of the zinc rod through this stopper to the upper end of a wooden rod which also enters the stopper and serves for the purpose of handling the electrode. A capillary tube is also run through the stopper in order to have the interior of the tube at normal atmospheric pressure. The zinc sulphate paste is made by adding saturated zinc sulphate solution to fine zinc sulphate crystals until the mixture has attained a semi-fluid condition. A sketch showing the construction of this device is shown by Fig. 71.

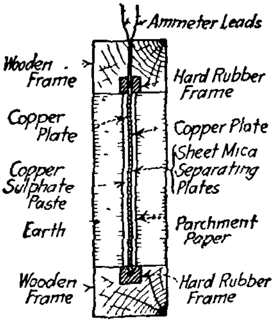


Fig. 72.—Haber earth current collector. Cross section.

Earth Current Collector. The Haber earth current collector consists of two thin copper sheets laid one upon the other with a thin sheet of mica or other non-absorbent insulating material between them. These two plates are gripped in a hard rubber rim which forms part of a square wooden frame. A paste made by mixing powdered copper sulphate crystals with a 20 per cent aqueous solution of sulphuric acid is spread over the exterior surfaces of each of the two sheets of copper, the paste being enclosed on each exterior surface by a covering of parchment paper or of some similar tough permeable membrane. Insulated wire leads of suitable length are run from each plate through the frame to connect with the measuring instrument. The opening in the frame may conveniently be square. Four inches is a convenient dimension for the sides of this square opening as this will yield an area of one-ninth of a square foot which is approximately equivalent to a square decimeter. The detailed construction is shown in Fig. 72. When using the instrument, the spaces between the parchment paper and the outer edges of the wooden frame are first filled with closely packed soil taken from the spot where it is intended to make the measurement, and the frame is then placed in a position perpendicular to the flow of current which it is desired to measure and completely buried in earth removed in the course of making the excavation to reach the structure whose condition is to be determined. A suitable low resistance milliammeter can then be connected to the two terminal wires and observations of the current flow made.

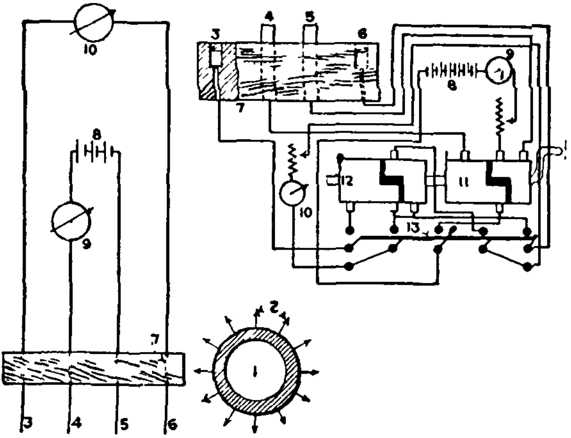


FIG. 73.—Principle and circuits of earth ammeter.

Earth Ammeter. This instrument, designed for the measurement of earth currents, was developed by the Bureau of Standards and described in the *Elec. Ry. Jour.*, 1921. Its purpose is to afford a means for the accurate determination of the polarity of pipes with respect to earth and for the quantitative measurement of current density at any desired point in the earth. If a measurement be made of the resistivity of the earth at any particular point, and if then a measurement be made of the voltage drop between two points a known distance apart, within the same region in which the resistivity has been measured, these two measurements will permit a calculation of the current density in the earth in the region immediately under investigation. The method described involves something of the principle here stated, although in its actual carrying out neither the resistivity of the earth nor the true potential drop between two points is determined. The principle of this method of measuring earth currents can best be understood by reference to Fig. 73, which is a diagrammatic illustration of the elements of the apparatus. Let us assume that the pipe is discharging current in all directions as indicated by the arrows 2. Four electrodes 3, 4, 5 and 6, may be imbedded in the earth immediately adjoining the pipe, on whatever side the current intensity is to be measured, or placed against the wall of an excavation made near the pipe. An excavation is here assumed tentatively to simplify the explanation of the principle of the method. It later will be shown how the method can be applied without making excavations of any kind. For convenience these several electrodes may be mounted on a single insulating frame 7. Two of these electrodes, for example 3 and 6, may be connected to a suitable voltage indicator 10, which need not read in any particular units. Suppose, now, a current I be caused to flow between the terminals 4 and 5 through the earth from the battery 8, which current will be measured by the ammeter 9. It will be evident that this current distributes itself in all directions through the earth and produces a certain voltage drop between the terminals 3 and 6 due to the resistance in the earth immediately surrounding the group of electrodes. This voltage drop between the terminals 3 and 6 will be indicated by the voltmeter 10 and will be proportional to the current flowing between the terminals 4 and 5 and to the resistivity of the surrounding earth. If E_0 is the voltage between the terminals 3 and 6 and if θ_0 is the corresponding deflection of the voltage indicator 10 we have

$$\theta_0 = KE_0$$

where K is the constant of the voltage indicator 10 which includes the effect due to the resistance of the leads and the electrodes 3 and 6. Further it will be seen that E_0 is proportional to the current I sent between the electrodes 4 and 5 and to the resistivity r of the surrounding earth, or

$$E_0 = AIr$$

where A is a constant depending upon the geometrical arrangement of the group of electrodes. Substituting this value of E_0 in the first equation, we have

$$\theta_0 = KAIr$$

Here it is assumed that the voltage drop across the terminals 3 and 6 is due solely to the current sent through the terminals 4 and 5. In order that this may be true, conditions must be such that no other current flowing through the earth at the time the measurement is made will in any way affect the apparatus. For the present, we will assume that this is actually the case; it will be explained later how it is realized in practice. After the above measurement of I and the corresponding θ_0, the circuit of the battery 8 is opened, after which the voltage drop E, between the voltage terminals 3 and 6 would be due solely to the current i which is flowing through the earth, or

$$E_1 = irL$$

where L is the distance between the terminals 3 and 6, i is the mean current density in the region between the terminals 3 and 6 and r, as above, is the resistivity of the earth. The corresponding deflection of the instrument 10 is θ_1 and we will have

$$\theta_1 = KE_1 = KirL$$

From these equations we have

$$\frac{\theta_0}{\theta_1} = \frac{KAIr}{KirL} = \frac{AI_0}{iL}$$

Solving for i, we have

$$i = \frac{AI\theta_1}{L\theta_0}$$

As stated above, A is a constant depending upon the geometrical form of the electrode group 3, 4, 5 and 6. This can be determined once for all for a given electrode group by immersing the electrode in a medium such as water through which a current density of known value is sent. Under these circumstances, if we perform the two measurements indicated above and substitute the values in the above equation, i being in this case known, we can once for all calculate the value of A, and as soon as the distance L between the two electrodes 3 and 6 is known, the proportional factor $\frac{A}{L}$ becomes known. Calling this factor R for brevity, we have

$$i = \frac{RI\theta_1}{\theta_0},$$

In this equation, i is the current per unit area, or the quantity which is to be measured, and R is the known constant. To obtain the value of i, we have therefore to perform the two operations mentioned above, namely, to send a known current I through the two electrodes 4 and 5 and at the same time measure the corresponding deflection θ_0 of the instrument 10, this being done in a manner described below, such that the instrument 10 will not be affected by any earth current other than that which flows from the battery 8 through the terminals 4 and 5. We then disconnect the battery 8 and measure the deflection θ_1 of the instrument 10 due solely to the earth current i. These three values, θ_0, I, and θ_1 are then substituted in the above equation and the value of i calculated.

As stated above, the indication of the voltage indicator 10 is a function of the resistance in series with its leads, and therefore of the resistance of the electrodes 3 and 6 and of the earth immediately surrounding them. In practice it is found that this resistance is often very high and quite variable, so that the instrument 10 does not in general give a true value of the voltage impressed in the earth between the two electrodes 3 and 6, and often not even an approximation to the true value. It will be observed, however, that the resistivity r of the earth in the region in which the test is being made and the constant K of the voltage indicator 10 disappear from the equation from which the earth current i is calculated. It will be seen, therefore, that in making this measurement, neither the resistivity of the earth, nor the true value of the voltage drop between the electrodes 3 and 6 need be known. This constitutes one of the important advantages of the method of procedure above described.

As stated, in carrying out the first of the two operations above described, it is essential that some arrangement be provided whereby the deflection θ_0 will be due only to the current I which flows through the terminals 4 and 5 and will not be influenced by any earth current already flowing. This can be accomplished in a very simple manner, by an arrangement shown in Fig. 73, which shows also a complete wiring diagram of the test set. In this arrangement, two commutators, 11 and 12, mounted on the same shaft, are employed. These commutators are so mounted on the shaft that commutation takes place on both at exactly the same instant, and are provided with a crank whereby they may be rotated by hand at a suitable speed. The commutator 11 is interposed between the battery 8 and the test terminals 4 and 5, while the commutator 12 is interposed between the terminals 3 and 6 and the voltage indicator 10. It will be seen that an alternating current flows through the earth from the terminals 4 and 5 and impresses an alternating voltage on the terminals 3 and 6 which are being commutated simultaneously with the current through the leads 4 and 5, which gives rise to a unidirectional voltage on the voltage indicator 10. This instrument being of the direct current type will therefore give a deflection θ_0 proportional to the current I sent through the terminals 4 and 5. At the same time, any unidirectional voltage impressed on the terminals 3 and 6 due to an earth current will be commutated so frequently that it will exercise no appreciable effect on the voltage indicator, and hence the reading of the latter will be just the same as if for the time being the earth current to be measured did not exist. After the measurement of the current I_0 and the deflection θ_0 is made under these conditions, a double-throw switch 13 is reversed, which, as will be seen from Fig. 73, disconnects the battery 8 from the terminals 4 and 5 and at the same time eliminates the commutator 12 from the circuit between the electrodes 3 and 6 and the voltage indicator 10. In the new position of the switch the voltage between the electrodes 3 and 6 due to the earth current i will produce a corresponding deflection in the voltage indicator 10 which is then read as the value θ_1. These three values θ_0, I and θ_1 are then substituted in the equation, and the value of the earth current i is calculated in any desired units, depending upon the value of the constant R used.

The electrode group 3, 4, 5 and 6, mounted on the insulating support 7, may be permanently buried in the earth in the region in which it is desired to measure the earth current at any time, or the four electrodes may be placed against the wall of an excavation, so that all four terminals make contact with the earth, while a measurement of current intensity in the earth adjoining the wall of the excavation is being made. The constant R of the instrument will, however, be different in the two cases, but can be determined once for all for the two types of measurements. In most cases, however, where it is desired to measure the current density discharged from a pipe at any given point, it is unnecessary to make an excavation. For measurements of this kind, a special type of four-terminal electrode has been designed which can be placed down in a hole extending from the surface of the earth to the pipe, as shown in Fig. 74. This hole may be made by means of an auger, or by simply driving a pipe or rod of suitable size into the earth, and then removing the rod from the hole. The four electrodes 3, 4, 5 and 6 are then put down in this hole and the measurement is made in exactly the same manner as described above. The electrodes 4 and 5 through which the test current is sent from the battery in the first part of the test can be made from any ordinary metal, such as iron and copper. The electrodes 3 and 6, however, should be made on the well-known principle of the nonpolarizable electrode, that is, they should comprise a cup having an electrode at the base of copper, the cup being filled with a concentrated solution of copper sulphate. This electrolyte is confined in the cup by a stopper of wood or other porous material. Fig. 74 shows a typical form.

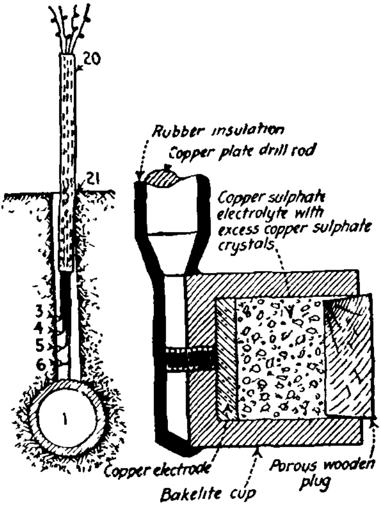


Fig. 74.—Electrodes for earth ammeter.

It is well to have clearly in mind just what current is measured by this apparatus. It is the mean current per unit area, as for instance per square centimeter or per square inch, in the earth in a small region immediately surrounding the center of the four-electrode group. A good approximation will be had by stating that it gives the mean current density throughout the volume of a sphere having a diameter approximately equal to the distance between the two potential electrodes of the four-electrode group. It will thus be seen that by using electrodes of very small size the current density in a very small volume of earth can be studied. By the use of electrodes an inch or an inch and a half apart, the mean current density in a sphere as small as three or four inches in diameter can be definitely determined. For most ordinary purposes, however, it is found desirable to use a two- or three-inch spacing of the electrodes, in which case we secure the mean current density in a volume of perhaps half a cubic foot of earth immediately surrounding the center of the electrode group.

The voltage indicator 10 used in this test set must be of very special design to have an extremely high current sensitivity. The instrument used gives a full scale deflection for one microampere, and was designed and built especially for this apparatus by the Rawson Electrical Instrument Company of Cambridge, Mass., which company is now manufacturing the complete sets. The instrument has been in use for some time at the Bureau of Standards, where it has been subjected to careful tests and experimental work, and has been found to be a very convenient, economical and accurate means of measuring the current intensity discharged from buried pipes.

Leakage Resistance between Railway Tracks and Earth. The determination of the average resistance of the leakage path between railway tracks and surrounding earth is often very desirable, particularly where it is necessary to determine what over-all potential drops may safely be permitted in the track return. It will be evident that if the resistance of the leakage paths is very high it will be safe to allow higher potential drops on the track than if the leakage resistance be low, although the voltage drop which may be considered safe is not directly proportional to the average resistance of the leakage path. The Bureau of Standards has made a considerable number of tests on various types of roadbeds, during the course of which two methods have been found satisfactory. One of these methods consists of inserting insulating joints in the track at two points 1000 ft. or more apart and bonding around these with a heavy bond designed to be conveniently opened at any time for testing. The test is made at night when no traffic is on the line, the shunt around the insulating joint being opened; a low-voltage battery then is applied between the isolated section of track and a suitable earthed terminal giving a very low resistance to ground. For this purpose the railway track on either side of the isolated section can be used. These tracks have substantially the potential of a point on the earth quite remote from the track section under test. When this connection is made the current flowing from the battery to the isolated section of track must practically all pass off through the track roadbed in this section, the leakage around the insulating joints being very small compared to the total leakage through the roadbed of the section under test; and this current is measured simultaneously with the potential difference between the track and a second earth terminal which should preferably be remote both from the isolated track section under test and from the earth terminal which is carrying the current of the battery. The resistance of the leakage path between the isolated track section and ground is then calculated from the ammeter and voltmeter readings.

The second method of testing which has given satisfactory results in some cases eliminates the necessity of inserting insulating joints in the track, a procedure which is often quite difficult, especially where cross bonds or space bars are frequently used. In this method two batteries are required and the arrangement is shown in Fig. 75. The batteries are stationed from one to several thousand feet apart and connected as in the preceding test, one terminal being connected to track and the other to an earthed terminal some distance away. It is desirable to connect the positive terminal of both batteries to the track and the negative terminal to earth, since this represents the polarity existing in practice where a current is leaking from the track into the earth. From an examination of the figure it will be seen that a great deal of current flowing from each battery will flow off in the directions away from the section

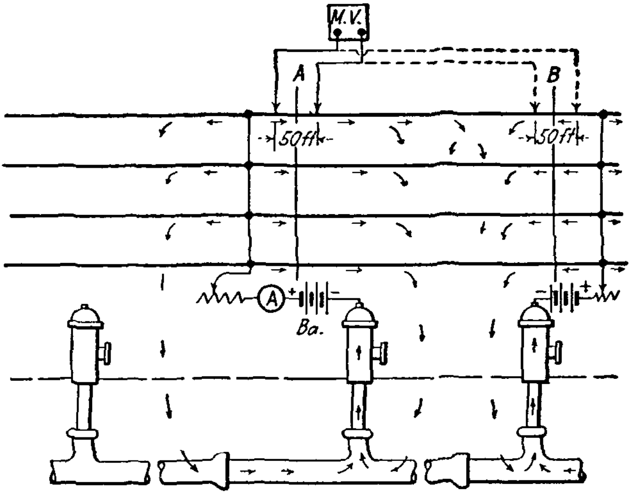


Fig. 75.—Diagram of connections between a double track railway line and adjoining pipe system, for measuring leakage resistance of a section of roadbed.

under test, as indicated by the arrows, but a certain amount, corresponding to the total leakage of the current on the section between the batteries, will flow into this section. If now we measure the millivolt drop on short lengths of the rails at the points *A* and *B*, just inside the points at which the batteries are connected to the tracks, we can calculate from this millivolt drop on a measured length of rail of known weight the approximate current which is flowing into the section under test from each end. The sum of these two currents will then be the total leakage current from the test section. At the same time a voltmeter is used to measure the potential difference between the section under test and a remote point in the ground; and from this voltmeter reading and the total leakage current the resistance of the section can be calculated.

Determination of Drop of Potential on Rails. As the difference of potential between two points on the rails of an electric railway system is in a way a measure of the tendency to cause stray earth currents, and as many municipal and other public requirements are based on this figure, this test is an important one, but not difficult to make, the essential matter being that of getting a potential wire between the two points on the rail and then measuring the drop with an ordinary voltmeter. Where the company has telephone or signal circuits which may be disconnected temporarily these are often used for this purpose. In some cases a trolley feeder is used, but a spare feeder is not often available during the time when such tests are to be made. Often it is possible to arrange with the local telephone company for its cooperation in furnishing spare wires from a terminal box in one location through proper connections at the telephone exchange to a terminal box in another location, and from these terminal boxes a continuation of the potential wire may be run to the voltmeter and rail. In using such circuits through a telephone exchange, it is important that all connections be removed on the terminal board which might serve to impress telephone battery current on the particular circuit in use. If the instrument used be of high resistance and if the telephone or signal wires used as potential wires be of copper, it is rarely necessary to allow for the resistance of the latter in correcting the voltmeter readings.

Limiting Rail Drop Should Be Average Rather Than Maximum. In many municipal or other public requirements a limit has been placed on the potential drop in rails, and in many cases this limit has been expressed in terms of a maximum permissible drop. That this is unfair and that the limit should be expressed as an average drop is brought out by Messrs. McCollum and Logan in their paper on electrolytic corrosion (Trans. A.I.E.E., 1913) as follows:

"It is evident that if the total amount of damage which results is proportional to the average current, then the limitation of the average voltage is more logical than the limitation of the peak load voltage, since in the former case the cost of meeting the voltage limitation in any given case is proportionate to the danger involved irrespective of the station load factor; whereas if the voltage at peak load is the determining factor, the cost of complying with the requirement depends not only on the danger involved, but on the load factor of the system, and the poorer the load factor, the greater its cost will be. The rate of damage does not increase as fast as the voltage increases, because of the tendency toward lower corrosion efficiencies at higher current densities. This indicates that, with a given average all-day current, the actual amount of electrolysis that would occur would be less with a bad load factor than with a good load factor, and hence points to the undesirability of penalizing a high peak of short duration. It would appear very much more logical, therefore, in so far as the damage itself is concerned, to make the average all-day voltage the basis of the limitation rather than the voltage at time of peak load."

Methods of Reducing Earth Potentials and Currents, thus Mitigating Electrolytic Corrosion

The methods which have been most often used with success in this country to reduce earth potentials and currents have been three; the insulated negative return feeder system, the insulating joint system, and the drainage system.

Insulated Negative Return Feeder System. In this system the tracks are drained of current at radially disposed points about the power station by insulated negative return feeder cables. The negative bus bar is not connected to the tracks nor to ground in the vicinity of the power station except through a resistance approximating that of the negative feeders. Pressures as nearly equal as are required may be maintained at the track end of the return feeders by the use of rheostats on short feeders and boosters on long ones, or by greatly increasing the amount of copper in long feeders without the use of boosters or rheostats. The cost of the system increases very rapidly as the permissible maximum variation between the various portions of the track decreases. In a few American cities insulated negative return feeders have been installed in a portion of the city so as to secure an equipotential condition of the track in an area where the amount of underground cables or pipes subject to damage is the greatest. This method is feasible and most often used where the railway company's power station or substation is located very close to the center of distribution of a system composed of several or many lines radially disposed about the center of distribution.

The Insulating Joint System requires that cable sheaths or piping systems should have insulating joints at frequent intervals. If the insulating joints be sufficiently numerous, this system may give good results, but there is some difficulty in applying them to an extensive system of underground cables, especially where the number of manholes is large and manholes crowded. Unless the number of insulating joints is sufficiently large to break up the potential around such joints to a very low value around each, there is also danger of current leaving the pipe or cable sheath on one side of the insulating joint and entering it on the other to such an extent that more or less serious corrosion is caused on the positive side of the joint. If every joint in the piping system were an insulating joint, this method would probably be an ideal one, as it would so increase the total resistance of the pipe line that the current flowing along it would be practically *nil*. This is illustrated in the case of some cast iron gas mains where cement joints of high resistance are used and on which mains practically no current can be found.

Pipe Drainage System. In this system the pipe or cable sheaths in the positive district are connected to the negative side of the railway return circuits through low resistance cables. This system, as its name implies, drains the stray currents from the pipes or cable sheaths through metallic conductors so that practically no current leaves the pipe by way of the earth path. The drainage cables preferably should be connected directly to the negative bus bar at the power station or substation, and should be of very low resistance. In many cases a meter is installed in the connection to the negative bus, and in cases where the station shuts down for a portion of the day there should be a switch in the bus connections and the circuit should be opened when the station is not in operation. If the drainage system is properly installed, it will maintain the piping system and cable sheaths throughout their length at a potential lower than that of the rails. An objection which has been raised to the drainage system is that by reducing the total resistance to the flow of stray currents it thereby increases their volume and thus increases the damage which may be expected from "joint electrolysis," or the electrolytic action which may take place around a high resistance pipe joint due to the stray currents leaving the pipe on the positive side of the joint and returning on the negative side. Some evidence is on record of such damage, but the cases where such damage has occurred are extremely rare and where noted it has been caused either by isolated exceptionally high resistance joints or excessive current flow on the piping mains. Many cases are on record where a current density of 5 to 10 amp. per pound-foot or more has been carried on cast iron water mains for years with no evidence of joint electrolysis. The drainage system has been used by the Bell telephone companies throughout the country for the protection of their cables for many years. In some cases city authorities have compelled the installation of the drainage system and in many cases they have allowed its use where the water system is owned by the city. The drainage system is very simple and much cheaper to install than the insulating joint method on a system of cables or pipes which are already in place in the ground, and is the system which has been in most common use in American cities either alone or in connection with the insulated negative return feeder system.

Three Wire Distribution. This system, using parts of the trolley wire as positive and parts as negative, with the rails as the neutral member, greatly reduces the current in the rails, as well as the rail potentials, and thus minimizes stray earth currents. According to W. Nelson Smith, the first instance of this of which there is any record was at Portland, Oregon, about the year 1891, when several small trolley systems in that city were unified under one management and supplied from one source of power, on the three wire system. This was done at that time partly to keep two separate trolley lines entirely separate from each other as regards their power supply, and partly to get some saving in copper feeders, although just how this latter result was to be secured is not clear from the information available; but it was noticed then that the districts operating on the three wire system were immune from damage by electrolysis of water pipes, while those traversed by other lines operating on the two wire system were frequently annoyed by this trouble. The next demonstration of this kind was in Brisbane, Australia, about ten to twelve years ago, where the manager of the system, who was a former Edison engineer from the United States, changed it to sectional three wire operation for the express purpose of preventing electrolysis damage, in which he was entirely successful from the start. In 1915 the Pacific Electric Railway stopped electrolysis damage to water pipes in their interurban territory surrounding Los Angeles by changing to three wire operation, subsequently enlarging their first installation to cover more of their territory. In Omaha, Nebraska, the trolley system was changed to three wire operation in 1916, and the installation enlarged in 1917. It is worthy of note that in Omaha the gas mains were injured by stray current only on streets where there were railway tracks. The soil there is of clay and of very low resistance, but it does not carry any amount of alkaline salts. In 1917 one of the substation districts in Milwaukee was changed to three wire operation with very successful results in reducing track potentials. A survey made in 1919, under the direction of the Bureau of Standards, resulted in the recommendation that the three wire system be ultimately extended to other districts. In 1918 the three-wire system was adopted at Wilmington, Delaware. One of the special problems there was the protection of a very important long-distance underground telephone cable system in the outskirts of the territory, a negative booster with a rather long feeder being applied for this purpose. In 1919 the three wire system was applied at Winnipeg, first in the Fort Rouge district, then in the St. Boniface, and successively thereafter in the Mill Street, North End, Sherbrooke and St. James districts, all of which had more than one machine in each substation. The Logan Avenue district, having only a single machine in the substation, was continued on the two wire system and the installation of insulated track feeders in that district was extended and improved.

Surface Insulation of Underground Metallic Structures. The U. S. Bureau of Standards, after an extended investigation, presents the following conclusions: "Such pipe paints, dips, and wrappings as have been brought to our attention are, with practically no exception, of no value whatever for protecting pipes from electrolysis when applied in the positive areas near the power houses. If, however, they are applied in negative areas they may be of considerable temporary value in reducing the current picked up by the pipe, and in that way indirectly they may reduce damage in positive areas. We wish to emphasize the fact that the results of these tests are not to be considered as throwing light on the value of these coatings for protecting various metals from natural soil corrosion, as the tests were designed solely for the purpose of testing their value as a means for protecting against electrolysis from stray currents, where the forces tending to corrode the pipes are of much greater magnitude than those producing galvanic action, which is largely responsible for slow corrosion of iron in soil. Whatever use may be made of such coatings in the negative areas for reducing the amount of current flow in the pipes it should always be looked upon as a secondary means of mitigation only and not depended upon as a chief means of protecting pipes."

Electrolysis from Alternating Current. J. L. R. Hayden (Trans. A.I.E.E., 1907) draws the following conclusions relative to alternating-current electrolysis: It is not a phenomenon like direct-current electrolysis, on which definite quantitative general laws can be formulated, but is of the character of a secondary effect; that is, the action of the positive half wave is not quite reversed by the action of the negative half wave, leaving a small difference, rarely exceeding one-half of 1 per cent of the electrolytic action of an equal direct current. Alternating current electrolysis varies from practically nothing to somewhat less than 1 per cent of direct current electrolysis, varying with the chemical nature of the electrolyte and being practically independent of the current density. He states that protection from alternating current electrolysis may be absolutely obtained by the superimposition of a very small quantity of direct current upon the alternating, the amount of direct current being only 1.5 per cent of the alternating current.

Transmission Lines

Voltage Determination. The determination of the proper voltage for the transmission of energy requires a cost study made up of two parts, namely, the cost of the energy lost and the cost of the line, the transmitting and receiving apparatus. The amount and thus the cost of power lost in a given transmission line in delivering a given amount of power at the receiver end of that line decreases if the voltage at which that power is transmitted be increased. Such an increase in voltage necessitates an increase in the cost of insulation, apparatus and protective devices. Thus, under a given set of market conditions, the voltage at which a given amount of power will be delivered to a distant point at the least cost is limited to that value at which the sum of the cost of losses resulting and cost of line, transmitting and receiving apparatus necessary is a minimum. Kapp's modification of Kelvin's law of economy is: "The most economical area cf conductor is that for which the annual cost of energy wasted is equal to the annual interest on that portion of the capital outlay which can be considered proportional to the weight of metal used." As the distance at which energy is transmitted increases, the cost of transmission line conductor and the losses in the line become of increasing importance relati e to the cost of transmitting and receiving apparatus and the continuance of a given economy demands an increase in the voltage of operation. The amount of conductor required for a transmission line to transmit a given amount of power with a given loss on the line is reduced 75 per cent each time the voltage at the receiver is doubled.

The more common transmission voltages are 2200, 4400, 6600, 11,000, 13,200, 22,000, 33,000, 44,000, 66,000, 88,000, 110,000, 140,000.

Spacing of Transmission Line Conductors. Transmission line conductors must be spaced far enough apart so that there will be no danger of arcing nor sparking between them and no danger of appreciable loss due to creepage or corona. The spacing must be given particular attention with regard to the possibility of the conductors swinging dangerously near together in the wind, especially where the direction of the wind is not at right angles to that of the transmission line.

Power Loss and Regulation of Transmission Line. The values of the power loss and regulation depend upon the amount of power delivered, the voltage at which it is delivered at the receiver end, the power factor of the load, the number of phases and the frequency of the system, the length of the line and the material, temperature, *size, arrangement and distance apart of the conductors. For an approximate calculation of the power loss and regulation accurate* enough for all ordinary electric railway work, it is sufficient to take into consideration the following items: Amount of power delivered, number of phases, voltage at receiver, power factor at receiver, resistance of the line and inductive reactance of the line. (Note *that capacitance of the line is here omitted.)*

Calculation of Power Loss and Regulation for Single Phase Two-wire Line. For three-phase see page 705.

Power Loss. (kilowatts line loss) $= \dfrac{I^2R}{1000}$

in which I = current in line, amperes

$$= \frac{P \times 1000}{E \cos\theta}$$

P = power delivered at receiver, kilowatts
E = electromotive force at receiver, volts
$\cos\theta$ = power factor of load = power factor of receiver
R = resistance of line, ohms
= 2 × resistance of one conductor, ohms

Example: To find the power loss in line delivering 2000 kw. at the receiver over a single-phase two-wire transmission line 60,000 ft. long composed of No. 1 A.W.G. (B. & S.) hard drawn solid copper wire; spacing of conductors, 36 in.; frequency, 60 cycles per second; electromotive force at receiver, 22,000 volts; power factor at receiver, 0.90 (current lagging).

Solution:

$$(\text{Current in line}) = \frac{2000 \times 1000}{22{,}000 \times 0.90}$$

$$= 101 \text{ amperes}$$

The resistance per 1000 ft. of No. 1 A.W.G. hard-drawn solid copper conductor is 0.1272 ohms; therefore:

$$(\text{resistance of line}) = R = 2 \times 60 \times 0.1272 = 15.26 \text{ ohms}$$

$$(\text{kilowatts loss in line}) = \frac{101^2 \times 15.26}{1000} = 155.7$$

Regulation.

$$(\text{per cent regulation}) = 100 \times \frac{E_G - E}{E} \text{ (see Fig. 76).}$$

in which E_G = electromotive force at generator end of line, volts

$$= \sqrt{(E\cos\theta + IR)^2 + (E\sqrt{1 - (\cos\theta)^2} + IX)^2}$$

I = current in line, amperes

$$= \frac{P \times 1000}{E \cos\theta}$$

P = power delivered at receiver, kilowatts
E = electromotive force at receiver, volts
$\cos\theta$ = power factor of receiver (load)
R = resistance of line, ohms
= $2 \times$ resistance of one conductor, ohms
X = inductive reactance of line, ohms
= $2 \times$ inductive reactance of one wire, ohms

Following are three methods of calculating regulation: I. Solution by means of above analytical method. II. Solution by the use of the "Mershon Diagram," (Fig. 77) introduced by Ralph D. Mershon, *American Electrician*, 1897. III. Solution by a "drop-factor" method based on that of Charles F. Scott and Clarence P. Fowler, *Electric Journal*, 1907.

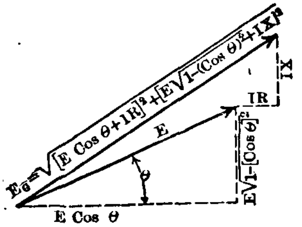


FIG. 76.—E. M. F., diagram for single phase line, capacitance neglected.

I. Direct Analytical Method of Calculating Regulation. Example: To find the per cent regulation of line delivering 2000 kw. at the receiver over a single-phase, two-wire transmission line 60,000 ft. long composed of No. 1 A.W.G. (B. & S.) hard-drawn solid copper wire; spacing of conductors, 36 in.; frequency, 60 cycles per second; electromotive force at receiver, 22,000 volts; power factors at receiver, 0.90 (current lagging).

Solution:

$$(\text{current in line}) = I = \frac{P \times 1000}{E \cos\theta} = \frac{2000 \times 1000}{22{,}000 \times 0.90} = 101 \text{ amperes.}$$

The resistance per 1000 ft. of No. 1 A.W.G. hard-drawn solid copper conductor is 0.1272 ohms; therefore

$$(\text{resistance of line}) = R = 2 \times 60 \times 0.1272 = 15.26 \text{ ohms.}$$

The inductive reactance per 1000 ft. of No. 1 A.W.G. solid conductor, spaced (interaxial distance) 36 in., is 0.13218 ohms; therefore

$$(\text{inductive reactance of line}) = X = 2 \times 60 \times 0.13218 = 15.86 \text{ ohms}$$

$$E_G = \sqrt{(E\cos\theta + IR)^2 + (E\sqrt{1-(\cos\theta)^2} + IX)^2}.$$

$$= \sqrt{(22{,}000 \times 0.90 + 101 \times 15.26)^2 + (22{,}000\sqrt{1-0.90^2} + 101 \times 15.86)^2}$$
$$= 24098$$

$$(\text{Per cent regulation}) = 100 \times \frac{E_G - E}{E}$$

$$= 100 \times \frac{24{,}098 - 22{,}000}{22{,}000}$$

$$= 9.5 \text{ per cent of receiver voltage.}$$

II. Method of Calculating Regulation by Use of Mershon Diagram. The method consists of first determining the resistance drop (called for convenience, "resistance volts") and the electromotive force required to balance the inductive reactance (called for convenience, "reactance volts"); these are then expressed in per cent of receiver voltage and applied to the Mershon diagram (Fig. 77) from which the per cent regulation is then read directly.

Example: To find the per cent regulation of a line delivering 2000 kw. at the receiver over a single-phase, two-wire transmission line 60,000 ft. long composed of No. 1 A.W G. (B. & S) hard-drawn solid copper wire; spacing of conductors, 36 in.; frequency, 60 cycles per second; electromotive force at receiver, 22,000 volts; power factor at receiver, 0.90 (current lagging).

Solution:

$$(\text{current in line}) = I = \frac{P \times 1000}{E \cos \theta}$$

$$= \frac{2000 \times 1000}{22{,}000 \times 0.90}$$

$$= 101 \text{ amperes}$$

The resistance per 1000 ft. of No. 1 A.W.G. hard-drawn solid copper conductor is 0.1272 ohms; therefore

$$(\text{resistance of line}) = R = 2 \times 60 \times 0.1272$$

$$= 15.26 \text{ ohms}$$

$$(\text{"resistance volts"}) = IR = 101 \times 15.26$$

$$= 1541 \text{ volts}$$

$$(\text{per cent of receiver voltage}) = \frac{1541 \times 100}{22{,}000} = 7.0$$

The inductive reactance per 1000 ft. of No. 1 A.W.G. solid conductor, spaced (interaxial distance) 36 in., is 0 13218 ohms; therefore

$$(\text{inductive reactance of line}) = X = 2 \times 60 \times 0.13218$$

$$= 15.86 \text{ ohms}$$

$$(\text{"reactance volts"}) = IX = 101 \times 15.86$$

$$= 1602 \text{ volts}$$

$$(\text{per cent of receiver voltage}) = \frac{1602 \times 100}{22{,}000} = 7.3$$

Now on the Mershon diagram (Fig. 77) from the point at which the vertical line at load power factor 0.90 cuts the arc 0, lay off the per cent resistance volts (7) horizontally to the right. At the end of the resistance volts line thus laid off, lay off the per cent reactance volts (7.3) vertically upward. The end of this line intersects the arc 9.5, which is the per cent regulation.

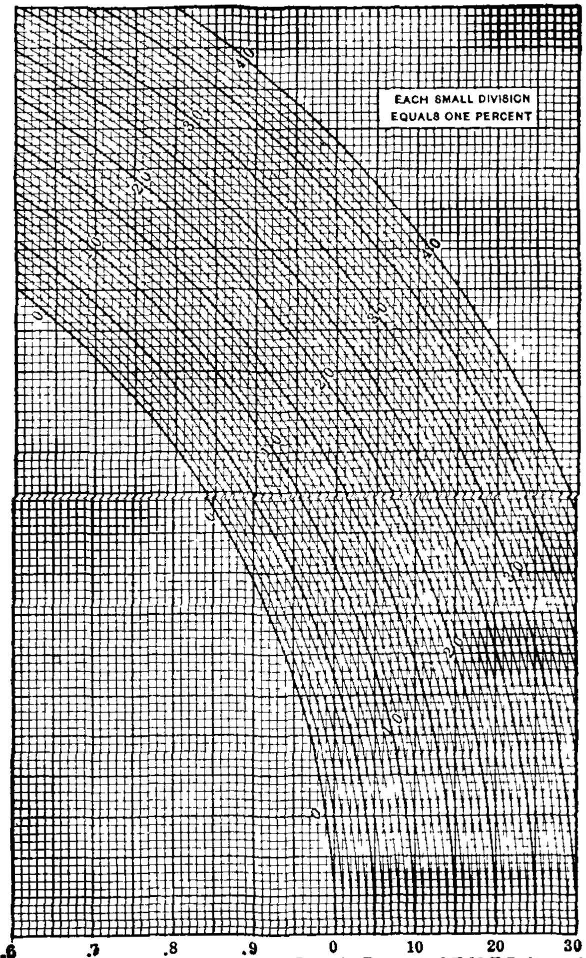


FIG. 77.—Mershon diagram.

III. "Drop Factor" Method of Calculating Regulation. In this method the regulation is found by multiplying the "resistance volts" by the "drop factor." This "drop factor" is the ratio which the difference between "generator volts" and "receiver volts" bears to the "resistance volts." It depends upon (1) the ratio of the "reactance volts" to the "resistance volts" (the ratio of the reactance of the line to the resistance of the line), (2) the power factor of the load, and (3) the ratio of the "resistance volts" to the "receiver volts." The "drop factor" is given on page 705 for "resistance volts" equal to 10 per cent of the "receiver volts." The ratio of the "resistance volts" to the "receiver volts" has a comparatively small effect on the drop factor, consequently this table may be used, with small error resulting, in cases where the "resistance volts" do not exceed 15 or 20 per cent of the "receiver volts."

Example: To find the per cent regulation of line delivering 2000 kw. at the receiver over a single-phase, two-wire transmission line 60,000 ft. long composed of No. 1 A.W.G. (B. & S.) hard-drawn solid copper wire; spacing of conductors, 36 in.; frequency, 60 cycles per second; electromotive force at receiver, 22,000 volts; power factor at receiver, 0.90 (current lagging).

Solution: The inductive reactance per 1000 ft. of No. 1 A.W.G. solid conductor, spaced (interaxial distance) 36 in., is 0.13218 ohm.

The resistance per 1000 ft. of No. 1 A.W.G. hard-drawn solid copper conductor is 0.1272 ohm.

Therefore
$$\frac{\text{reactance}}{\text{resistance}} = \frac{0.13218}{0.1272} = 1.04$$

$$(\text{resistance of line}) = R = 2 \times 60 \times 0.1272 = 15.26 \text{ ohms}$$

$$(\text{current in line}) = I = \frac{P \times 1000}{E \cos \theta} = \frac{2000 \times 1000}{22{,}000 \times 0.90} = 101 \text{ amperes}$$

$$(\text{"resistance volts"}) = IR = 101 \times 15.26 = 1541 \text{ volts}$$

From the table opposite, the "drop factor" for a load power factor of 90 per cent and the ratio of reactance to resistance of 1.04 is found by interpolation to be 1.39, therefore

$$(\text{regulation volts}) = (\text{"resistance volts"}) \times (\text{"drop factor"}) = 1541 \times 1.39 = 2142 \text{ volts}$$

$$(\text{per cent regulation}) = \frac{2142 \times 100}{22{,}000} = 9.7$$

DROP FACTORS WHEN RESISTANCE VOLTS ARE 10 PER CENT OF THE RECEIVER VOLTS

Ratio of reactance to resistance	Drop factors for power factors of (current lagging)								
	100%	95%	90%	85%	80%	70%	60%	40%	20%
0.10	1.00	1.00	1.00	0.94	0.88	0.80	0.70	0.60	0.30
0.20	1.00	1.01	1.01	0.98	0.92	0.86	0.82	0.67	0.40
0.30	1.00	1.05	1.05	1.02	0.99	0.93	0.89	0.74	0.50
0.40	1.00	1.08	1.10	1.08	1.04	1.00	0.93	0.82	0.60
0.50	1.00	1.11	1.14	1.13	1.10	1.07	1.01	0.92	0.70
0.60	1.01	1.15	1.18	1.19	1.15	1.14	1.09	1.01	0.80
0.70	1.02	1.18	1.23	1.24	1.21	1.20	1.17	1.11	0.91
0.80	1.02	1.21	1.28	1.29	1.28	1.27	1.24	1.20	1.01
0.90	1.03	1.25	1.33	1.34	1.34	1.35	1.32	1.29	1.11
1.00	1.04	1.28	1.37	1.39	1.40	1.41	1.39	1.38	1.20
1.10	1.05	1.32	1.41	1.44	1.45	1.48	1.47	1.46	1.30
1.20	1.06	1.35	1.46	1.50	1.51	1.55	1.54	1.55	1.40
1.30	1.07	1.39	1.51	1.55	1.57	1.62	1.63	1.64	1.49
1.40	1.08	1.43	1.55	1.61	1.64	1.70	1.71	1.72	1.59
1.50	1.10	1.47	1.60	1.67	1.70	1.77	1.80	1.81	1.70
1.60	1.10	1.51	1.65	1.74	1.77	1.85	1.87	1.90	1.80
1.70	1.13	1.55	1.70	1.79	1.84	1.92	1.95	1.99	1.90
1.80	1.15	1.59	1.76	1.85	1.91	1.99	2.04	2.08	1.99
1.90	1.17	1.63	1.82	1.91	1.98	2.06	2.11	2.16	2.08
2.00	1.18	1.68	1.87	1.96	2.04	2.14	2.19	2.25	2.18
2.10	1.20	1.72	1.92	2.03	2.10	2.21	2.28	2.35	2.28
2.20	1.22	1.77	1.98	2.09	2.17	2.29	2.37	2.45	2.38
2.30	1.23	1.82	2.03	2.15	2.23	2.37	2.45	2.53	2.48
2.40	1.25	1.87	2.09	2.22	2.30	2.44	2.53	2.62	2.58
2.50	1.27	1.91	2.14	2.28	2.37	2.52	2.60	2.71	2.67
2.60	1.30	1.95	2.20	2.34	2.44	2.60	2.67	2.80	2.76
2.70	1.32	1.99	2.26	2.41	2.51	2.68	2.74	2.98	2.86
2.80	1.35	2.05	2.32	2.47	2.57	2.76	2.82	3.07	2.95
2.90	1.37	2.10	2.39	2.54	2.64	2.83	2.91	3.15	3.05
3.00	1.40	2.15	2.45	2.60	2.72	2.90	3.00	3.23	3.15
3.10	1.42	2.20	2.51	2.66	2.80	2.97	3.10	3.31	3.25
3.20	1.45	2.26	2.57	2.73	2.87	3.05	3.20	3.39	3.35
3.30	1.48	2.31	2.63	2.80	2.93	3.12	3.30	3.47	3.45
3.40	1.51	2.36	2.69	2.87	3.00	3.20	3.39	3.56	3.54
3.50	1.53	2.42	2.74	2.94	3.08	3.27	3.48	3.65	3.63
3.60	1.57	2.47	2.80	3.00	3.15	3.35	3.56	3.75	3.72
3.70	1.60	2.52	2.86	3.07	3.23	3.43	3.65	3.85	3.80

Calculations of Power Loss and Regulation for Three Phase Three-wire Line. The power loss and regulation for a given three-phase, three-wire line having its wires at the vertices of an equilateral triangle are equal to those, respectively, for two single-phase two-wire lines of the same size of wire and the same spacing and supplying a load equal to that supplied by the given three-phase line.

Example: To find the power loss and the per cent regulation in a three-phase, three-wire line 60,000 ft. long composed of No. 1

A.W.G. (B. & S.) hard-drawn solid copper wire delivering 4000 kw. at the receiver; conductors at the vertices of an equilateral *triangle and spaced 36 in.; frequency, 60 cycles per second; elec*tromotive force at receiver, 22,000 volts; power factor at receiver, 0.90 (current lagging).

Solution: The power loss and per cent regulation in this line will be equal to the power loss and per cent regulation, respectively, in delivering 2000 kw. at the receiver over each of two single-phase, two-wire transmission lines 60,000 ft. long composed of No. 1 A.W.G. (B. & S.) hard-drawn solid copper wire; spacing of conductors, 36 in.; frequency, 60 cycles per second; electromotive force at receiver, 22,000 volts; power factor at receiver, 0.90 (current lagging). Solving such a single-phase line as on pages 701 to 704, the loss is 155.7 kw. and the regulation is 9.5 per cent. That is, the loss and regulation in the above three-phase line are 311.4 kw. and 9.5 per cent, respectively.

Positive Feeder System

The design of distribution feeders and the location of substations must be considered together in determining the best distribution system for any particular track, equipment and condition of traffic. There is no general method of locating substations and designing distribution feeders. Each system must be dealt with according to its particular requirements and limitations. A change in track, equipment, condition of traffic, any or all of them, may not be sufficient to justify a relocation of substations unless the portable substation be used, but may be such as to necessitate a redesign of feeders. It should be noted that the use of the portable substation demands a provision for ample transmission supply to the point at which the substation is to be connected to the transmission line. The distribution system must be so designed that the maximum drop in potential to cars at all points on the line shall not exceed that which will leave a sufficient voltage for the satisfactory operation of traction motors and auxiliary apparatus at such points. In addition to this, the conductors must be of such a size and length and must be so related that each will carry its share of current without undue heating. The original distribution should be designed to fill these requirements at the minimum yearly cost possible for the particular installation being investigated and this should be attempted in all future developments of that installation. In many cases the limitations met in such future development will prevent the attainment of such a low yearly cost as was possible in the case of the original distribution at the beginning of its operation. In either case the quality of service required should first be considered, and the financial considerations made secondary.

The design of feeders and the location of substation must be according to load and its distance rather than distance alone, thus for the same drop in potential the distance from substation to a given load would have to be less than that with a smaller load. That the energy be delivered with no greater than the maximum allowable drop in potential must be the first provision. That is, the energy must be delivered at a potential at least sufficient to accelerate the trains, enable them to climb grades, operate auxiliary apparatus such as air compressors and contactors and furnish satisfactory lights, heat and ventilation.

A graphical train schedule (see page 120) is useful in the study of an interurban distribution system. This shows the location and speed of trains at any and all parts of the day, it also shows the location of stations, turnouts, grades and curves. Trolley, feeder, transmission line layouts and substation locations may be added to the graphical schedule for convenience in the future. From the schedule the condition of regular traffic which will cause the greatest drop in potential may be approximated. The condition of greatest demand upon a given substation may also be approximated from the schedule.

Where local regulations require that an insulated return such as may be provided by a double trolley or conduit system shall be used, the supply and return portions of the distribution are generally designed alike. Where the track rails are used for a return the supply feeders must be so designed that the potential drop in the supply feeders shall not be greater than the difference between the total allowable drop from substation to car and the drop in the track return.

Distribution systems divide into two general classes, namely, interurban and city. These classes are characterized by the frequency of service. The interurban distribution must be designed to limit the maximum drop to starting trains or trains operating under very severe grade conditions. In city distribution where the current required to start an individual train alters the average current but slightly, the distribution is designed for the average current demanded by the trains operating on the section under consideration. Such design will be further influenced by concentrated rush loads of considerable duration.

The position and equipment of substations will often be determined by prominent load centers together with real estate and other business conditions peculiar to the problem in hand. Where, however, there is not such a guide it may be necessary to decide upon the number of substations necessary to feed the section under consideration. The number of stations with which there will result the least total annual cost will in this case be the best. This may be decided upon by calculating the total annual cost for each of several numbers of stations. For convenience in arriving at this total cost the value of each of the eight following parts may be determined and these added together:

1. Annual charges on substation buildings and land.
2. Annual charges on substation equipment.
3. Annual charges on distribution conductor (feeders, trolley wire, third rail, their supports, etc.).
4. Annual charges on transmission system.
5. Annual cost of substation attendance.
6. Annual cost of substation losses.
7. Annual cost of losses in distribution conductors.
8. Annual cost of transmission losses.

Note that the recent practical development of the automatic substation makes this an important factor in such determinations. Whether or not the substations will be automatic or manually operated may make a considerable difference in the values assigned to numbers 2, 5 and 6 as listed above.

H. F. Parshall, British Institution of Civil Engineers, 1914, draws the following conclusions from an elaborate study based largely upon operating costs of the Central London Railway. (Note that in this consideration, Parshall evidently did not have the automatic substation in mind.) They illustrate briefly the influence of the important factors which determine the cost of a distribution system. With the given energy consumption per unit of length of line that follows from a given train movement, the capacity of the substations increases directly with the distance between them. The energy loss in distribution conductors of a given section varies with the cube of the distance between substations. The cost of attendance is within wide limits independent of the size of the substation. The cost of the plant per kilowatt falls off with the size of the units, but the maintenance and renewals per kilowatt are more or less constant. With rotary converter substations and a working voltage of 600, and for certain assumed energy consumption, the most economical substation spacings were 8½ miles, 5½ miles and 3¼ miles for train service of six, twelve and twenty-four trains per hour, respectively. For a working voltage of 1200, the substation spacings were 11 miles, 7½ miles and 5 miles, respectively, while when 2400 volts was adopted the most economical substation spacings were 16 miles, 12 miles and 8½ miles for the three train services, respectively. With the present arrangement of rotary converter substations, there was little advantage in a higher voltage than 2400 for the track conductor. The economy of higher voltages was shown to be approximately the same whatever the train service. As between 600 volts and 1200 volts there was a saving of 14 per cent in the total annual costs of the distribution system; as between 1200 volts and 2400 volts there was a further saving of 7 per cent or 21 per cent as between 600 volts and 2400 volts. If the working voltage be further increased to 3600, there was a decrease in total annual expenditure on substation and overhead conductor equipment of only 3 per cent, which would be less than the additional cost of the rolling stock. For single phase distribution at 5000 volts the most economical substation spacings were 31 miles, 24 miles and 16 miles for train services of two, three and six trains per hour, respectively. At 10,000 volts single phase, the most economical substation spacings were 45 miles, 34 miles and 26 miles for the same three train services, respectively. With three phase distribution at 5000 volts, the most economical distances between substations were 38 miles, 31 miles and 18 miles for the same respective train services. In most of these last cases, however, the economical distance between substations thus determined was greater than would be permissible in practice from considerations of both traffic operation and voltage drop. Further, in the case of the single phase operation, the lower pressure of

5000 volts was found to be the most economical for certain services and the higher pressures of 10,000 volts, 12,000 volts and 15,000 volts in vogue in Europe were explained by consideration of voltage drop.

W. S. Murray, in commenting on the above quoted paper, points out that Mr. Parshall apparently implied a power factor for the single phase distribution of from 40 to 45 per cent. Mr. Murray states that this is but little more than one-half of that which would obtain in actual practice and points out that the single phase substation spacing should be considerably greater than that given in Mr. Parshall's paper if the proper power factor and the same size train units are assumed.

City Distribution System. The design of a city distribution system may be well outlined by a typical case, namely, the rehabilitation of the Chicago surface lines. The following is from the Second Annual Report of the Board of Supervising Engineers, Chicago Traction:

"It was the sense of the Board that the direct current feeder copper for the Chicago Railways Company and the Chicago City Railway Company should be figured on a basis of 75 amp. per car between the direct current bus bars of substations or power houses, and the point of delivery at the car; it being understood that it is the intention of both roads to carry the voltage somewhat lower than 600 volts at the station bus bars until such time as, through the elimination of low voltage motors and otherwise, they are able to raise the voltage to 600 volts. After discussion the following resolution was unanimously adopted:

"*Resolved*, That the system of secondary or direct current electrical conductors or feeders for the Chicago Railways Company and the Chicago City Railway Company shall be calculated and plans made therefor by the Chief Engineer of the Work on the following basis:

"1. That the direct current bus bar at power houses or substations will be operated at approximately 600 volts.

"2. That an allowance of 40 kw. in power house and substation capacity for each standard double truck car of the type approved by the Board of Supervising Engineers, weighing approximately 26 tons light, or its equivalent, will be provided at each direct current bus bar.

"3. That in calculating the copper for current carrying capacity an allowance of 75 amp. for each standard double truck car, as described above, or its equivalent, shall be allowed.

"4. That an average drop of 50 volts will be allowed between the direct current bus bars and the center of gravity of the trolley section, due provision being made for suitable tie lines to take care of emergency cases.

This refers to the average maximum for the 2-hour morning and evening rush periods. In majority of cases the drop is less than 50 volts.

"5. That the carrying capacity of insulated underground cables shall be calculated upon the following basis:

	Lead covered	Paper covered	Triple braided weatherproof
1,000,000 circ. mils cable amp...	800	1000	1250
500,000 circ. mils cable amp...	500	600	625
350,000 circ. mils cable amp...	375	425	325
4/0 cable amperes............	....	....	325

"In arriving at the kilowatts and amperes per car stated in the foregoing resolution a series of tests was made jointly by representatives of the Chicago City Railway Company, the Chicago Railways Company and the Division of Electrical Transmission and Distribution.

"Tests were first made upon a single car by equipping it with instruments and stationing observers upon it to record the results in actual operation on different kinds of service.

"Tests were also made upon groups of eighteen to seventy-six cars operating on an isolated trolley section of a mile or more in length by stationing observers to note the cars entering and leaving the section and also to take readings on station switchboards of the current and voltage on the feeders to the sections at 15-second intervals.

Tests on Single Car. Car tested: Chicago City Railway Company's pay-as-you-enter car No. 5446. Scale weight: 55,800 lb. or 27.9 tons. Motor equipment: Four 40-h.p., No. G. E.—80 motors. Motor control: No. K–28–E. Gear ratio: 69 : 17 or 4.06 : 1 with 33-in. wheels. Air brakes: With 16-ft. compressor set for range of 60 to 85 lb. Heaters: Electric, consisting of 14 truss plank heaters, 4 panel heaters, 2 platform heaters. Lighting: Eighteen 16-c.p. side lights, three 32-c.p. center lights, one 16-c.p. platform light, one 32-c.p. headlight.

Summary of Fifteen Tests on Single Car, May 28 to June 1, 1908

	Minimum	Maximum	Average
Passengers, crew and observers........	17.0	63.0	30.4
Weight in tons, car and live load......	29.09	32.31	30.03
Schedule speed in miles per hour.......	7.69	11.04	8.91
Volts at car........................	450.0	545.0	494.0
Heater energy, kw.:			
First point (5 tests)...............	2.865	2.89	2.87
Second point (5 tests)..............	5.006	5.015	5.01
Third point (6 tests)...............	7.62	7.79	7.74
Lighting energy, kw. (5 tests).......	1.603	1.612	1.61
Total energy, kw. at car:			
Motors and compressor............	28.21	46.50	37.19
Motors, compressor and lights.......	29.82	48.11	38.80
Same with 1 point heat............	32.69	50.98	41.61
Same with 2 points heat...........	34.83	53.12	43.81
Same with 3 points heat...........	37.56	55.85	46.54
Amperes at 550 volts at car motors, compressor, lights and 2 points heat.	63.33	96.58	79.66

Summary of Section Tests, June 11, 1908
Two Hours Maximum Service

	Minimum	Maximum	Average
Number of cars on section	23.3	75.7	62.7
Amperes to section at station	1686.0	4076.0	3430.0
Amperes per car at station	49.4	80.2	55.45
Adding 10 per cent for winter load	54.3	88.2	61.00
Adding 12 amperes for lights and 2 points heat	66.3	100.2	73.00
Kw. at station	27.5	44.0	32.5
Adding 10 per cent for winter load	30.3	48.4	35.8
Adding 6.62 for lights and 2 points heat	36.9	55.0	42.4

"It was assumed that the average maximum condition for which feeders should be provided would be when a car was operating with motors, compressor, lights and two points on heaters. On this basis the individual and section tests compare as follows per car:

	Minimum	Maximum	Average
Individual car tests:			
Kw. at car	34.83	53.12	43.81
Amperes at car	63.33	96.58	79.66
Section tests:			
Kw. at station	36.9	55.0	42.4
Amperes at station	66.3	100.2	73.0

"The calculations for feeder requirements for the Chicago City Railway system are briefly outlined as follows:

"1. From the proposed operating schedules the total number of cars which were required during the rush hours was distributed and plotted upon a skeleton map of the system, which is called a 'Spot Map.' The afternoon maximum period is usually the heaviest service period, so that the car distribution for 2 hours of what is styled the 'P. M. Rush' was used on this map.

"2. The trolley sections were then drawn and the number of cars on each multiplied by 75, which gives the total average maximum load for each individual trolley section in amperes. This amount was placed in a small circle at the center of gravity of the trolley section.

"3. A study was then made of the proper location of stations. The best probable locations were selected and a calculation of load centers was made by finding the combined center of gravity of the loads about a given station. If a given system was to be fed by a single power house, the system load center was also determined, which showed the most economical location so far as distribution of copper was concerned for the generating station. If the locations chosen were not the most economical for distribution of copper, studies were made of comparative costs for other locations where the company might have property or where real estate for substation purposes might be obtained to advantage.

"4. After the station locations were definitely settled and the sections to be fed from each station were decided upon, a 'spider diagram' was added to the drawing, which then became a drawing of record, and showed at a glance what sections were fed from any given station and what average maximum load was to be expected upon that section (see Fig. 78).

"5. The most desirable routes for the cables were then determined, drawn upon a diagram and the distances measured from station bus bar to the load center of each trolley section.

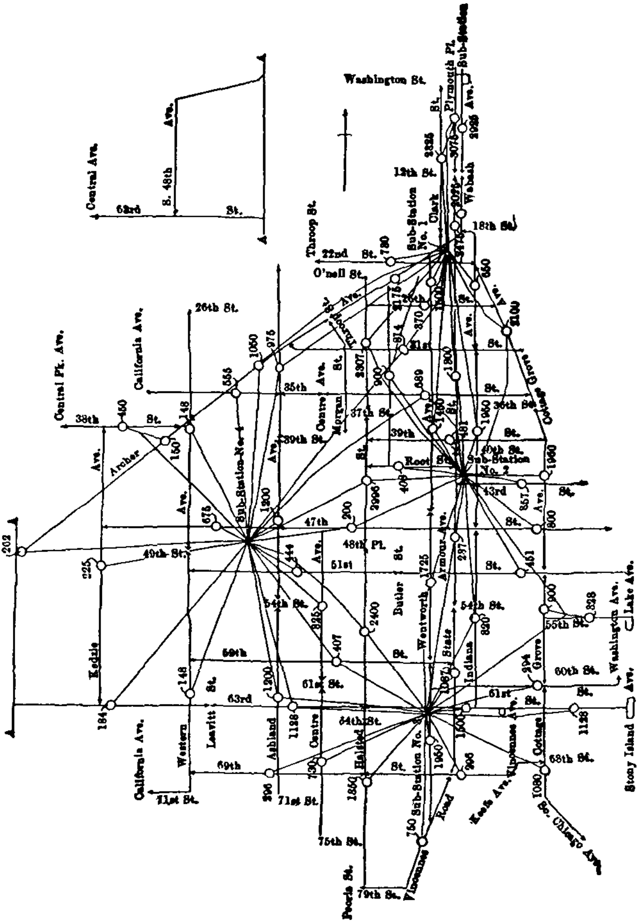


FIG. 78.—Trolley section map and spider diagram. Chicago.

"6. The size of each feeder was then calculated in accordance with the requirements of Resolution No. 562 (page 709). The known elements are:

A. Load in amperes.
B. Distance in feet.
C. Drop in potential, 50 volts more or less.
D. Unit sizes of cables and carrying capacity of each.

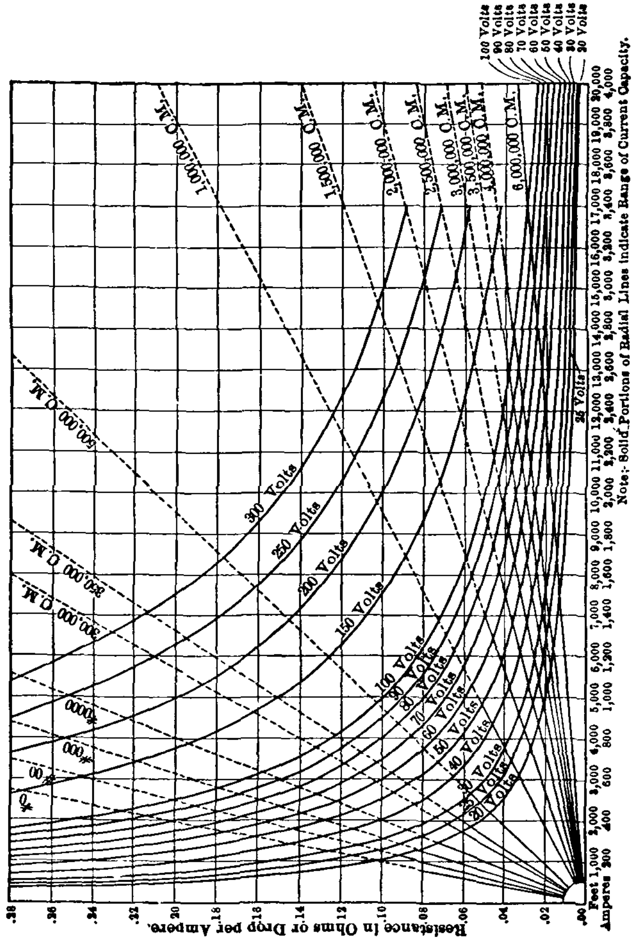


FIG. 79.—Chart for calculating feeder drop and capacity.

From these the size of cable was calculated or read directly from the curves without calculation (see Fig. 79).

"7. A certain number of the more important trolley sections are fed from two separate stations in such a way that in case of the shut-down of one station or of accident to one feeder, the cars on the section could still be operated from the other station, or by means of the other feeder. These are designated as 'tie-sections' and in addition to the above advantages, are so proportioned and calculated that on the whole system in case of the shut-down of one or two stations, a certain proportion of the cars can be carried on the remaining stations by interconnecting through these tie lines.

"8. Where the ordinances required feeders to be placed underground, it was necessary to lay out underground conduit lines. A diagram is used for this; the number of cables over a given section being represented arbitrarily by the numerator of a fraction, and the number of ducts by the denominator. Extra ducts are installed in all conduit lines where practicable, to provide for future growth without tearing up pavements. The percentage of extra ducts will vary for different locations, depending upon the estimates of future requirements."

Feeder Calculations

Significance of symbols in following Cases I to X, inclusive:

$(C.M.)$, $(C.M.)_1$, $(C.M.)_2$ = cross-sectional area of conductor at 20 deg. C., 68 deg. F., expressed in circular mils. [NOTE: This may be the cross-sectional area of one solid conductor or the combined cross-sectional areas of two or more conductors of equal length operating in parallel to give the equivalent cross-sectional area of one large conductor. This includes the case of combined working conductor (trolley wire or third rail) and feeder tied together at short intervals.]

e, e_a, e_b = potential drop in feeder as indicated below, volts.

I = total current uniformly distributed along section, amperes.

I_a and I_b = current taken from conductor at points a and b, respectively, amperes.

I_A, I_B, I_1, I_2, I_3 = current as indicated below, amperes.

K = direct current resistivity of conductor, ohms per circular mil foot.

	K ohms at 20 deg. C., 68 deg. F.
International Annealed Copper Standard, solid	10.37
International Annealed Copper Standard, stranded	10.58
Hard-drawn copper, approximate average, solid	10.65
Hard-drawn copper, approximate average, stranded	10.86
Hard-drawn aluminum, average, solid	16.4
Hard-drawn aluminum, average, stranded	16.7
Siemens Martin steel, stranded	119.7
Third rail or slot contact rail, steel	83.0

NOTE: This is for steel of a resistivity eight times that of the International Annealed Copper Standard. See page 635. For sizes, weights and resistances of bare solid and stranded copper and stranded aluminum see tables on pages 728 to 733.

$l, l_1, l_2, l_3, l_4, l_5$ = length as indicated below, feet.
$r, r_1, r_2, r_3, r_4, r_5$ = resistance per foot of conductor, ohms.
R = resistance per foot of track (two rails) return, ohms.
S, S_1, S_2 = source of supply, this may be generating station, substation or another feeder.

Case I. (Fig. 80.) Load concentrated at a point on conductor of uniform cross-section.

$$(C.M.) = \frac{I_a K l}{e_a}$$

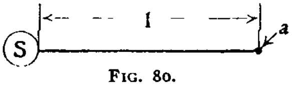


FIG. 80.

$$(\text{Volts drop to point } a) = e_a = I_a r l.$$

Curves for Calculating Drop and Capacity of Feeders. Fig. 79 gives curves by which the solution of problems of Case I may be rapidly approximated. These curves are plotted for copper having a resistance of 10.56 ohms per circular-mil foot. By the application of proportion they may be used for other resistivities. The following examples illustrate the use of the curves:

(*a*) Find the drop on a 500,000 cir.-mil cable, 5000 ft. long and carrying 450 amp. Follow the 5000-ft. ordinate up to the 500,000 cir.-mil radial line, then horizontally until the 450-amp. ordinate is crossed. This intersection at 47.5 gives the volts drop.

(*b*) Find the circular mils cross-section of a cable to carry 800 amp. 8000 ft. with 30 volts drop. Follow the 800-ampere ordinate up to the 30-volt line, then horizontally until the 8000-ft. ordinate is crossed. The location of this intersection gives the size as 2,250,000 cir. mils.

(*c*) Find the distance a 1,500,000 cir.-mil cable will carry 600 amp. with a 40-volt drop. Follow the 600-amp. ordinate to the 40-volt curve, then horizontally to the 1,500,000 cir.-mil line. The ordinate through this line gives the distance as 9350 ft.

Case II. (Fig. 81.) Load concentrated at a point, two conductors of different cross-section in series. (NOTE: This solution is convenient when the size of conductor as determined by Case I lies between two sizes available and it is desirable to so construct the line that the drop to point *a* will be equal to that for the size of wire as determined by Case I.)

$$l_1 = \frac{e_a - I_a r_2 l}{I_a r_1 - I_a r_2}$$

$$l_2 = \frac{e_a - I_a r_1 l}{I_a r_2 - I_a r_1}$$

also $$l_2 = l - l_1$$

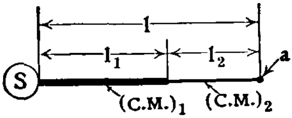


FIG. 81.

Case III. (Fig. 82.) Total load of I amperes uniformly distributed, conductor of uniform cross-section.

$$(C.M.) = \frac{IKl}{2e}$$

$$(\text{Volts drop to end of section}) = e = \frac{Irl}{2}$$

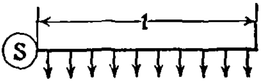


FIG. 82.

Case IV. (Fig. 83.) Load concentrated at a point, conductors as in Fig. 83.
(Volts drop to point a) =

$$e_a = I_a\left[r_1l_1 + \frac{1}{\left(\frac{1}{r_2l_2}\right) + \left(\frac{1}{r_3l_3 + r_4l_4}\right)}\right]$$

(Amperes in conductor A) =

$$I_A = I_a\left[\frac{(r_3l_3 + r_4l_4)}{(r_2l_2 + r_3l_3 + r_4l_4)}\right]$$

(Amperes in conductor B) =

$$I_B = I_a\left[\frac{r_2l_2}{(r_2l_2 + r_3l_3 + r_4l_4)}\right]$$

also $$I_B = I_a - I_A.$$

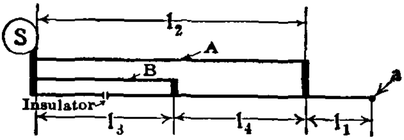


FIG. 83.

Case V. Load concentrated at a point, conductors as in Fig. 84.
(Volts drop to point a) =

$$e_a = Ia\left[\frac{1}{\left(\frac{1}{r_3l_3 + r_4l_4}\right) + \left(\frac{1}{r_2l_2 + r_5l_5}\right)}\right]$$

(Amperes in conductor A) =

$$I_A = I_a\left[\frac{(r_3l_3 + r_4l_4)}{(r_2l_2 + r_3l_3 + r_4l_4 + r_5l_5)}\right]$$

(Amperes in conductor B) =

$$I_B = I_a \left[\frac{(r_2 l_2 + r_5 l_5)}{(r_2 l_2 + r_3 l_3 + r_4 l_4 + r_5 l_5)} \right]$$

also $\quad I_B = I_a - I_A.$

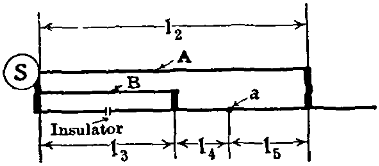


FIG. 84.

Case VI. (Fig. 85.) Load concentrated at a point, conductor of uniform cross-section ($C.M.$, of r ohms resistance per foot), source of supply at each end of conductor, sources of supply at the same potential, track return assumed to be of uniform resistance per foot.

$$(C.M.) = \frac{I_a K l_1 l_2}{e_a l}$$

$$\text{(Volts drop to point } a) = e_a = \frac{I_a r l_1 l_2}{l}$$

$$\text{(Amperes in section } l_1 \text{, supplied by } S_1) = I_1 = \frac{I_a l_2}{l}$$

$$\text{(Amperes in section } l_2 \text{, supplied by } S_2) = I_2 = \frac{I_a l_1}{l}$$

also $\quad I_2 = I_a - I_1$

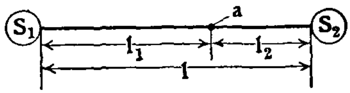


FIG. 85.

Case VII. (Fig. 85.) Load concentrated at a point, conductor not of uniform cross-section but of cross-section $(C.M.)_1$ having a resistance of r_1 ohms per foot from source S_1 to point a and of cross-section $(C.M.)_2$ having a resistance of r_2 ohms per foot from source S_2 to point a, source of supply at each end of conductor, sources of supply at the same potential, track return assumed to be of uniform resistance (R ohms per foot).

(Volts drop to point a) =

$$e_a = I_a \left[\frac{r_1 l_1 l_2 (r_2 + R)}{l_1 (r_1 + R) + l_2 (r_2 + R)} \right]$$

(Amperes in section l_1 supplied by S_1) =

$$I_1 = I_a \left[\frac{l_2 (r_2 + R)}{l_1 (r_1 + R) + l_2 (r_2 + R)} \right]$$

(Amperes in section l_2 supplied by S_2) =

$$I_2 = I_a\left[\frac{l_1(r_1+R)}{l_1(r_1+R)+l_2(r_2+R)}\right]$$

also $\quad I_2 = I_a - I_1$

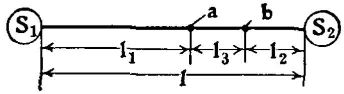


FIG. 86.

Case VIII. (Fig. 86.) Load concentrated at two points, conductor of uniform cross-section (having a resistance of r ohms per foot), source of supply at each end of conductor, sources of supply at the same potential, track return assumed to be of uniform resistance per foot.

(Volts drop to point a) $= e_a = I_1 r l_1$
(Volts drop to point b) $= e_b = I_2 r l_2$
(Amperes in section l_1 supplied by S_1) =

$$I_1 = \frac{I_a(l_2+l_3)+I_b l_2}{l}$$

(Amperes in section l_2 supplied by S_2) =

$$I_2 = \frac{I_a l_1 + I_b(l_1+l_3)}{l}$$

also $\quad I_2 = I_a + I_b - I_1$
(Amperes in section l_3) $= I_3 = I_1 - I_a$

(NOTE: In case the value of I_3 thus obtained is minus, this minus sign indicates that I_3 is supplied by S_2.)

Case IX. (*Fig. 86.*) Load concentrated at two points, conductor *not* of uniform cross-section (sections l_1, l_2 and l_3 having resistances of r_1, r_2 and r_3 ohms per foot respectively), source of supply at each end of conductor, sources of supply at the same potential, track return assumed to be of uniform resistance (R ohms per foot).

(Volts drop to point a) $= e_a = I_1 r_1 l_1$
(Volts drop to point b) $= e_b = I_2 r_2 l_2$
(Amperes in section l_1 supplied by S_1) =

$$I_1 = \frac{I_a l_3(r_3+R) + l_2(I_a+I_b)(r_2+R)}{l_1(r_1+R)+l_2(r_2+R)+l_3(r_3+R)}$$

(Amperes in section l_2 supplied by S_2) =

$$I_2 = \frac{l_1(I_a+I_b)(r_1+R)+I_b l_3(r_3+R)}{l_1(r_1+R)+l_2(r_2+R)+l_3(r_3+R)}$$

also $\quad I_2 = I_a + I_b - I_1$
(Amperes in section l_3) $= I_3 = I_1 - I_a$

(NOTE: In case the value of I_3 thus obtained is minus, this minus sign indicates that I_3 is supplied by S_2.)

Case X. (Fig. 87.) Total load of I amperes uniformly distributed, conductor of uniform cross-section, source of supply at each end of conductor, sources of supply at the same potential.

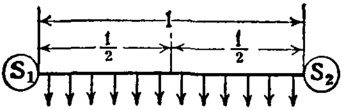


FIG. 87.

$$(\text{Volts drop to center of section}) = e = \frac{Irl}{8}$$

(Cross-section of conductor for drop e to center of section) =

$$(C.M.) = \frac{IKl}{8e}$$

$$(\text{Amperes supplied by } S_1) = I_1 = \frac{I}{2}$$

$$(\text{Amperes supplied by } S_2) = I_2 = \frac{I}{2}$$

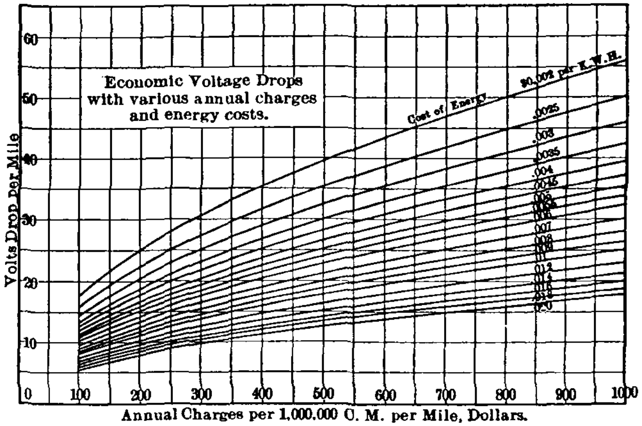


FIG. 88.

Economical Voltage Drop in Conductors. Fig. 88 affords a means of rapidly approximating the most economical voltage drop per mile of conductor for a given cost of energy lost and given annual charges against the installation. These curves were derived according to Kapp's modification of Kelvin's law, which is: "The most economical area of conductor is that for which the annual cost of energy wasted is equal to the annual interest on that portion of the capital outlay which can be considered proportional to the weight of metal used." Example: When the annual charges are $450 per mile per 1,000,000 circular mils of conductor and the cost of energy lost is $0.007 per kilowatt hour, what is the most economical voltage drop per mile of conductor? Beginning at the $450 point on the horizontal scale, follow the vertical to its intersection with the $0.007 curve. From this point follow the horizontal to the scale on the left. The intersection with this scale shows that the most economical voltage drop per mile in this particular case is 20.

Negative Return Systems

Uniformly Distributed Load, Track Extending in One Direction (No Network). The curves and formulas, Figs. 89 to 95, inclusive, from a paper by G. I. Rhodes, A.I.E.E., 1907, are theoretical, giving convenient comparisons of voltage drop, earth currents and amounts of auxiliary negative conductor necessary, for various return systems in which the load is uniformly distributed over the whole line which extends in one direction only from the power station. It is also assumed that the only resistance in the path of the earth current is the contact resistance between rail and ground. This is a safe assumption, as it gives the highest possible values of earth current. These curves show that the way to obtain a minimum of stray current from the grounded rails of a single trolley electric road is to insulate the negative bus bar, and to employ two or more insulated return feeders either so proportioned in resistance or provided with negative boosters as to produce equal potentials at their connecting points to the rails. The following seven general cases are considered: I. No copper in return circuit. II. Copper of uniform section bonded to the rails at short intervals. III. Copper distributed to give uniform drop, bonded to the rails at short intervals. IV. A single insulated negative feeder connected to rails at middle of the line and at the power station. V. A single insulated feeder connecting the rails to the negative bus bar only at the middle of the line. VI. Several insulated feeders. VII. Several insulated feeders with equal potentials at all feed points. In Figs. 89 to 91, inclusive, the ordinates of the curves are proportional to the voltage drop (100 per cent voltage drop $= \frac{\rho I L}{2 S_i}$ = drop to end of line, Case I) and the area beneath the curve is a measure of the earth or leakage current.

The significance of symbols is as follows:

i = current at any point on line
I = total current
v = potential at any point referred to bus bar as zero
l = distance from power station
L = total length of line
L_o = distance to point at which, with bus bar insulated, rail and earth are at same potential
r = resistance per foot of return

ρ = resistance per cir. mil-foot of copper = 10.37 ohms
w = weight per cir. mil-foot of copper = 0.00000302 lb.
S_i = equivalent conductivity of track rails in cir. mils of copper
S_c = cir. mils of return copper at power station
W = weight of return copper
A = area representing leakage current with negative bus bar grounded
a = area representing leakage current with negative bus bar insulated.

I. No return copper:

$$v = \frac{\rho IL}{2S_i},\ A = \frac{\rho IL^2}{3S_i},\ L_e = L\left(1 - \frac{1}{\sqrt{3}}\right),\ a = \frac{1}{3\sqrt{3}}\left[\frac{\rho IL^2}{3S_i}\right]$$

II. Return copper of uniform section:

$$v = \frac{\rho I}{S_i + S_c}\left(l - \frac{l^2}{2L}\right),\ A = \frac{S_i}{S_i + S_c}\left[\frac{\rho IL^2}{3S_i}\right],\ L_e = L\left(1 - \frac{1}{\sqrt{3}}\right)$$

$$a = \frac{S_i}{3\sqrt{3}(S_i + S_c)}\left[\frac{\rho IL^2}{3S_i}\right], \qquad W = wS_cL$$

III. Copper distributed for uniform drop:

$v = \dfrac{\rho Il}{S_i + S_c}$ a straight line up to the point at which $l = L_1 =$ $L\dfrac{S_c}{S_i + S_c}$, but for values of l greater than L_1,

$$v = \frac{\rho I}{S_i}\left[\frac{(L - L_1)(l - L_1)}{L} - \frac{(l - L_1)^2}{2L}\right] + \frac{\rho IL_1}{S_i + S_c}$$

$$A = \frac{S_i(3S_c^2 + 6S_iS_c + 2S_i^2)}{2(S_i + S_c)^3}\left[\frac{\rho IL^2}{3S_i}\right], \quad L_e = \frac{L}{2}\left[1 - \frac{S_i^2}{3(S_i + S_c)^2}\right]$$

$$a = \frac{S_i(3S_c^2 + 6S_iS_c + 2S_i^2)^2}{24(S_i + S_c)^5}\left[\frac{\rho IL^2}{3S_i}\right], \qquad W = \frac{wS_c^2L}{2(S_i + S_c)}$$

IV. Single insulated return feeder from middle of line. Connection to rails at power house:

From $l = o$ to $l = \dfrac{L}{2}$ $\qquad v = \dfrac{\rho I}{S_i}\left[l\left(1 - \dfrac{3S_c}{4(S_i + S_c)}\right) - \dfrac{l^2}{2L}\right]$

From $l = \dfrac{L}{2}$ to $l = L$

$$v = \frac{\rho I}{S_i}\left[\frac{\left(l - \frac{L}{2}\right)}{2} - \frac{\left(l - \frac{L}{2}\right)^2}{2L} + \frac{L}{2}\left(\frac{3S_i}{4(S_i + S_c)}\right)\right]$$

$$A = \left[1 - \frac{27S_c}{32(S_i + S_c)}\right]\frac{\rho IL^2}{3S_i}$$

$$L_e = L\left[1 - \frac{1}{\sqrt{3}}\sqrt{1 - \frac{9S_c}{16(S_i + S_c)}}\right]$$

$$a = \frac{1}{3\sqrt{3}}\left(1 - \frac{9S_c}{16(S_i + S_c)}\right)^{3/2}\left[\frac{\rho IL^2}{3S_i}\right], \qquad W = \frac{wS_cL}{2}$$

V. A single insulated return feeder from middle of line. No other connection to rail:

From $l = 0$ to $l = \frac{L}{2}$ $\qquad v = \rho I L\left[\frac{1}{2S_c} + \frac{1}{8Si}\right] - \frac{\rho I l^2}{2LS_i}$

and from $l = \frac{L}{2}$ to $l = L$ $\qquad v = \frac{\rho I}{S_i}\left[\frac{\left(l - \frac{L}{2}\right)}{2} - \frac{\left(l - \frac{L}{2}\right)^2}{2L} + \frac{LS_i}{2S_c}\right]$

$A = \frac{6S_i + S_c}{4S_c}\left[\frac{\rho I L^2}{3S_i}\right] \qquad L_o = \frac{L}{2\sqrt{3}}$ and $L_o = \frac{L}{2}\left[1 - \frac{1}{\sqrt{3}}\right]$

$a = \frac{1}{12\sqrt{3}}\left[\frac{\rho I L^2}{2S_i}\right]$

(The weight of copper is the same as in Case IV.)

VI. With several insulated feeders the potential curve will take a form similar to that of Case V in which the curve between any two feeders is a portion of the parabola of Case I. If the feeders

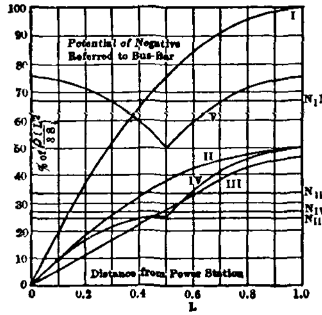


FIG. 89.—Potential curves, Cases I–V.

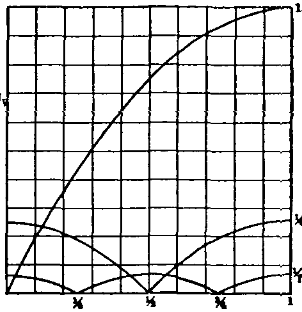


FIG. 90.—Potential of negative referred to bus bar, Case VII.

are of uniform size, the general form of the potential curve will approach the straight line of Case III as the number of feeders is increased indefinitely. With a single feeder the area measuring leakage current is larger than for the case with the copper distributed for uniform drop, and with increasing number of feeders this area will approach it as a limit. With feeders graded in size to give uniform potential at the feed points, the curves will be of the forms in Case VII.

VII. Several insulated feeders with equal potential at feed points. In this case the uniformly equal potentials at the ends of feeders may be secured by properly adjusting the sizes or by means of negative boosters or by resistances in series. By adjustment of sizes, the potential of the track must necessarily be greater than that of the bus bar, and if there is a solid ground at the power station there will be a leakage current due to this difference; but with the use of boosters or resistances the potentials at the ends of the feeders can be maintained uniform with the negative bus bar. For the best results with a given number of feeders it is evident that they must be connected to the rails at such points as will give equal maxima on the potential curves. This may be accomplished in two ways:

1. No connection to rails at power station, and distance between feed points twice the distance from the power house to the first point, and from the last point to the end of the line (see Fig. 90).

2. Connection to rails at power house with the distance between feed points and from the power station to the first feed point equal to twice the distance from the last point to the end of the line (see Fig. 91).

1. With a single feeder and no connection at power house, the feed point for equal maxima will be at the center of the line (see Fig. 90), the potential curve being made up of the upper portions of the parabola of Case I, also shown in Fig. 90.

With the bus bar grounded the area beneath this curve is

$$A = \frac{1}{4}\left[\frac{\rho I L^2}{3S_i}\right]$$

With two feeders the feed points will be one-fourth and three-fourths of the distance from the power house and the area will be

$$A = \frac{1}{16}\left[\frac{\rho I L^2}{3S_i}\right]$$

With n feeders the feed points will be at $\frac{1}{2n}, \frac{3}{2n} \cdots \frac{2n-1}{2n}$ of the distance from the power house to the end of the line, making

$$A = \frac{1}{4n^2}\left[\frac{\rho I L^2}{3S_i}\right]$$

2. With one feeder the point of connection to the rails will be at $= \frac{2}{3}L$, making

$$A = \frac{1}{9}\left[\frac{\rho I L^2}{3S_i}\right]$$

With two feeders the point of connection will be at $l = \frac{2}{5}L$ and $\frac{4}{5}L$, making

$$A = \frac{1}{25}\left[\frac{\rho I L^2}{3S_i}\right]$$

With n feeders the feed points will be at $l = \frac{2}{2n+1}L, \frac{4}{2n+1}L$

$\frac{2n}{2n+1}L$, making

$$A = \frac{1}{(4n + \frac{1}{2})^2}\left[\frac{\rho I L^2}{3S_i}\right]$$

Making a connection to the bus bar, in addition to using feeders, has the effect in the equation $A = \frac{1}{4n^2}\left[\frac{\rho I L^2}{3S_i}\right]$ of increasing the number by half a feeder. This will explain the points as the curve in Fig. 95 of ½, 1½, 2½ feeders, etc.

If the bus bar is insulated, the equation for area representing leakage current is $a = \frac{1}{12\sqrt{3}n^2}\left[\frac{\rho IL^2}{3S_i}\right]$.

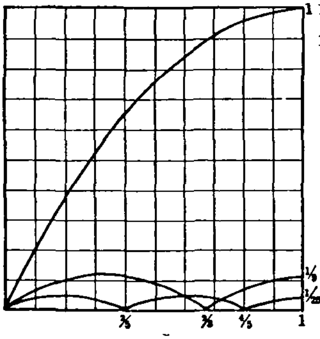


FIG. 91.—Potential of negative referred to bus bar, Case VII.

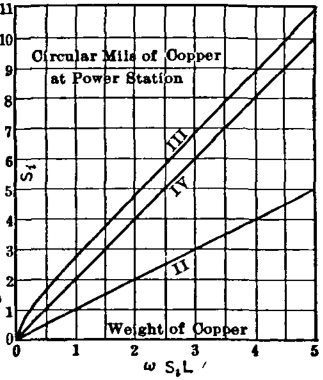


FIG. 92.—Weight of copper in negative feeders.

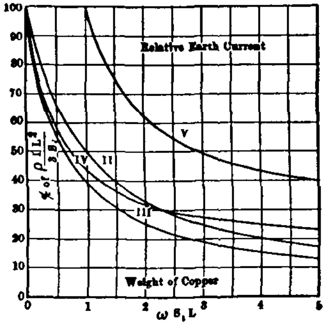


FIG. 93.—Relative earth current, negative bus grounded.

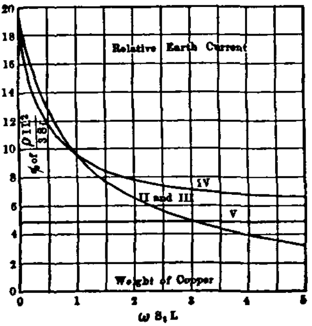


FIG. 94.—Relative earth current, negative bus insulated.

Cross-section of Copper at Power Station. Fig. 92 gives the cross-section of the copper at the power station in the several cases, to give equal weights of copper, the unit of weight being wS_iL, or a weight of copper of length L, which gives a conductivity equal to that of the rails.

Leakage Current. Fig. 93 gives the relative leakage currents for the several cases (in which the bus bar is grounded) plotted to amount of copper used. The unit of leakage current is taken as that which occurs on a line without negative copper, of which the bus bar is grounded. Fig. 94 gives the relative leakage currents with the bus bar insulated. Fig. 95 gives the relative leakage currents for Case VII with bus bar insulated, plotted with the number of feeders as abscissas, the points representing half feeders being as explained above.

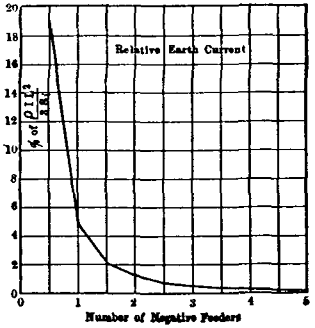


Fig. 95.—Relative earth current, negative bus insulated, Case VII.

Negative Return System for Network. The proper negative return system for a network, such as is made up of the return paths in a city system, may be approximated by the following method: The locations of the points at which the load currents which are assumed to return to the power station are determined. The current and voltage drop in each branch of the network are then

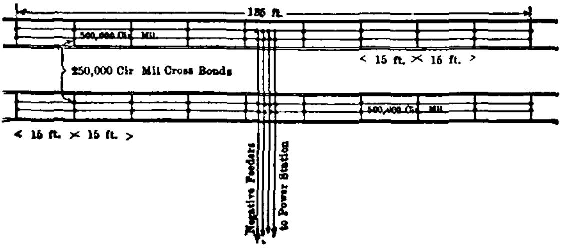


Fig. 96.—Connections between rails and negative feeders.

calculated and the probable amount of auxiliary conductor is then estimated and added to the network. The values of current and voltage drop in each branch of the new network thus found is calculated. The process of adjustment and calculation is continued until the voltage drops are reasonably below the limit placed for good operation, energy economy, to provide against electrolysis or as demanded by local regulation.

Connections from Track to Overhead Line. The conductor used from the rail to manhole or foot of the pole is bare, weatherproof, or rubber covered stranded conductor, with or without lead sheath. Where the rails are in fairly good electrical contact with the earth, it is not necessary that the conductor be insulated. Where the conductor is likely to be disturbed mechanically, it may be encased in pipe. Fig. 96 shows a typical scheme of connections between rails and negative feeders. The size of the longitudinal cables will depend upon the load to be cared for, but for reliability they should be divided into at least two cables per track as shown. The number and size of cross bonds also will depend upon the load. They should be spread out along the track as shown.

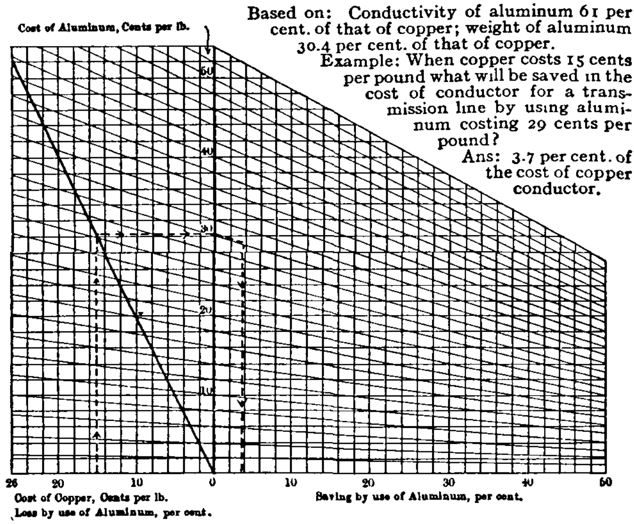


FIG. 97.—Comparison of costs, copper and aluminum conductor.

Aluminum Conductor Compared with Copper Conductor. Since aluminum has a greater temperature coefficient of linear expansion, its use demands a greater height or greater number of poles or towers and greater spacing of conductors to prevent the conductors from sagging dangerously low or swinging too near together. Since aluminum weighs less it will not require such heavy supports and will cost less to string, but because it is softer more care must be taken in handling it, in order that it may not be badly scratched. Aluminum wire of the same conductivity will gather less sleet but presents a greater surface to the wind. Due to its greater radius, leakage from it is less and the skin effect is slightly greater. Because of its lower melting point, there is greater danger from arcing. The initial saving (or loss) in the use of aluminum conductor of such a size as to have a conductivity equal to that of the copper wire which would be required, may be rapidly approximated by the use of Fig. 97. The figure is manipulated as follows for the example shown: From the 15-cent point on the "Cost of Copper" scale follow the vertical to its intersection with the heavy diagonal. Follow the horizontal through this point to its intersection with the "Cost of Aluminum" scale. Follow the diagonal through this point to its intersection with the horizontal through the 29-cent point and from this intersection follow the vertical down to the "Saving by the use of aluminum" scale. Here it is found that, *for the example shown,* the first cost of the aluminum is 3.7 per cent less than the first cost of the necessary copper would have been.

Copper Wire Tables

A value for the standard resistivity of annealed copper was adopted in 1910 by the U. S. Bureau of Standards and the American Institute of Electrical Engineers, and it has also been agreed upon at an International Conference between representatives of United States, Germany and France. It is also of interest to note that the resistivity and temperature coefficient and density of copper on which the following wire tables are based were adopted in 1913 as International Standards for all countries by the International Electro Technical Commission. The value is known as the *Annealed Copper Standard* and is equivalent to 0.15328 ohm (meter, gram) at 20 deg. C.

This value in various units is equal to:
875.20 ohms (mile, pound) at 20 deg. C.

(The term "ohms (mile, pound)" is exactly equivalent to "pounds per mile-ohm," which is sometimes used.)

1.7241 microhms—circular mil, at 20 deg. C.
0.67879 microhm—inch at 20 deg. C.
10.371 ohms (mil, foot) at 20 deg. C.

RESISTANCE AND WEIGHT OF STANDARD ANNEALED COPPER WIRE
(American Wire Gage (B. & S.))

Gage No.	Diameter in mils	Cross-section in circular mils	Weight in pounds		Resistance at 20° C. (=68° F.)	
			Per 1000 feet	Per mile	Ohms per 1000 feet	Ohms per mile
0000	460.	212,000.	641.	3382.	0.0490	0.259
000	410.	168,000.	508.	2682.	.0618	.326
00	365.	133,000.	403.	2127.	.0779	.411
0	325.	106,000.	319.	1687.	.0983	.519
1	289.	83,700.	253.	1337.	.124	.654
2	258.	66,400.	201.	1061.	.156	.825
3	229.	52,600.	159.	841.	.197	1.04
4	204.	41,700.	126.	667.	.248	1 31
5	182.	33,100.	100.	529.	.313	1.65
6	162.	26,300.	79.5	420.	.395	2.09
7	144.	20,800	63.0	333.	.498	2.63
8	128.	16,500.	50.0	264.	.628	3.32
9	114.	13,100.	39.6	209.	.792	4.18
10	102.	10,400.	31.4	166.	.999	5.28
11	91.	8,230.	24.9	132.	1.26	6.65
12	81.	6,530.	19.8	104.	1.59	8.38
13	72.	5,180.	15.7	82.8	2.00	10.6
14	64.	4,110.	12.4	65.6	2.53	13.3
15	57.	3,260	9.86	52.0	3.18	16.8
16	51.	2,580.	7.82	41.3	4.01	21.2
17	45.	2,050.	6.20	32.7	5.06	26.8
18	40.	1,620.	4.92	26.0	6 39	33.7
19	36.	1,290.	3.90	20.6	8.05	42.5
20	32.	1,020.	3.09	16.3	10.1	53.6
21	28.5	810.	2.45	12.9	12.8	67.6
22	25.3	642.	1.94	10.3	16.1	85.2
23	22.6	509.	1.54	8.14	20.4	107.
24	20.1	404.	1.22	6.46	25.7	136.
25	17.9	320.	.970	5 12	32.4	171.
26	15.9	254.	.769	4.06	40.8	216.

Resistance and Weight of Standard Annealed Copper Wire (*Concluded*)

(American Wire Gage (B. & S.))

Gage No.	Diameter in mils	Cross-section in circular mils	Weight in pounds		Resistance at 20° C. (= 68° F.)	
			Per 1000 feet	Per mile	Ohms per 1000 feet	Ohms per mile
27	14.2	202.	.610	3.22	51.5	272.
28	12.6	160.	.484	2.55	64.9	343.
29	11.3	127.	.384	2.03	81.8	432.
30	10.0	101.	.304	1.61	103.	545.
31	8.9	79.7	.241	1.27	130.	687.
32	8.0	63.2	.191	1.01	164.	866.
33	7.1	50.1	.152	0.801	207.	1092.
34	6.3	39.8	.120	.635	261.	1378.
35	5.6	31.5	.0954	.504	329.	1737.
36	5.0	25.0	.0757	.400	415.	2190.
37	4.5	19.8	.0600	.317	523.	2762.
38	4.0	15.7	.0476	.251	660.	3483.
39	3.5	12.5	.0377	.199	832.	4392.
40	3.1	9.9	.0299	.158	1050.	5539.

Note 1: The fundamental resistivity used in calculating the table is the Annealed Copper Standard, viz., 0.153 28 ohm (meter, gram) at 20 deg. C. The temperature coefficient for this particular resistivity is $\alpha_{20} = 0.003\ 93$, or $\alpha_0 = 0.004\ 27$. However, the temperature coefficient is proportional to the conductivity, and hence the change of resistivity per degree C. is a constant, 0.000 597 ohm (meter, gram). The "constant mass" temperature coefficient of any sample is

$$A_t = \frac{0.000\ 597 + 0.000\ 005}{\text{resistivity in ohms (meter, gram) at } t^\circ \text{ C.}}$$

The density is 8.89 grams per cubic centimeter.

Note 2: The values given in the table are only for annealed copper of the standard resistivity. The user of the table must apply the proper correction for copper of any other resistivity. Hard-drawn copper may be taken as about 2.7 per cent higher resistivity than annealed copper.

Note 3: This table is the same as Table VIII, page 48, Circular 31, 2d Edition, of the Bureau of Standards, except that Cross-section in square inches is omitted, resistances are shown at 20 deg. C. instead of 25 deg. C. and 65 deg. C., and Resistances and Weights per mile have been added.

Stranding Table Bare Concentric Cables of Standard Annealed Copper

American wire gage (B. & S.) No.	Cir. mils	Weight in pounds		Resistance at 20° C. (=68° F.)		Class A Bare, insulated or weather-proof cable for aerial use			Class B Insulated cable, for other than aerial use		
		Per 1000 feet	Per mile	Ohms per 1000 feet	Ohms per mile	Number of wires[3]	Diameter of wires, mils	Outside diameter, mils	Number of wires[3]	Diameter of wires, mils	Outside diameter, mils
	2,000,000	6,180	32,630	0.005 29	0.0279	91	148.2	1,632	127	125.5	1,632
	1,900,000	5,870	30,990	.005 57	.0294	91	144.5	1,590	127	122.3	1,590
	1,800,000	5,560	29,360	.005 87	.0310	91	140.6	1,548	127	119.1	1,548
	1,700,000	5,250	27,720	.006 23	.0329	91	136.7	1,504	127	115.7	1,504
	1,600,000	4,940	26,080	.006 61	.0349	91	132.6	1,460	127	112.2	1,460
	1,500,000	4,630	24,450	.007 05	.0372	61	156.8	1,413	91	128.4	1,413
	1,400,000	4,320	22,810	.007 56	.0399	61	151.5	1,365	91	124.0	1,365
	1,300,000	4,010	21,170	.008 14	.0430	61	146.0	1,315	91	119.5	1,315
	1,200,000	3,710	19,590	.008 82	.0466	61	140.3	1,264	91	114.8	1,264
	1,100,000	3,400	17,950	.009 62	.0508	61	134.3	1,210	91	109.9	1,210
	1,000,000	3,090	16,320	.0106	.0560	61	128.0	1,153	61	128.0	1,153
	950,000	2,930	15,470	.0112	.0590	61	124.8	1,124	61	124.8	1,124
	900,000	2,780	14,680	.0118	.0623	61	121.5	1,094	61	121.5	1,094
	850,000	2,620	13,830	.0125	.0658	61	118.0	1,063	61	118.0	1,063
	800,000	2,470	13,040	.0132	.0697	61	114.5	1,031	61	114.5	1,031
	750,000	2,320	12,250	.0141	.0746	61	110.9	999	61	110.9	999
	700,000	2,160	11,400	.0151	.0797	61	107.1	965	61	107.1	965
	650,000	2,010	10,610	.0163	.0860	61	103.2	930	61	103.2	930
	600,000	1,850	9,770	.0177	.0935	37	127.3	893	61	99.2	893
	550,000	1,700	8,980	.0192	.101	37	121.9	855	61	95.0	855

	500,000	1,540	8,130	0212	.112	37	116 2	815	37	116 2	815
	450,000	1,390	7,340	0235	124	37	110 3	773	37	110 3	773
	400,000	1,240	6,550	0265	140	19	145 1	729	37	104 0	729
	350 000	1,080	5,700	0302	.160	19	135 7	682	37	97 3	682
	300,000	926	4,890	0353	186	19	125 6	631	37	90 0	631
	250,000	772	4,080	0424	.224	19	114 7	576	37	82 0	576
0000	212,000	653	3,450	0500	264	19 or 7	105 5–174 0	533	19	105 5	533
000	168 000	518	2,740	0631	333	19 or 7	94 0–154 8	471	19	94 0	471
00	133,000	411	2,170	0795	420	7	137 9	420	19	83 7	420
0	106 000	326	1,720	100	528	7	125 6	374	19	74 5	374
1	83,700	258	1,360	127	671	7	109 3	333	19	66 4	333
2	66,400	205	1,080	160	845	7	97 4	296	7	97 4	296
3	52,600	163	861	201	1 06	7	86 7	263	7	86 7	263
4	41,700	129	681	253	1 34	7	77 2	234	7	77 2	234
5	33,100	102	539	320	1 69	7	68 8	209	7	68 8	209
6	26,000	81 0	428	403	2 13	7	61 2	186	7	61 2	186
7	20,800	64 3	340	508	2 68		144 3	166	7	54 5	166
8	16,500	51 0	269	641	3 38		128 5	147	7	48 6	147

NOTE 1: The fundamental resistivity used in calculating the table is the Annealed Copper Standard, viz, 0 153 28 ohm (meter, gram) at 20 deg C The temperature coefficient is A_{20} = 0 003 93, or A_0 = 0.004 27 The density is 8.89 grams per cubic centimeter.

NOTE 2 This table is in accord with the standards adopted by the Standards Committee of the American Institute of Electrical Engineers in June 1916; in respect to the "number of wires" The values given for "Ohms per 1000 feet" and "Pounds per 1000 feet" are 2 per cent greater than for a solid rod of cross-section equal to the total cross-section of the wires of the cable. This increment of 2 per cent means that the values are correct for cables having a lay of 1 in 15 7. For any other lay, equal to 1 in n, resistance or mass may be calculated by increasing the above tabulated values by $\left(\frac{493}{n^2} - 2\right)$ per cent.

NOTE 3 For intermediate sizes use stranding for next larger size.

Conductors of 0000 A.W.G. and smaller are often made solid and this table of stranding should not be interpreted as excluding this practice.

Class A cable, sizes 0000 and 000 A.W.G. is usually made of seven (7) strands when bare and nineteen (19) strands when insulated or weather-proof.

HARD-DRAWN ALUMINUM WIRE AT 20° C. (OR 68° F.)
(American Wire Gage (B. & S.))

Gage No.	Diameter in mils	Cross-section		Ohms per 1000 ft.	Pounds per 1000 ft.	Pounds per ohm	Feet per ohm
		Cir. mils	Sq. in.				
0000	460.	212,000.	0.166	0.0804	195.	2420.	12,400.
000	410.	168,000.	.132	.101	154.	1520.	9,860.
00	365.	133,000.	.105	.128	122.	957.	7,820.
0	325.	106,000.	.0829	.161	97.0	602.	6,200.
1	289.	83,700.	.0657	.203	76.9	379.	4,920.
2	258.	66,400	.0521	.256	61.0	238.	3,900.
3	229.	52,600.	.0413	.323	48.4	150.	3,090.
4	204.	41,700.	.0328	.408	38.4	94.2	2,450.
5	182.	33,100.	.0260	.514	30.4	59.2	1,950.
6	162.	26,300.	.0206	.648	24.1	37.2	1,540.
7	144.	20,800.	.0164	.817	19.1	23.4	1,220.
8	128.	16,500.	.0130	1.03	15.2	14.7	970.
9	114.	13,100.	.0103	1.30	12.0	9.26	770.
10	102.	10,400.	.008 15	1.64	9.55	5.83	610.
11	91.	8,230.	.006 47	2.07	7.57	3.66	484.
12	81.	6,530.	.005 13	2.61	6.00	2.30	384.
13	72.	5,180.	.004 07	3.29	4.76	1.45	304.
14	64.	4,110.	.003 23	4.14	3.78	0.911	241.
15	57.	3,260.	.002 56	5.22	2.99	.573	191.
16	51.	2,580.	.002 03	6.59	2.37	.360	152.
17	45.	2,050.	.001 61	8.31	1.88	.227	120.

18	40.	1,620.	.001 28	10.5	1.49	.143	95.5
19	36.	1,290.	.001 01	13.2	1.18	.0897	75.7
20	32.	1,020.	.000 802	16.7	0.939	.0564	60.0
21	28.5	810.	.000 636	21.0	.745	.0355	47.6
22	25.3	642.	.000 505	26.5	.591	.0223	37.8
23	22.6	509.	.000 400	33.4	.468	.0140	29.9
24	20.1	404.	.000 317	42.1	.371	.008 82	23.7
25	17.9	320.	.000 252	53.1	.295	.005 55	18.8
26	15.9	254.	.000 200	67.0	.234	.003 49	14.9
27	14.2	202.	.000 158	84.4	.185	.002 19	11.8
28	12.6	160.	.000 126	106.	.147	.001 38	9.39
29	11.3	127.	.000 099 5	134.	.117	.000 868	7.45
30	10.0	101.	.000 078 9	169.	.0924	.000 546	5.91
31	8.9	79.7	.000 062 6	213.	.0733	.000 343	4.68
32	8.0	63.2	.000 049 6	269.	.0581	.000 216	3.72
33	7.1	50.1	.000 039 4	339.	.0461	.000 136	2.95
34	6.3	39.8	.000 031 2	428.	.0365	.000 085 4	2.34
35	5.6	31.5	.000 024 8	540.	.0290	.000 053 7	1.85
36	5.0	25.0	.000 019 6	681.	.0230	.000 033 8	1.47
37	4.5	19.8	.000 015 6	858.	.0182	.000 021 2	1.17
38	4.0	15.7	.000 012 3	1080.	.0145	.000 013 4	0.924
39	3.5	12.5	.000 009 79	1360.	.0115	.000 008 40	.733
40	3.1	9.9	.000 007 77	1720.	.0091	.000 005 28	.581

NOTE: This table is the same as Table XIV, page 54, Circular 31, 2d edition, of the Bureau of Standards and is based on data furnished by the Aluminum Company of America as representing the mean conductivity of its output for all sizes and wire for five years.

APPROXIMATE WEIGHT OF WEATHERPROOF COPPER CONDUCTOR
(American Steel & Wire Co.)

Size, A. W. G. (B. & S.)	Diam. bare conductor, mils	Double braid		Triple braid	
		Lb per 1000 ft.	Lb. per mile	Lb. per 1000 ft.	Lb. per mile
		SOLID CONDUCTOR			
0000	460.0	723	3817	767	4,050
000	409.6	587	3098	629	3,320
00	364.8	467	2467	502	2,650
0	325.0	377	1989	407	2,150
1	289 3	294	1553	316	1,670
2	257.6	239	1264	260	1,370
3	229.4	185	977	199	1,050
4	204.3	151	795	164	865
5	181.9	122	646	135	710
6	162.0	100	529	112	590
8	128.5	66	349	75	395
9	114.4	54	283	62	325
10	101.9	46	241	53	280
12	80.8	30	158	35	185
14	64.1	20	107	25	130
Cir. mils		STRANDED CONDUCTOR			
2,000,000	1.6302	6,690	35,323	7,008	37,000
1,750,000	1.5246	5,894	31,119	6,193	32,700
1,500,000	1 4124	5,098	26,915	5,380	28,400
1,250,000	1.2892	4,264	22,516	4,508	23,800
1,000,000	1.1520	3,456	18,246	3,674	19,400
900,000	1.0935	3,127	16,513	3,332	17,600
800,000	1.0305	2,799	14,779	2,992	15,800
750,000	0.9981	2,635	13,913	2,822	14,900
700,000	0.9639	2,471	13,046	2,650	14,000
600,000	0.8928	2,093	11,052	2,235	11,800
500,000	0.8134	1,765	9,318	1,894	10,000
450,000	0.7721	1,601	8,452	1,724	9,100
400,000	0.7280	1,436	7,584	1,553	8,200
350,000	0 6811	1,248	6,589	1,345	7,100
300,000	0.6283	1,083	5,721	1,174	6,200
250,000	0.5735	907	4,788	985	5,200
0000	0.5275	745	3,935	800	4,220
000	0.4700	604	3,190	653	3,450
00	0 4134	482	2,544	522	2,760
0	0.3684	388	2,051	424	2,240
1	0.3279	303	1,599	328	1,735
2	0.2919	246	1,301	270	1,425
3	0.2601	190	1,004	206	1,090
4	0.2316	155	820	170	900
5	0.2061	126	668	140	740
6	0.1836	103	544	115	610
8	0.1455	68	359	78	410

Allowable Current-carrying Capacity of Copper Wires and Cables

National Board of Fire Underwriters—1913

Size A.W.G. (B. & S.) circular mils.	Rubber insulation, amperes	Other insulations, amperes
18	3	5
16	6	10
14	15	20
12	20	25
10	25	30
8	35	50
6	50	70
5	55	80
4	70	90
3	80	100
2	90	125
1	100	150
0	125	200
00	150	225
000	175	275
0000	225	325
200,000	200	300
300,000	275	400
400,000	325	500
500,000	400	600
600,000	450	680
700,000	500	760
800,000	550	840
900,000	600	920
1,000,000	650	1,000
1,100,000	690	1,080
1,200,000	730	1,150
1,300,000	770	1,220
1,400,000	810	1,290
1,500,000	850	1,360
1,600,000	890	1,430
1,700,000	930	1,490
1,800,000	970	1,550
1,900,000	1,010	1,610
2,000,000	1,050	1,670

LOADING OF BARE STRANDED COPPER CABLE ALONE, WITH ICE COAT, AND WITH ICE COAT AND WIND

Cir. mils	Strands of conductor		Diameter of cable overall	Loading per lineal ft.			
	Number	Diameter		Cable alone	Cable with ½ in. ice coat	Wind at 8 lb. per sq. ft.	Resultant cable ice, and wind
		In.	In.	Lb.	Lb.	Lb.	Lb.
2,000,000	91	0.1482	1.630	6.180	7.518	1 729	7 719
1,750,000	91	0 1387	1.526	5.404	6.677	1.659	6 885
1,500,000	91	0 1284	1.412	4.630	5.831	1.584	6 048
1,250,000	91	0.1172	1.289	3.868	4.992	1.516	5.220
1,000,000	61	0.1280	1.152	3.090	4.128	1.425	4.370
950,000	61	0.1248	1.123	2.935	3.955	1.406	4.200
900,000	61	0.1215	1.094	2.780	3.782	1.386	4.030
850,000	61	0.1181	1.063	2.625	3.607	1.365	3.860
800,000	61	0.1145	1.031	2.470	3.432	1.344	3 688
750,000	61	0.1109	0.998	2.315	3.257	1.322	3 518
700,000	61	0.1071	0.964	2.160	3.080	1.297	3.346
650,000	61	0.1032	0.929	2.005	2.903	1.276	3.174
600,000	61	0.0992	0.893	1.850	2.725	1.252	3.003
550,000	37	0.1219	0.853	1.693	2.543	1.226	2.827
500,000	37	0.1162	0.813	1.540	2.369	1.201	2.659
450,000	37	0.1103	0.772	1.387	2.187	1.175	2.485
400,000	37	0.1040	0.728	1.240	2.011	1.153	2.318
350,000	37	0.0973	0.668	1.083	1.824	1.119	2.140
300,000	19	0.1256	0.629	0.926	1.635	1.086	1.963
250,000	19	0.1147	0.574	0.772	1.446	1.049	1.786
211,600	19	0.1055	0 528	0.653	1.299	1.018	1.630

Loading of Triple Braid Weatherproof Stranded Copper Cable Alone, with Ice Coat and with Ice Coat and Wind

Cir. mils	Strands of conductor		Diameter of cable overall	Loading per lineal foot			
	Number	Diameter		Cable alone	Cable, with ½ in. ice coat	Wind at 8 lb. per sq. ft.	Resultant cable, ice, and wind
		In.	In.	Lb.	Lb.	Lb.	Lb.
2,000,000	91	0.1482	2.000	7.008	8.579	2.000	8.809
1,750,000	91	0.1387	1.906	6.193	7.705	1.937	7.945
1,500,000	91	0.1284	1.781	5.380	6.813	1.858	7.064
1,250,000	91	0.1172	1.656	4.508	5.863	1.771	6.124
1,000,000	61	0.1280	1.531	3.674	4.950	1.687	5.230
950,000	61	0.1248	1.468	3.503	4.745	1.645	4.922
900,000	61	0.1215	1.437	3.332	4.549	1.625	4.831
850,000	61	0.1181	1.406	3.162	4.360	1.604	4.645
800,000	61	0.1145	1.375	2.992	4.170	1.583	4.461
750,000	61	0.1109	1.343	2.822	3.977	1.562	4.272
700,000	61	0.1071	1.312	2.650	3.789	1.541	4.090
650,000	61	0.1032	1.250	2.443	3.543	1.500	3.847
600,000	61	0.0992	1.234	2.235	3.325	1.489	3.556
550,000	37	0.1219	1.156	2.064	3.105	1.437	3.421
500,000	37	0.1162	1.108	1.894	2.908	1.405	3.230
450,000	37	0.1103	1.062	1.724	2.705	1.375	3.035
400,000	37	0.1040	1.031	1.553	2.515	1.354	2.856
350,000	37	0.0973	0.968	1.345	2.267	1.312	2.620
300,000	19	0.1256	0.921	1.174	2.067	1.281	2.431
250,000	19	0.1147	0.875	0.985	1.849	1.250	2.232
211,600	19	0.1055	0.812	0.800	1.624	1.208	2.024

LOADING OF BARE STRANDED ALUMINUM CABLE ALONE, WITH ICE COAT AND WITH ICE COAT AND WIND

Cir. mils	Number of strands	Diameter of cable overall	Loading per lineal foot			
			Cable alone	Cable, with ½ in. ice coat	Wind at 8 lb. per sq. ft.	Resultant cable, ice and wind
		In.	Lb.	Lb.	Lb.	Lb.
1,590,000	61	1.438	1.462	2.679	1.625	3.133
1,575,500	61	1.406	1.393	2.591	1.604	3.047
1,431,000	61	1.359	1.317	2.485	1.573	2.941
1,351,500	61	1.328	1.243	2.392	1.552	2.851
1,272,000	61	1.281	1.171	2.290	1.521	2.749
1,192,500	37	1.250	1.098	2.195	1.500	2.658
1,113,000	37	1.203	1.025	2.095	1.469	2.559
1,033,500	37	1.156	0.950	1.990	1.438	2.455
954,000	37	1.109	0.877	1.888	1.406	2.354
874,500	37	1.063	0.805	1.787	1.375	2.255
795,000	37	1.016	0.732	1.684	1.344	2.155
715,500	37	0.969	0.658	1.587	1.313	2.060
636,000	37	0.906	0.585	1.469	1.271	1.943
556,500	19	0.859	0.512	1.366	1.240	1.845
477,000	19	0.781	0.439	1.244	1.188	1.720
397,500	19	0.719	0.365	1.131	1.146	1.610
336,420	7	0.656	0.310	1.037	1.104	1.515

LOADING OF TRIPLE BRAID WEATHERPROOF STRANDED ALUMINUM CABLE ALONE, WITH ICE COAT AND WITH ICE COAT AND WIND

Cir. mils	Number of strands	Diameter of cable overall	Loading per lineal foot			
			Cable alone	Cable with ½ in. ice coat	Wind at 8 lb. per sq. ft.	Resultant cable, ice and wind
		In.	Lb.	Lb.	Lb.	Lb.
1,590,000	61	1.875	2.070	3.562	1.917	4.045
1,515,500	61	1.813	1.977	3.430	1.875	3.909
1,431,000	61	1.781	1.877	3.310	1.854	3.794
1,351,500	61	1.750	1.779	3.193	1.833	3.680
1,272,000	61	1.719	1.683	3.077	1.813	3.571
1,192,500	37	1.688	1.586	2.961	1.792	3.461
1,113,000	37	1.563	1.489	2.785	1.708	3.267
1,033,500	37	1.531	1.390	2.666	1.688	3.155
954,000	37	1.500	1.293	2.550	1.667	3.047
874,500	37	1.469	1.197	2.434	1.646	2.938
795,000	37	1.438	1.100	2.217	1.625	2.749
715,500	37	1.344	0.994	2.152	1.563	2.660
636,000	37	1.250	0.886	1.983	1.500	2.486
556,500	19	1.188	0.772	1.832	1.458	2.341
477,000	19	1.031	0.657	1.615	1.354	2.107
397,500	19	1.000	0.544	1.487	1.333	1.997
336,420	7	0.938	0.460	1.363	1.298	1.882

Minimum Sags Allowable for Bare Conductor (A.E.R.E.A. Recommended Specification). In the tables, pages 740 to 742, are given the sags at which bare conductors should be strung in order that, when loaded with the specific requirement of ½ in. of ice and a wind load of 8.0 lb. per square foot of projected area at 0 deg. F., the tension in the conductor will not exceed the allowable value of the ultimate strength of the conductor. The sags given for 120 deg. F. are greater in every case than the vertical component of the sags at 0 deg. F. under the maximum wind and ice load. The physical constants of the conductor used are as follows:

Modulus of elasticity

Copper, hard drawn, solid or stranded	16,000,000
Copper, soft drawn, solid	12,000,000
Aluminum, hard drawn	9,000,000

Temperature coefficient

Copper	0.0000096
Aluminum	0.0000128

* Minimum Sags for Stranded Hard-drawn Bare copper Wire

Size, A.W.G. (B.&S.)	Temp. Fahr.	Span, feet: 100 or less	125	150	200	250	300	400	500	600
		Sags: In.	In.	In.	In.	In.	In.	Ft.	Ft.	Ft.
0000	—20	2	3	5	8	13	20.	3.5	6.	10.
	0	2	4	5	9	14	22.	3.5	6.5	10.5
	20	3	4	6	10	16	24.	4.	7.	11.5
	40	3	4	6	11	18	27.	4.5	8.	12.
	60	3	5	7	13	20	31.	5.	8.5	13.
	80	4	6	8	15	24	35.	5.5	9.	13.5
	100	4	7	10	17	27	40.	6.	10.	14.5
	120	5	8	12	20	31	46.	7.	10.5	15.
000	—20	2	3	5	8	13	21.	4.	7.	12.
	0	2	4	5	9	15	23.	4.	7.5	12.5
	20	3	4	6	10	17	25.	4.5	8.5	13.5
	40	3	4	6	12	19	29.	5.	9.	14.
	60	3	5	7	13	22	33.	6.	9.5	15.
	80	4	6	8	15	25	38.	6.5	10.5	15.5
	100	4	7	10	18	29	43.	7.	11.	16.
	120	5	8	12	21	34	49.	7.5	12.	17.
00	—20	2	3	5	9	14	23.	4.5	9.	15.
	0	2	4	5	10	16	26.	5.	9.5	15.5
	20	3	4	6	11	18	29.	5.5	10.	16.
	40	3	4	7	12	21	33.	6.	11.	17.
	60	3	5	7	14	24	37.	6.5	11.5	17.5
	80	4	6	9	16	28	43.	7.	12.	18.
	100	5	7	10	19	32	48.	8.	12.5	18.5
	120	6	9	12	23	37	54.	8.5	13.5	19.5
0	—20	2	3	5	9	16	2.5	5.5	11.5	18.5
	0	2	4	5	10	18	2.5	6.5	12.	19.
	20	3	4	6	11	21	3.	7.	12.5	19.5
	40	3	5	7	13	24	3.5	7.5	13.	20.
	60	3	5	8	15	27	4.	8.	14.	20.5
	80	4	6	9	18	32	4.5	8.5	14.5	21.5
	100	5	7	11	21	37	5.	9.	15.	22.
	120	6	9	13	25	42	5.	9.5	15.5	22.5

* See page 739.

* Minimum Sags for Hard-drawn Bare Copper Wire

Size, A.W.G. (B.&S.)	Temp. Fahr.	Span, feet								
		100 or less	125	150	200	250	300	400	500	600
		Sags								
		In.	In.	In.	In.	In.	Ft.	Ft.	Ft.	Ft.
1	—20	2	4	5	10.	19.	3.	8.	14.5	23.
	0	3	4	6	11.	22.	3.5	8.5	15.	23.5
	20	3	4	6	13.	25.	4.	9.	16.	24.
	40	3	5	7	15.	30.	4.5	9.5	16.	24.5
	60	4	6	8	18.	34.	5.	10.	17.	25.
	80	4	7	10	21.	39.	5.5	10.5	17.	25.5
	100	5	8	12	25.	44.	6.	11.	18.	26.
	120	6	10	16	30.	49.	6.	11.5	18.	26.5
2	—20	2	4	5	12.	25.	4.	10.5	18.5	29.
	0	3	4	6	14.	29.	4.5	11.	19.	29.5
	20	3	5	7	16.	33.	5.	11.5	19.5	30.
	40	3	5	8	19.	39.	5.5	12.	20.	30.5
	60	4	6	10	23.	43.	6.	12.5	20.5	31.
	80	4	7	12	27.	48.	6.5	13.	21.	31.
	100	5	9	14	31.	53.	7.	13.	21.5	31.5
	120	7	11	18	35.	58.	7.5	13.5	22.	32.
3	—20	3	4	6	17.	3.	6.	14.	24.	37.5
	0	3	4	7	20.	3.5	6.5	14.5	24.5	37.5
	20	3	5	8	23.	4.	7.	15.	25.	38.
	40	3	6	10	27.	4.5	7.5	15.	25.	38.
	60	4	7	12	30.	5.	8.	15.5	25.5	38.5
	80	5	9	14	35.	5.5	8.5	16.	26.	39.
	100	6	11	17	39.	5.5	8.5	16.5	26.	39.
	120	8	14	22	44.	6.	9.	16.5	26.5	39.5
4	—20	3	4	8	25.	5.	9.	18.	31.	46.
	0	3	5	9	29.	5.5	9.	18.5	31.5	46.
	20	3	6	11	33.	6.	9.5	19.	31.5	46.5
	40	4	7	13	38.	6.5	10.	19.	32.	46.5
	60	4	9	16	42.	6.5	10.	19.5	32.5	47.
	80	5	11	19	46.	7.	10.5	19.5	32.5	47.5
	100	7	13	23	50.	7.5	11.	20.	32.5	47.5
	120	9	16	27	54.	7.5	11.	20.5	33.	48.
6	—20	3	8	22	5.5	10.	15.	30.	.....	.....
	0	4	10	26	6.	10.	15.	30.	.....	.....
	20	5	13	30	6.	10.5	15.5	30.5	.....	.....
	40	6	16	33	6.	10.5	15.5	30.5	.....	....
	60	8	19	36	6.5	11.	16.	31.	.....	....
	80	10	22	39	6.5	11	16.	31.	.....	.....
	100	13	25	41	7.	11.5	16.5	31.	.....	.....
	120	16	28	44	7.	11.5	16.5	31.5	.....	....

* See page 739.

Minimum Sags for Stranded Bare Aluminum Wire

Size, A.W.G. (B.&S.)	Temp. Fahr.	Span, feet									
		80 or less	100	125	150	200	250	300	400	500	600
		Sags									
		In.	In.	In.	In.	In.	Ft.	Ft.	Ft	Ft.	Ft.
0000	—20	1	2	3	5	11.	2.5	5.	11.	19.	29.
	0	1	2	3	6	15.	3.	5.5	12.	19.5	29.5
	20	2	3	5	8	21.	3.5	6.	12.5	20.5	30.
	40	2	4	7	11	27.	4.5	7.	13.	21.	31.
	60	4	6	11	17	34.	5.	7.5	13.5	21.5	31.5
	80	6	10	16	22	41.	5.5	8.	14.	22.	32.
	100	10	14	20	27	46.	6.	8.5	14.5	22.5	33.
	120	13	18	25	32	52.	6.5	9.	15.	23.	33.5
000	—20	1	2	3	5	12.	3.	5.5	13.	·22.	33.5
	0	1	2	4	6	17.	3.5	6.5	13.5	22.5	34.
	20	2	3	5	8	24.	4.	7.	14.	23.	34.5
	40	2	4	7	12	31.	5.5	7.5	14.5	23.5	35
	60	3	5	11	18	38.	5.5	8.	15.	24.	35.5
	80	6	9	16	23	43.	6.	8.5	15.5	24.5	36.
	100	10	13	20	29	49.	6.5	9.	16.	25.	36.5
	120	13	17	25	33	54.	7.	9.5	16.5	25.5	37.
00	—20	1	2	3	6	2.	5.	8.5	16.5	28.	42.
	0	2	2	4	8	2.5	5.5	9.	17.	28.5	42.5
	20	2	3	6	12	3.	6.	9.	17.5	29.	43.
	40	2	4	9	18	3.5	6.5	9.5	18.	29.5	43.
	60	4	7	14	24	4.	7.	10.	18.5	29.5	43.5
	80	7	12	19	29	4.5	7.	10.5	19.	30.	44.
	100	10	16	24	33	5.	7.5	11.	19.5	30.5	44.5
	120	14	19	28	38	5.5	8.	11.5	20.	31.	44.5
0	—20	1	2	4	9	3.5	7.	10.5	21.	36.5	.....
	0	2	3	6	14	4.	7.	11.	21.5	36.5	.....
	20	2	4	8	20	4.5	7.5	11.5	22.	37.	.....
	40	3	6	13	26	5.	8.	12.	22.	37.	.....
	60	5	10	18	31	5.	8.5	12.	22.5	37.5	.....
	80	8	14	23	35	5.5	8.5	12.5	23.	38.	.....
	100	12	18	27	39	6.	9.	13.	23.	38.	.....
	120	15	21	31	43	6.	9.5	13.5	23.5	38.5	.....
1	—20	1	3	7	20	5.	9.	13 5	26.5	43.5	.....
	0	2	4	11	25	5.5	9.	14.	27.	43.5	.....
	20	2	5	16	30	5.5	9.5	14.5	27.	44.	.....
	40	4	9	21	34	6.	10.	14.5	27.5	44.	.....
	60	7	13	25	39	6.5	10.	15.	27.5	44.5	.....
	80	10	18	29	42	6.5	10.5	15.5	28.	44.5	.....
	100	14	21	32	45	7.	11.	15.5	28.	45.	.....
	120	17	24	36	49	7.	11.	16.	28.5	45.	.....

* See page 739.

The values of inductive reactance in the following tables are from the N. E. L. A. Handbook on Overhead Construction, 1914

Inductive Reactance

Ohms per 1000 Ft. (for one wire)—Solid Conductor 25 Cycles

Inter-axial distance, inches	Size of wire—A. W. G. (B. & S.)											
	0000	000	00	0	1	2	3	4	5	6	7	8
3/8	....	....	0.00926	0.01037	0.01148	0.01259	0.01370	0.01481	0.01591	0.01702	0.01813	0.01924
1/2	0.00982	0.01093	0.01204	0.01315	0.01426	0.01537	0.01648	0.01759	0.01870	0.01981	0.02092	0.02203
3/4	0.01368	0.01479	0.01590	0.01701	0.01812	0.01923	0.02034	0.02145	0.02256	0.02367	0.02478	0.02589
1	0.01644	0.01755	0.01866	0.01977	0.02088	0.02199	0.02310	0.02421	0.02532	0.02643	0.02754	0.02865
2	0.02306	0.02417	0.02528	0.02639	0.02750	0.02861	0.02972	0.03083	0.03194	0.03305	0.03416	0.03527
3	0.02691	0.02802	0.02913	0.03024	0.03135	0.03246	0.03357	0.03468	0.03579	0.03690	0.03801	0.03912
4	0.02968	0.03079	0.03190	0.03301	0.03412	0.03523	0.03634	0.03745	0.03856	0.03967	0.04078	0.04189
5	0.03181	0.03292	0.03403	0.03514	0.03625	0.03736	0.03847	0.03958	0.04069	0.04180	0.04291	0.04402
6	0.03353	0.03464	0.03575	0.03686	0.03797	0.03908	0.04019	0.04130	0.04241	0.04352	0.04463	0.04574
7	0.03501	0.03612	0.03723	0.03834	0.03945	0.04056	0.04167	0.04278	0.04389	0.04500	0.04611	0.04722
8	0.03629	0.03740	0.03851	0.03962	0.04073	0.04184	0.04295	0.04406	0.04517	0.04628	0.04739	0.04850
9	0.03742	0.03853	0.03964	0.04075	0.04186	0.04297	0.04408	0.04519	0.04630	0.04741	0.04852	0.04963
10	0.03843	0.03954	0.04065	0.04176	0.04287	0.04398	0.04509	0.04620	0.04731	0.04842	0.04953	0.05064
11	0.03933	0.04044	0.04155	0.04266	0.04377	0.04488	0.04599	0.04710	0.04821	0.04932	0.05043	0.05154
12	0.04015	0.04126	0.04237	0.04348	0.04459	0.04570	0.04681	0.04792	0.04903	0.05014	0.05125	0.05236
15	0.04230	0.04341	0.04452	0.04563	0.04674	0.04785	0.04896	0.05007	0.05118	0.05229	0.05340	0.05451
18	0.04403	0.04514	0.04625	0.04736	0.04847	0.04958	0.05069	0.05180	0.05291	0.05402	0.05513	0.05624
21	0.04551	0.04662	0.04773	0.04884	0.04995	0.05106	0.05217	0.05328	0.05439	0.05550	0.05661	0.05772
24	0.04678	0.04789	0.04900	0.05011	0.05122	0.05233	0.05344	0.05455	0.05566	0.05677	0.05788	0.05899
30	0.04892	0.05003	0.05114	0.05225	0.05336	0.05447	0.05558	0.05669	0.05780	0.05891	0.06002	0.06113
36	0.05064	0.05175	0.05286	0.05397	0.05508	0.05619	0.05730	0.05841	0.05952	0.06063	0.06174	0.06285

INDUCTIVE REACTANCE

Ohms per 1000 Ft. (for one wire) Solid Conductor 25 cycles —*Continued*

Inter-axial distance, inches	Size of wire—A. W. G. (B. & S.)											
	0000	000	00	0	1	2	3	4	5	6	7	8
42	0.05214	0.05325	0.05436	0.05547	0.05658	0.05769	0.05880	0.05991	0.06102	0.06213	0.06324	0.06435
48	0.05339	0.05450	0.05561	0.05672	0.05783	0.05894	0.06005	0.06116	0.06227	0.06338	0.06449	0.06560
54	0.05452	0.05563	0.05674	0.05785	0.05896	0.06027	0.06118	0.06229	0.06340	0.06451	0.06562	0.06673
60	0.05553	0.05664	0.05775	0.05886	0.05997	0.06108	0.06219	0.06330	0.06441	0.06552	0.06663	0.06774
66	0.05644	0.05755	0.05866	0.05977	0.06088	0.06199	0.06310	0.06421	0.06532	0.06643	0.06754	0.06865
72	0.05727	0.05838	0.05949	0.06060	0.06171	0.06282	0.06393	0.06504	0.06615	0.06726	0.06837	0.06948
78	0.05804	0.05915	0.06026	0.06137	0.06248	0.06359	0.06470	0.06581	0.06692	0.06803	0.06914	0.07025
84	0.05875	0.05986	0.06097	0.06208	0.06319	0.06430	0.06541	0.06652	0.06763	0.06874	0.06985	0.07096
90	0.05941	0.06052	0.06163	0.06274	0.06385	0.06496	0.06607	0.06718	0.06829	0.06940	0.07051	0.07162
96	0.06002	0.06113	0.06224	0.06335	0.06446	0.06557	0.06668	0.06779	0.06890	0.07001	0.07112	0.07223
102	0.06060	0.06171	0.06282	0.06393	0.06504	0.06615	0.06726	0.06837	0.06948	0.07059	0.07170	0.07281
108	0.06115	0.06226	0.06337	0.06448	0.06559	0.06670	0.06781	0.06892	0.07003	0.07114	0.07225	0.07336
114	0.06167	0.06278	0.06389	0.06500	0.06611	0.06722	0.06833	0.06944	0.07055	0.07166	0.07277	0.07388
120	0.06216	0.06327	0.06438	0.06549	0.06660	0.06771	0.06882	0.06993	0.07104	0.07215	0.07326	0.07437
126	0.06261	0.06372	0.06483	0.06594	0.06705	0.06816	0.06927	0.07038	0.07149	0.07260	0.07371	0.07482
132	0.06305	0.06416	0.06527	0.06638	0.06749	0.06860	0.06971	0.07082	0.07193	0.07304	0.07415	0.07526
138	0.06348	0.06459	0.06570	0.06681	0.06792	0.06903	0.07014	0.07125	0.07236	0.07347	0.07458	0.07569
144	0.06389	0.06500	0.06611	0.06722	0.06833	0.06944	0.07055	0.07166	0.07277	0.07388	0.07499	0.07610
150	0.06428	0.06539	0.06650	0.06761	0.06872	0.06983	0.07094	0.07205	0.07316	0.07427	0.07538	0.07649
156	0.06465	0.06577	0.06688	0.06799	0.06910	0.07021	0.07132	0.07243	0.07354	0.07465	0.07576	0.07687
162	0.06502	0.06613	0.06724	0.06835	0.06946	0.07057	0.07168	0.07279	0.07390	0.07501	0.07612	0.07723
168	0.06536	0.06647	0.06758	0.06869	0.06980	0.07091	0.07202	0.07313	0.07424	0.07535	0.07646	0.07757
174	0.06569	0.06680	0.06791	0.06902	0.07013	0.07124	0.07235	0.07346	0.07457	0.07568	0.07679	0.07790
180	0.06602	0.06713	0.06824	0.06935	0.07046	0.07157	0.07268	0.07379	0.07490	0.07601	0.07712	0.07823

Inductive Reactance

Ohms per 1000 Ft. (for one wire) Solid Conductor 60 cycles

Inter-axial distance, inches	Size of wire—A. W. G. (B. & S.)											
	0000	000	00	0	1	2	3	4	5	6	7	8
3/8	....	....	0.02228	0.02494	0.02760	0.03026	0.03292	0.03558	0.03824	0.04090	0.04356	0.04622
1/2	0.02356	0.02622	0.02888	0.03154	0.03420	0.03686	0.03952	0.04218	0.04484	0.04750	0.05016	0.05282
3/4	0.03284	0.03550	0.03816	0.04082	0.04348	0.04614	0.04880	0.05146	0.05412	0.05678	0.05944	0.06210
1	0.03945	0.04211	0.04477	0.04743	0.05009	0.05275	0.05541	0.05807	0.06073	0.06339	0.06605	0.06871
2	0.05534	0.05800	0.06066	0.06332	0.06598	0.06864	0.07130	0.07396	0.07662	0.07928	0.08194	0.08460
3	0.06458	0.06724	0.06990	0.07256	0.07522	0.07788	0.08054	0.08320	0.08586	0.08852	0.09118	0.09384
4	0.07123	0.07389	0.07655	0.07921	0.08187	0.08453	0.08719	0.08985	0.09251	0.09517	0.09783	0.10049
5	0.07634	0.07900	0.08166	0.08432	0.08698	0.08964	0.09230	0.09496	0.09762	0.10028	0.10294	0.10560
6	0.08047	0.08313	0.08579	0.08845	0.09111	0.09377	0.09643	0.09909	0.10175	0.10441	0.10707	0.10973
7	0.08403	0.08669	0.08935	0.09201	0.09467	0.09733	0.09999	0.10265	0.10531	0.10797	0.11063	0.11329
8	0.08708	0.08974	0.09240	0.09506	0.09772	0.10038	0.10304	0.10570	0.10836	0.11102	0.11368	0.11634
9	0.08980	0.09246	0.09512	0.09778	0.10044	0.10310	0.10576	0.10842	0.11108	0.11374	0.11640	0.11906
10	0.09223	0.09489	0.09755	0.10021	0.10287	0.10553	0.10819	0.11085	0.11351	0.11617	0.11883	0.12149
11	0.09440	0.09706	0.09972	0.10238	0.10504	0.10770	0.11036	0.11302	0.11568	0.11834	0.12100	0.12366
12	0.09635	0.09901	0.10167	0.10433	0.10699	0.10965	0.11231	0.11497	0.11763	0.12029	0.12295	0.12561
15	0.10153	0.10419	0.10685	0.10951	0.11217	0.11483	0.11749	0.12015	0.12281	0.12547	0.12813	0.13079
18	0.10571	0.10833	0.11099	0.11365	0.11631	0.11897	0.12163	0.12429	0.12695	0.12961	0.13227	0.13493
21	0.10921	0.11187	0.11453	0.11719	0.11985	0.12251	0.12517	0.12783	0.13049	0.13315	0.13581	0.13847
24	0.11227	0.11493	0.11759	0.12025	0.12291	0.12557	0.12823	0.13089	0.13355	0.13621	0.13887	0.14153
30	0.11740	0.12006	0.12272	0.12538	0.12804	0.13070	0.13336	0.13602	0.13868	0.14134	0.14400	0.14666
36	0.12154	0.12420	0.12686	0.12952	0.13218	0.13484	0.13750	0.14016	0.14282	0.14548	0.14814	0.15080

Inductive Reactance

Ohms per 1000 Ft. (for one wire) Solid Conductor 60 Cycles —*Continued*

Inter-axial distance, inches	Size of wire—A. W. G. (B. & S.)											
	0000	000	00	0	1	2	3	4	5	6	7	8
42	0.12512	0.12778	0.13044	0.13310	0.13576	0.13842	0.14108	0.14374	0.14640	0.14906	0.15172	0.15438
48	0.12814	0.13080	0.13346	0.13612	0.13878	0.14144	0.14410	0.14676	0.14942	0.15208	0.15474	0.15740
54	0.13085	0.13351	0.13617	0.13883	0.14149	0.14415	0.14681	0.14947	0.15213	0.15479	0.15745	0.16011
60	0.13327	0.13593	0.13859	0.14125	0.14391	0.14657	0.14923	0.15189	0.15455	0.15721	0.15987	0.16253
66	0.13545	0.13811	0.14077	0.14343	0.14609	0.14875	0.15141	0.15407	0.15673	0.15939	0.16205	0.16471
72	0.13745	0.14011	0.14277	0.14543	0.14809	0.15075	0.15341	0.15607	0.15873	0.16139	0.16405	0.16671
78	0.13930	0.14196	0.14462	0.14728	0.14994	0.15260	0.15526	0.15792	0.16058	0.16324	0.16590	0.16856
84	0.14099	0.14365	0.14631	0.14897	0.15163	0.15429	0.15695	0.15961	0.16227	0.16493	0.16759	0.17025
90	0.14258	0.14524	0.14790	0.15056	0.15322	0.15588	0.15854	0.16120	0.16386	0.16652	0.16918	0.17184
96	0.14405	0.14671	0.14937	0.15203	0.15469	0.15735	0.16001	0.16267	0.16533	0.16799	0.17065	0.17331
102	0.14545	0.14811	0.15077	0.15343	0.15609	0.15875	0.16141	0.16407	0.16673	0.16939	0.17205	0.17471
108	0.14676	0.14942	0.15208	0.15474	0.15740	0.16006	0.16272	0.16538	0.16804	0.17070	0.17336	0.17602
114	0.14801	0.15067	0.15333	0.15599	0.15865	0.16131	0.16397	0.16663	0.16929	0.17195	0.17461	0.17727
120	0.14918	0.15184	0.15450	0.15716	0.15982	0.16248	0.16514	0.16780	0.17046	0.17312	0.17578	0.17844
126	0.15027	0.15293	0.15559	0.15825	0.16091	0.16357	0.16623	0.16889	0.17155	0.17421	0.17687	0.17953
132	0.15133	0.15399	0.15665	0.15931	0.16197	0.16463	0.16729	0.16995	0.17261	0.17527	0.17793	0.18059
138	0.15234	0.15500	0.15766	0.16032	0.16298	0.16564	0.16830	0.17096	0.17362	0.17628	0.17894	0.18160
144	0.15332	0.15598	0.15864	0.16130	0.16396	0.16662	0.16928	0.17194	0.17460	0.17726	0.17992	0.18258
150	0.15427	0.15693	0.15959	0.16225	0.16491	0.16757	0.17023	0.17289	0.17555	0.17821	0.18087	0.18353
156	0.15517	0.15783	0.16049	0.16315	0.16581	0.16847	0.17113	0.17379	0.17645	0.17911	0.18177	0.18443
162	0.15604	0.15870	0.16136	0.16402	0.16668	0.16934	0.17200	0.17466	0.17732	0.17998	0.18264	0.18530
168	0.15686	0.15952	0.16218	0.16484	0.16750	0.17016	0.17282	0.17548	0.17814	0.18080	0.18346	0.18612
174	0.15766	0.16032	0.16298	0.16564	0.16830	0.17096	0.17362	0.17628	0.17894	0.18160	0.18426	0.18692
180	0.15845	0.16111	0.16377	0.16643	0.16909	0.17175	0.17441	0.17707	0.17973	0.18239	0.18505	0.18771

Inductive Reactance

Ohms per 1000 Ft. (for one conductor)—Stranded Conductor 25 Cycles

Inter-axial distance, inches	Circular mils—Size of conductor—A. W. G. (B. & S.)													
	500,000	450,000	400,000	350,000	300,000	250,000	0000	000	00	0	1	2	3	4
1	0.0119	0.0124	0.0130	0.0136	0.0143	0.0152	0.0160	0.0171	0.0182	0.0193	0.0204	0.0216	0.0227	0.0238
2	0.0185	0.0190	0.0196	0.0202	0.0209	0.0218	0.0226	0.0237	0.0248	0.0259	0.0270	0.0282	0.0293	0.0304
3	0.0224	0.0229	0.0235	0.0241	0.0248	0.0257	0.0265	0.0276	0.0287	0.0298	0.0309	0.0321	0.0332	0.0343
4	0.0251	0.0256	0.0262	0.0268	0.0275	0.0284	0.0292	0.0303	0.0314	0.0325	0.0336	0.0348	0.0359	0.0370
5	0.0273	0.0278	0.0284	0.0290	0.0297	0.0306	0.0314	0.0325	0.0336	0.0347	0.0358	0.0370	0.0381	0.0392
6	0.0290	0.0295	0.0301	0.0307	0.0314	0.0323	0.0331	0.0342	0.0353	0.0364	0.0375	0.0387	0.0398	0.0409
9	0.0329	0.0334	0.0340	0.0346	0.0353	0.0362	0.0370	0.0381	0.0392	0.0403	0.0414	0.0426	0.0437	0.0448
12	0.0357	0.0362	0.0368	0.0374	0.0381	0.0390	0.0398	0.0409	0.0420	0.0431	0.0442	0.0454	0.0465	0.0476
18	0.0395	0.0400	0.0406	0.0412	0.0419	0.0428	0.0436	0.0447	0.0458	0.0469	0.0480	0.0492	0.0503	0.0514
24	0.0423	0.0428	0.0434	0.0440	0.0447	0.0456	0.0464	0.0475	0.0486	0.0497	0.0508	0.0520	0.0531	0.0542
30	0.0444	0.0449	0.0455	0.0461	0.0468	0.0477	0.0485	0.0496	0.0507	0.0518	0.0529	0.0541	0.0552	0.0563
36	0.0461	0.0466	0.0472	0.0478	0.0485	0.0494	0.0502	0.0513	0.0524	0.0535	0.0546	0.0558	0.0569	0.0580
42	0.0476	0.0481	0.0487	0.0493	0.0500	0.0509	0.0517	0.0528	0.0539	0.0550	0.0561	0.0573	0.0584	0.0595
48	0.0489	0.0494	0.0500	0.0506	0.0513	0.0522	0.0530	0.0541	0.0552	0.0563	0.0574	0.0586	0.0597	0.0608
54	0.0500	0.0505	0.0511	0.0517	0.0524	0.0533	0.0541	0.0552	0.0563	0.0574	0.0585	0.0597	0.0608	0.0619
60	0.0510	0.0515	0.0521	0.0527	0.0534	0.0543	0.0551	0.0562	0.0573	0.0584	0.0595	0.0607	0.0618	0.0629
72	0.0528	0.0533	0.0539	0.0545	0.0552	0.0561	0.0569	0.0580	0.0591	0.0602	0.0613	0.0625	0.0636	0.0647
84	0.0543	0.0548	0.0554	0.0560	0.0567	0.0576	0.0584	0.0595	0.0606	0.0617	0.0628	0.0640	0.0651	0.0662
96	0.0555	0.0560	0.0566	0.0572	0.0579	0.0588	0.0596	0.0607	0.0618	0.0629	0.0640	0.0652	0.0663	0.0674
108	0.0566	0.0571	0.0577	0.0583	0.0590	0.0599	0.0607	0.0618	0.0629	0.0640	0.0651	0.0663	0.0674	0.0685
120	0.0576	0.0581	0.0587	0.0593	0.0600	0.0609	0.0617	0.0628	0.0639	0.0650	0.0661	0.0673	0.0684	0.0695
132	0.0585	0.0590	0.0596	0.0602	0.0609	0.0618	0.0626	0.0637	0.0648	0.0659	0.0670	0.0682	0.0693	0.0704
144	0.0594	0.0599	0.0605	0.0611	0.0618	0.0627	0.0635	0.0646	0.0657	0.0668	0.0679	0.0691	0.0702	0.0713
156	0.0601	0.0606	0.0612	0.0618	0.0625	0.0634	0.0642	0.0653	0.0664	0.0675	0.0686	0.0698	0.0709	0.0720
168	0.0608	0.0613	0.0619	0.0625	0.0632	0.0641	0.0649	0.0660	0.0671	0.0682	0.0693	0.0705	0.0716	0.0727
180	0.0615	0.0620	0.0626	0.0632	0.0639	0.0648	0.0656	0.0667	0.0678	0.0689	0.0700	0.0712	0.0723	0.0734

Inductive Reactance

Ohms per 1000 Ft. (for one conductor)—Stranded Conductor 60 Cycles

Inter-axial distance, inches	Circular mils—Size of conductor—A. W. G. (B. & S.)													
	500,000	450,000	400,000	350,000	300,000	250,000	0000	000	00	0	1	2	3	4
1	0.0286	0.0297	0.0311	0.0326	0.0344	0.0365	0.0384	0.0411	0.0437	0.0464	0.0491	0.0519	0.0545	0.0572
2	0.0445	0.0456	0.0470	0.0485	0.0503	0.0524	0.0543	0.0570	0.0596	0.0623	0.0650	0.0678	0.0704	0.0731
3	0.0538	0.0549	0.0563	0.0578	0.0596	0.0617	0.0636	0.0663	0.0689	0.0716	0.0743	0.0771	0.0797	0.0824
4	0.0604	0.0615	0.0629	0.0644	0.0662	0.0683	0.0702	0.0729	0.0755	0.0782	0.0809	0.0837	0.0863	0.0890
5	0.0655	0.0666	0.0680	0.0695	0.0713	0.0734	0.0753	0.0780	0.0806	0.0833	0.0860	0.0888	0.0914	0.0941
6	0.0696	0.0707	0.0721	0.0736	0.0754	0.0775	0.0794	0.0821	0.0847	0.0874	0.0901	0.0929	0.0955	0.0982
9	0.0790	0.0801	0.0815	0.0830	0.0848	0.0869	0.0888	0.0915	0.0941	0.0968	0.0995	0.1023	0.1049	0.1076
12	0.0855	0.0866	0.0880	0.0895	0.0913	0.0934	0.0953	0.0980	0.1006	0.1033	0.1060	0.1088	0.1114	0.1141
18	0.0948	0.0959	0.0973	0.0988	0.1006	0.1027	0.1046	0.1073	0.1099	0.1126	0.1153	0.1181	0.1207	0.1234
24	0.1014	0.1025	0.1039	0.1054	0.1072	0.1093	0.1112	0.1139	0.1165	0.1192	0.1219	0.1247	0.1273	0.1300
30	0.1065	0.1076	0.1090	0.1105	0.1123	0.1144	0.1163	0.1190	0.1216	0.1243	0.1270	0.1298	0.1324	0.1351
36	0.1107	0.1118	0.1132	0.1147	0.1165	0.1186	0.1205	0.1232	0.1258	0.1285	0.1312	0.1340	0.1366	0.1393
42	0.1143	0.1154	0.1168	0.1183	0.1201	0.1222	0.1241	0.1268	0.1294	0.1321	0.1348	0.1376	0.1402	0.1429
48	0.1173	0.1184	0.1198	0.1213	0.1231	0.1252	0.1271	0.1298	0.1324	0.1351	0.1378	0.1406	0.1432	0.1459
54	0.1200	0.1211	0.1225	0.1240	0.1258	0.1279	0.1298	0.1325	0.1351	0.1378	0.1405	0.1433	0.1459	0.1486
60	0.1225	0.1236	0.1250	0.1265	0.1283	0.1304	0.1323	0.1350	0.1376	0.1403	0.1430	0.1458	0.1484	0.1511
72	0.1267	0.1278	0.1292	0.1307	0.1325	0.1346	0.1365	0.1392	0.1418	0.1445	0.1472	0.1500	0.1526	0.1553
84	0.1302	0.1313	0.1327	0.1342	0.1360	0.1381	0.1400	0.1427	0.1453	0.1480	0.1507	0.1535	0.1561	0.1588
96	0.1332	0.1343	0.1357	0.1372	0.1390	0.1411	0.1430	0.1457	0.1483	0.1510	0.1537	0.1565	0.1591	0.1618
108	0.1360	0.1371	0.1385	0.1400	0.1418	0.1439	0.1458	0.1485	0.1511	0.1538	0.1565	0.1593	0.1619	0.1646
120	0.1383	0.1394	0.1408	0.1423	0.1441	0.1462	0.1481	0.1508	0.1534	0.1561	0.1588	0.1616	0.1642	0.1669
132	0.1405	0.1416	0.1430	0.1445	0.1463	0.1484	0.1503	0.1530	0.1556	0.1583	0.1610	0.1638	0.1664	0.1691
144	0.1425	0.1436	0.1450	0.1465	0.1483	0.1504	0.1523	0.1550	0.1576	0.1603	0.1630	0.1658	0.1684	0.1711
156	0.1443	0.1454	0.1468	0.1483	0.1501	0.1522	0.1541	0.1568	0.1594	0.1621	0.1648	0.1676	0.1702	0.1729
168	0.1460	0.1471	0.1485	0.1500	0.1518	0.1539	0.1558	0.1585	0.1611	0.1638	0.1665	0.1693	0.1719	0.1746
180	0.1476	0.1487	0.1501	0.1516	0.1534	0.1555	0.1574	0.1601	0.1627	0.1654	0.1681	0.1709	0.1735	0.1762

Preservative Treatment of Poles, Cross-arms and Ties. A joint committee of the Amer. El. Ry. Eng. Assn., made up of members of the Committees on Way Matters, Buildings and Structures, and Power Distribution, reported as follows in 1922, and its recommendations were adopted. (The various specifications referred to may be found in the A.E.R.E.A. Manual.)

After a careful study of service records and other available data, the joint committee on wood preservation arrived at the following conclusions:

1. That some form of creosote oil is the best timber preserving agent now in general use for all purposes, by reason of the fact that its preservative effect is not greatly affected by either rainfall or temperature, and in addition it has a lubricating effect on the wood, which diminishes the injury due to mechanical wear; this combination of qualities places it at the head of all timber preservatives.

2. That where for economical reasons coal tar creosote oil or water gas tar oil are not available, or other conditions of maintenance will not justify the expense of creosote treatment, the adoption of zinc chloride or some other preservative is, without question, justified in the treatment of timber. Climatic conditions will go further in determining the economy of these treatments than in any other, as one can generally figure on doubling the life of untreated timber by their use, and in dry climates this life probably will be extended.

3. That in localities where the rainfall is excessive and with humid atmosphere, where good zinc chloride treatment would be unfavorably influenced by leaching, and in any climate where the checking of the timber is likely to be excessive, or the mechanical abuse of the fiber is extreme, and where it is not considered possible to secure a straight creosote treatment, the introduction of some lubricating agent with the zinc chloride will have a beneficial effect in retarding the destruction of the timber from the above cause. In such treatments, the use of a poorer grade of creosote oil is justified. However, the committee calls attention to the undesirable features of zinc chloride as a wood preservative when the timber is used in electric railway construction (see pages 666 and 775).

4. That because of its extremely poisonous nature to humans, as well as its corrosive action on steel, the use of bi-chloride of mercury as a wood preservative is not recommended.

5. The full-cell or empty-cell pressure treatments of timber have proven to be the most efficient in prolonging its life and should therefore be used wherever possible.

6. That where the expense of some form of pressure treatment is not warranted by the probable results obtained, the open-tank treatment, using either hot bath only or hot and cold baths, offers the next method of securing maximum penetration and impregnation.

7. That where facilities for open-tank treatment or dipping are not available, the results obtained through brush treatment or spraying will more than justify the expense of such treatment.

8. That while the committee recognizes the fact that sodium fluoride or compounds containing sodium fluoride or other fluorides

possess many desirable qualities as wood preserving agents, experiments with its use in this country have not yet reached the point where sufficient data is available on which to base definite conclusions.

9. That in view of the extent of the study of service records as well as laboratory experiments and research work which led up to the adoption of specifications for the three grades of creosote oil, creosote-coal-tar mixture, water-gas-tar mixture and water gas-tar distillate by the American Railway Engineering Association and the adoption of these same specifications by the Amer can Society for Testing Materials and the American Wood Preservers' Association, both of which organizations collaborated with the American Railway Engineering Association in their preparation, and also considering the widespread use of the specifications in the United States, the committee recommends their adoption as Recommended Specifications.

10. That the committee recommends the adoption of Recommended Specifications for carbolineum and similar oils for open-tank and brush treatments, as these treatments, being superficial in character, require an oil of this type in order to give the best results.

11. That, in view of the adoption of standard specifications for coal tar oil, distillate oil and refined water-gas-tar for the preservation of wood block by the American Wood Preservers' Association in cooperation with the American Society for Municipal Improvements, and the extent of the use of such specifications throughout the United States, the committee recommends their adoption as Recommended Specifications.

12. That the committee recommends the adoption as Recommended Specifications of the "Specifications for Preservative Treatment of Wood," which have been adopted as standard by the American Railway Engineering Association, as these specifications cover the principal methods of impregnating timber by the various pressure processes.

13. That the committee submits in addition to these specifications for pressure treatment, certain other specifications for the treatment of timber by the open-tank and brush methods, for the benefit of those companies having occasion to use such methods, which specifications represent the result of a careful study of existing data and experience with these methods of treatment, and which we recommend for adoption as Recommended Specifications.

14. The American Railway Engineering Association, the American Society for Testing Materials and the American Wood Preservers' Association, recognizing the necessity of standardizing methods of analysis of creosote oil in connection with standardization of specifications for the oils themselves, have, after a large amount of study and investigation, adopted standard specifications for "Standard Methods of Sampling and Analysis of Creosote Oil." The committee appreciates the varying results which can be obtained from the analysis of the same oil by different methods, and while there may be some features of other methods offering

advantages over those referred to above, in the absence of facilities or time for research and experimental work of its own, and realizing that the specifications adopted by the three national associations above represent the consensus of opinion of some of the best authorities on the subject in the United States, recommends their adoption as Recommended Specifications.

15. There are also included specifications for the analysis of carbolineum and similar oils, since it was felt that the retort method of analysis did not give sufficiently accurate determinations of the fractions distilling off at the various lower temperatures, the limits of some of these fractions being held down to quite small percentages under the specifications. In the flask analysis with the bulb opposite the condensing tube, the temperature recorded is always the temperature of the vapor actually distilling off into the condensing tube, whereas in the retort analysis at the beginning of the distillation the temperature of the vapor which is recorded is that one-half inch above the surface of the oil, while, as the distillation progresses, this distance constantly increases and the temperatures recorded are those of different parts of the vapor. There is some difference of opinion with respect to the question of precision of fractionation between a retort and the usual type of distilling flask. There is a difference between these two devices with respect to the temperatures at which different parts of the distillate pass over to be condensed, that is, any given percentage of the sample will pass over at a lower temperature when the retort is used. The committee believes, however, that where the limits of the fractions are held to comparatively small limits the flask analysis offers the most scientifically accurate method of making this determination, and recommends the adoption of these specifications as Recommended Specifications.

16. The American Wood Preservers' Association, working in conjunction with the American Society for Testing Materials and the American Society for Municipal Improvements, has adopted specifications for "Creosoted Wood Block Pavement" as their standard, and the American Society for Municipal Improvements, acting alone in the matter, has adopted specifications covering the method of laying the block. The committee is convinced that a large percentage of wood block paving failures has been due to incorrect methods of laying the block. We feel that the principle of placing a sand cushion under wood block and of laying the block with sand joints is wrong and has been the cause of a large percentage of the ultimate failures of such pavement, and also that the use of dry sand-cement cushion is not the best practice. While we feel that wood block is not a desirable type of pavement in street railway tracks, we recognize that where its use is required, either by choice or by reason of municipal requirement, it should be laid in the most approved method possible and in such a way as experience has proved will insure the best results. We feel that the time has come to eliminate the use of a sand or mortar cushion and sand joints, and that the best results can only be expected where the blocks are laid on a smooth concrete base of adequate depth and bedded in a thin coat of coal-tar pitch or other suitable

water-proofing material and the joints poured with either a pitch or asphalt filler. In making this statement, we realize that it is a radical departure from the long established method of laying wood block, but the results being obtained where wood block has been laid by this method have more than justified the additional expense which it involves, and the process of manufacture makes it possible to keep the variation in depth of blocks within such limits as will make the method practical. We therefore recommend the adoption of these specifications as Recommended Specifications.

SECTION XI

SIGNALS AND COMMUNICATION

The requisites of signals on an electric railway vary with many conditions; whether the system is double or single track; whether it is a city, suburban or interurban system; whether the track is straight or has many curves; whether the track is level or has many grades; speed, headway and braking possibility of trains; and whether direct current or alternating current is used to propel the trains. The selection of proper signal apparatus and arrangement for a given system, therefore, calls for an individual study of that particular system. Development due to such studies has produced a great many types and varieties of signal apparatus.

Block System. The methods and rules by means of which the movements of a train on a section of track are controlled relative to the movements of another train or other trains on the same track and section constitute a block system.

Block. A section of track the use of which by trains is thus controlled constitutes a block.

Block Signal. A block signal is a signal which controls the use of a block.

Home Block Signal. A home block signal is a signal which controls trains entering and using a block.

Distant Block Signal. A distant block signal is a signal used in connection with the home block signal to regulate the approach of a train to that home block signal.

Signal Location and Arrangement. The most important features influencing the location of signals are: (*a*) Maximum braking distance of trains as affected by their speed and weight and by the grades encountered; (*b*) location of sidings, stations, curves and like physical features of the right of way; (*c*) headway of trains to maintain the schedule; (*d*) the distance at which signals can be easily read and interpreted.

Signals are usually, where possible, placed upon the right-hand side of the track looking in the direction of the traffic which they govern.

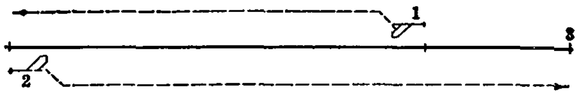


FIG. 1.—Curve protection

Curve Protection. (Fig. 1.) Signal 1 governs west-bound cars and Signal 2 east-bound, the dangerous or obscured track lying between the two signals. The dotted lines show the control limits of the two signals, or, in other words, the extent of the track which controls the signals. A car anywhere between Signal 2 and point 3 will cause Signal 2 to remain in the "Stop" indication, while

Signal 1 is affected only when the car is between Signals 1 and 2. *Should two following cars approach a protected curve close together,* the second car would be stopped by the signal until the first car had passed beyond the limits of control of that signal.

Preliminary. Were Signal 2 controlled only up to Signal 1, it would then be possible for two cars to simultaneously pass Signals 1 and 2, each receiving a proceed indication. This may be avoided by the use of a setting section, or *preliminary,* extending beyond one of the Signals, as 1–3. This preliminary section should not ordinarily be less than 1000 ft. in length and a length of 1500 ft. is preferable. By the use of this preliminary a car running west sets Signal 2 to stop after passing point 3, and in case an east-bound car has not already passed Signal 2, it would be stopped by it. In case, however, the east-bound car had passed Signal 2 before the one west-bound passed point 3, Signal 1 would be set to stop, being controlled to Signal 2, and the west-bound car would be stopped at Signal 1. *The fact that Signals 1 or 2 are in the proceed indication gives the governed car the right to proceed only up to the opposing signal, or through the block,* for if, as in the last-mentioned case, an east-bound car passes Signal 2 before the west-bound car passes point 3, the latter will stop at Signal 1 and the opposing car, having had a proceed signal at 2, might meet with an accident. If a preliminary were used on both ends, or the control of each signal were extended out beyond the other, two opposing cars might get into the preliminary sections at the same time. In this event both signals would assume the stop indication and neither car could proceed. It is therefore desirable to locate the signals, in a case of this kind, at least within braking distance from the point of curve so that if one car is standing at a signal an opposing car rounding the curve will be able to stop before reaching the signal.

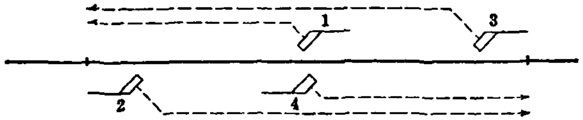


FIG. 2.—Intermediate signals.

Intermediate Signals. (Fig. 2.) Where the use of a preliminary (see preceding paragraph) is prevented, intermediate signals are sometimes used. The purpose of these intermediate signals (1 and 4) is to afford protection if two trains running in opposite directions should happen to pass points 2 and 3 at the same time.

Signaling Schemes for Suburban and Interurban Service

The selection of a signaling scheme for a given service calls for a special study of each case. The following general schemes (Figs. 3 to 13, inclusive) are illustrative of common present day practice.

Single-track Suburban Railway Headway 5 to 30 Minutes, Speed Not Exceeding 20 Miles per Hour. Employing either non-counting or car-counting signals with trolley contactor control.

(Figs. 3 to 6, inclusive; in which *S* indicates Set, *i.e.*, counting into block, *R* indicates Restore, *i.e.*, counting out of block. Dotted lines from a signal contactor indicate control by that contactor. Arrow heads pointing toward signals indicate setting movements; arrow heads pointing toward contactors indicate restoring movements in the direction as indicated by the arrow head.)

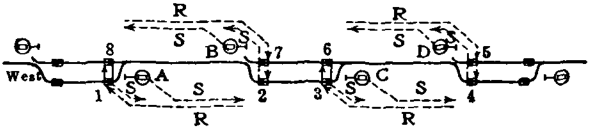


Fig. 3.

Scheme I. (Fig. 3.) East-bound cars take the siding and west-bound cars remain on the main track. An east-bound car from *A* to *B*, by 1, sets *B* to stop, and *A* is changed to register car into *A–B*. Upon reaching siding at *B* the car is counted out of *A–B* at 2. If *C* is in a permissive indication the car may proceed under 3, thereby counting itself into *C–D*. If, however, a west-bound car is in *C–D*, *C* will be at stop and the east-bound car will remain in the siding until the west-bound car has passed *C* which will be restored at 6. For regular movements east-bound cars count into *A–B* and *C–D* at 1 and 3, respectively, and count out at 2 and 4, respectively. West-bound cars count into *D–C* and *B–A* at 5 and 7, respectively, and count out at 6 and 8, respectively. The contactors are shown connected for irregular movements as, for instance, an east-bound car operating 2 counts out of *A–B*, but if for any reason it should back over 2 it would count into *B–A*; but as it again proceeeds eastward it will count out again. West-bound car movements are controlled as illustrated for east-bound, except that the west-bound waits, if necessary, on the main track opposite the siding.

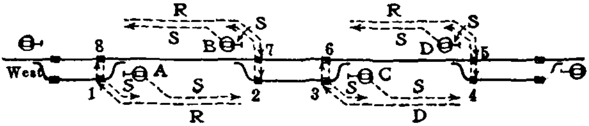


Fig. 4.

Scheme II. (Fig. 4.) Cars in both directions are permitted to pass sidings on the main line; car in both directions are permitted to run through sidings. An east-bound car having operated 1 sets *B* to stop and *A* to register it into *A–B*. If *C* is in a permissive indication the car passes *B* and counts itself out at 7 and proceeding forward counts itself into *C–D*. If, however, *C* is at stop, indicating a west-bound car in *D–C*, the east-bound car runs into the siding and counts itself out of *A–B* at 2. When *D–C* is restored by west-bound car passing 6, the east-bound car then proceeds into *C–D* by operating 3. The reversal of these movements may take place, that is, a west-bound car reaching the siding may pass it on the main track, or if *B* is at stop the car will enter the siding at the east end and wait there until the east-bound car has cleared *B–A* at 7. East-bound cars count into *A–B* and *C–D* at 1 or 8 and 3 or 6, respectively, and count out of *A–B* and *C–D* at 2 or 7 and 4 or 5, respectively. West-bound cars count into *D–C* and *B–A* at 4 or 5 and 2 or 7, respectively, and count out of *D–C* and *B–A* at 3 or 6 and 1 or 8, respectively.

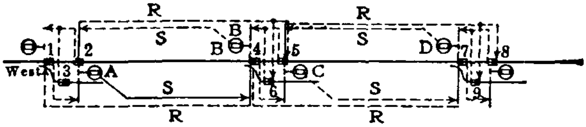


FIG. 5.

Scheme III. (Fig. 5.) East-bound cars enter the siding. An east-bound car approaching siding at *B* has signal *B* at stop protecting it. Upon reaching the siding if *C* shows a permissive indication, the car may proceed on the main line and count itself into *C-D* at 4 and count out of *A–B* at 5. If, however, *C* is found at stop, the car will pull into the siding and count itself out of *A–B* at 6. The west-bound car then passing *C* counts itself into *A–B* at 5 and counts out of *D–C* at 4 and proceeds through *B–A*; the east-bound car will then back out of the siding, count itself into *B-A* at 6 and run on to the main line. Proceeding eastward it will count itself into *C–D* at 4 and continuing forward will count itself out of *A–B* at 5. A west-bound car upon reaching the siding and finding *B* at stop will wait on the main line until the east-bound car takes the siding.

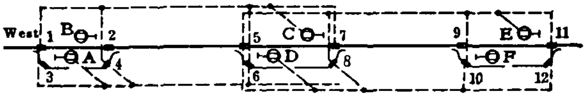


FIG. 6.

Scheme IV. (Fig. 6.) Either east- or west-bound cars take the siding, observing the rule that cars are to head in and back out. An east-bound car in *A–C* has *C* protecting it and upon reaching siding at which *C* is located and finding *D* at permissive, may proceed on the main track and count into *D–E* at 5; continuing forward it will count itself out of *A–C* at 7. If, however, *D* is at stop, the car will head into the siding and count out of *A–C* at 6. The west-bound car then approaching *C* and finding it clear counts into *C–A* at 7 and continuing forward restores *E–D* at 5. The east-bound car in the siding now backs out on to the main line and in passing under 6 counts into *C–A*; then proceeding eastward on the main line counts into *D–E* at 5 and counts out of *C–A* at 7. As the arrangement is symmetrical from the middle of the siding, movements the reverse of the above are controlled in an obvious manner.

Double-track Suburban or Interurban Railway, Headway 1 to 10 Minutes. (Figs. 7 to 10 inclusive.)

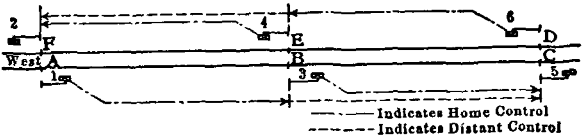


FIG. 7.

Scheme I. (Fig. 7.) Three-position semaphore or light signals with track circuit control. An east-bound car passing 1 sets it to stop, in which position it remains until this car has passed out of *A–B*. Upon passing insulated joints *B*, 3 indicates stop, 1 indicates caution. When the train proceeding eastward passes insulated joints at *C*, 1 will go to clear, 3 to caution and 5 to stop. A similar explanation holds for a west-bound train.

This is the signalling scheme used (*with 3-indication color light* signals) on the Washington, Baltimore & Annapolis Ry. between Baltimore and Washington, where heavy cars are operated at speeds up to 50 mi. per hr. Polarized track circuits are employed, which results in the track relay having three positions corresponding to the three signal indications, and consequently no line wires are required between signal locations except those for transmission of power and the special circuits which may be used for switch indicators. This typical circuit is regarded as the predominating system of control and type of signal for double-track railway installations.

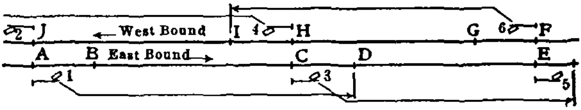


FIG. 8

Scheme II. (Fig. 8.) Two-position *signals with overlap track* circuit control. An east-bound car passing 1 sets it to stop when it passes insulated joints at *A*. The signal remains in this position until the car passes insulated joints at *D*. As the car passes 3, insulated joint *C*, this signal will also go to stop, and while the car is in *C–D* both 1 and 3 will be at stop. The *fact that* 1 *is controlled beyond* 3 to *D* prevents a car obtaining a clear indication at 1 and receiving a stop indication at 3, with the preceding car just beyond this signal. A similar explanation holds for a west-bound car. There are a number of such installations in service, but it is unlikely that *this scheme will be followed to any extent in connection with future* new work.

Scheme III. (Fig. 9.) Trolley contactors. Three indications by lights alone, a pilot light being used in advance of the main signal for an indication that the signal has operated. An east-bound car passing *A*, in case there are no cars between *A* and

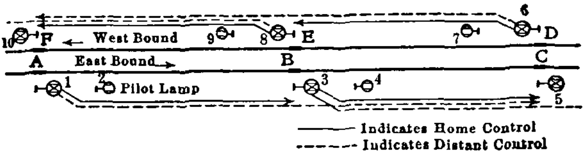


FIG. 9.

C, changes 1 from neutral to proceed. The fact that the pilot light, which is normally extinguished, lights up, shows the motorman that he has counted into the block and gives him the right to proceed. 1 is at stop while the car is in *A–B*. Upon passing under *B*, 1 is changed from stop to caution and 3 is set to stop, these indications remaining while the car is in *B–C*. Upon passing under *C*, 1 goes to clear or neutral, 3 changing from stop to caution and 5 being set to stop. The fact that the contactors are located immediately at the signals instead of in advance of them, as is customary for single-track operation, makes the pilot light necessary, so that the motorman can be assured that he has counted into the block. West-bound cars operate in a similar manner.

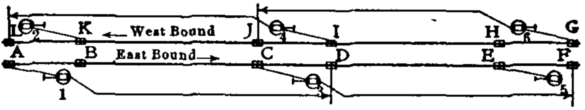


FIG. 10.

Scheme IV. (Fig. 10.) Trolley contactors. Light signals giving two indications. An east-bound car approaching *A* does not pass under it unless 1 is at neutral. If this is the case, the car in passing under *A* changes the signal from neutral to proceed, the signal remaining at this indication until the car passes under *B*, at which point the proceed indication is extinguished and the stop displayed. The car in proceeding eastward and approaching *C*, if it finds 3 at neutral, proceeds under *C* and if the signal changes from neutral to proceed, proceeds into *C–E*. The car in passing under *D* restores 1 to neutral, changing 3 from proceed to stop. West-bound movements are made in a similar manner.

Single-track Interurban Railway, Headway Not Less than 1 Hour, Speed 40 to 60 Miles per Hour. (Fig. 11.) Absolute blocking, semaphore signals, only one track circuit from siding to siding, continuous track circuits, following cars blocked from siding to siding, no track circuit preliminaries being used, intermediate signals replacing the preliminaries. Light signals may be used if desired. Cars on siding within clearance limits do not affect signals. An east-bound car passing *A* sets 1, 4 and 6 to stop and protects itself against both following and opposing movements. 3 does not go to stop immediately as the train passes it, due to the fact that the track circuit is not cut opposite the signal. It does go to stop, however, after the train reaches some point as *C* in advance of this signal. Proceeding eastward, as the car passes *D*, 4 goes to clear. West-bound cars are, however, blocked at 6. In passing the insulated joint at *F*, 1 and 3 behind the car change from stop to proceed, allowing a following car to enter the block recently occupied by the first car. 6 will also change to clear, due

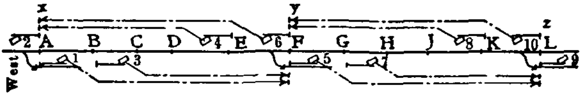


FIG. 11.

to the fact that until a following car enters the block at *A* the block is unoccupied. In case there had been a meeting point at siding *Y*, the east-bound car would, on approaching 5, have found it at stop, due to the west-bound car with which it had to meet, being between *L* and 5, because 5 is controlled to *L*. The purpose of 3 and 4, and 7 and 8 is to protect against the possibility of two opposing cars passing 1 and 6 simultaneously, both in this case being at clear. If this occurred, however, each car would receive the stop indication at 3 and 4 and in this way the advance signals 3 and 4 take the place of the additional track circuit usually used as a preliminary section.

This is the system which was installed on the W. B. & A. Ry. between Annapolis and Naval Academy Junction.

Single-track Interurban Railway, Headway 15 Minutes, Trains in Several Sections, Speed 40 to 60 Miles per Hour. (Fig. 12.) Absolute blocking, semaphore signals. following cars blocked

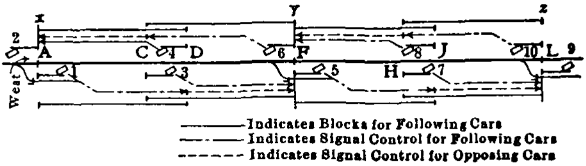


FIG. 12.

practically one-half the distance between sidings, no track circuit preliminary being used, intermediate signals replacing the preliminaries. Light signals may be used if desired. Cars on siding inside clearance limits do not affect the signals. This scheme is somewhat similar to that shown above, except that the intermediate signals have been moved farther into the block and arranged so that following cars can operate more closely than is possible with that arrangement. An east-bound car passing *A* sets 1, 4 and 6 to stop, thereby protecting itself against both following and opposing movements. Proceeding eastward, 3 goes to stop as the car passes *C*, 1 and 4 go to clear as the car passes *D*, thereby permitting a following car to pass 1 in the clear position, the distance between 3 and 4 being braking distance at maximum speed. If there be a meeting point at siding *Y*, the first east-bound car will find 5 at stop, due to the fact that the west-bound car that it is meeting, is between *L* and 5, 6 being at stop against the west-bound car. The east-bound car takes siding at *Y* which clears 6 for the west-bound car providing no following east-bound car is between *A* and 6. If a second east-bound car, however, followed the first into *A–F* after the first car passed *D*, 6 will remain at stop until the second east-bound car reaches siding *Y* and gets into clear, when 6 will clear for the west-bound movement. The two cars on siding *Y* now back out and the first proceeds eastward, 5 being clear. The second east-bound car, however, waits at 5 (which was put to stop when the first east-bound car passed it) until the first car passes *J*, when 5 will again clear. The second east-bound car may then proceed. 3 and 4, and 7 and 8 serve double purposes; they not only divide the blocks for following movements but take the place of a preliminary track section in the same manner as explained on page 759; *i.e.*, if two opposing cars passed 1 and 6 simultaneously, both receiving the clear indication, they will be stopped by 3 and 4, respectively, in which case one car would have to back up to the nearest siding. This scheme uses but one track circuit from siding to siding, continuous track circuits being used.

This system of signaling is commonly known as the T. D. B. system and is a most popular system of single-track signaling where high speeds are involved and continuous track circuits are employed. Either semaphore or light type signals may be employed in this system, the tendency being to favor 2-indication color light signals. For the last five years wherever this system is recommended it is based on the use of two track circuits. With average roadbed conditions where the ballast resistance would not be less than 4 ohms per 1000 feet, an end fed track circuit can be operated successfully up to approximately 10,000 ft. If one half of the distance between sidings would exceed 10,000 ft., it is recommended that a center-fed track circuit arrangement be installed.

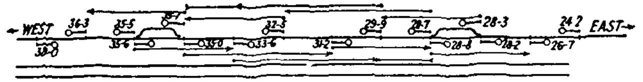


Fig. 13.

Heavy Electric Traction Single Track. The Chicago, Milwaukee & St. Paul Ry., throughout its 660 miles of electrification, has installed what is known as the A. P. B. system, a typical signal control limit diagram being shown as Fig. 13. The operation is as follows: Upon passing signal 35-0 an east-bound train sets signals 35-0, 32-5 and 29-9 to stop and 28-7 to caution. As the train passes signal 33-6 that signal is also set at stop. When the train passes signal 32-5 the aspect of signal 28-7 is changed from caution to stop, and when the rear end of the train has passed signal 32-5, that signal can clear up. As the train enters the track section between signals 31-2 and 29-9, signal 31-2 is set to stop. When the rear end of the train has passed signal 31-2, the control circuit for signal 35-0 is established so that this signal assumes the caution aspect. The train next passes signal 29-9 and upon shunting out the track relay for the section between signals 29-9 and 28-7 opens the circuit for the relay at signal 31-2. When the rear end of the train has passed signal 29-9 the circuit for signal 33-6 is established and this signal assumes the caution aspect. In this connection note that signal 33-6 indicated caution with an east-bound train between signals 29-9 and 28-7. Signal 33-6 will indicate stop with a west-bound train in the track section between signals 29-9 and 28-7 because a stick relay will be open when a west-bound train receives a proceed indication at signal 28-7 and the contacts of this stick relay will remain open with the west-bound train in the track section between signals 29-9 and 28-7. The control of signal 33-6 as described above allows a following east-bound train to pass signal 33-6 as soon as the rear end of the first east-bound train has passed signal 29-9. As the following east-bound train could leave the siding and pass signal 35-0 as soon as the rear end of the first train had passed signal 31-2, it is evident (on account of the short block between signals 35-0 and 33-6 and the long block between signals 31-2 and 28-8) that if both east-bound trains are moving at approximately the same speed, the second train might be delayed at signal 33-6 except for the control of signal 33-6 by a stick relay. When the first east-bound train has passed signal 29-9, this will indicate clear, provided the following train is not east of signal 35-0, but if the following train is approaching signal 33-6, then signal 29-9 will indicate stop. When the first east-bound train has passed signal 28-7 that signal will indicate caution if the following train has not entered the track section between signals 32-5 and 31-2, and if the following train has entered this track section then signal 28-7 will indicate stop. The control for the signals for west-bound movement can be understood from the above. It will be noted that the signals controlling movements into the passing siding are controlled by the track section which covers the siding plus the first track section beyond the siding. This is to prevent opposing trains attempting to use the main track at the siding at the same time. The clear indication of each signal is controlled from the indication of the next signal in advance in all cases. It is possible for opposing trains to pass signals 35-0 and 28-7 at the same time, but these trains would be stopped by signals 33-6 and 29-9. The inferior train would then have to back out of the block and allow the superior train to proceed. This condition would not occur unless one of the trains attempts to pass through the block when that train has orders to meet the opposing train at the siding. The signal controls are arranged so that if shifting movements are being made at one siding a train can enter the block from the siding at the opposite end of the block. This allows shifting movements to be made up to the last moment without delaying regular trains.

Signal Indications and Aspects.

(1) A.E.R.E.A. Standard use of the three fundamental indications. In all signaling the use of three fundamental indications: (a) Stop; (b) proceed-with-caution; (c) proceed.

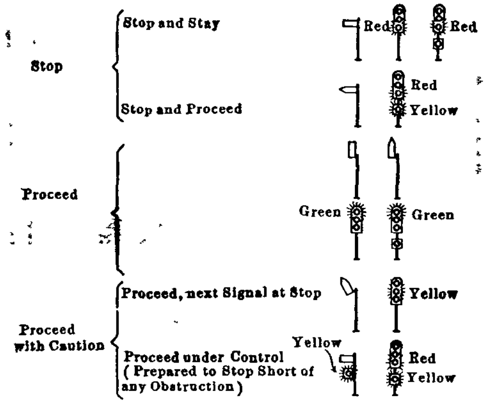


Fig. 14.—Aspects in three-position signaling. A.E.R.E.A. standard.

(2) A.E.R.E.A. Standard use of semaphore signals. Where sempahore signals are used they shall be so arranged as to indicate (indicating in) three positions, in the upper left-hand quadrant.

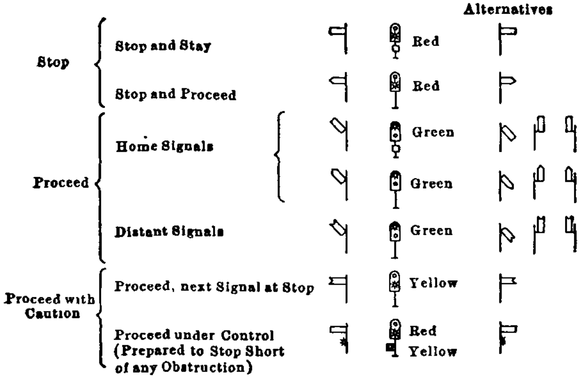


Fig. 15.—Aspects in two-position signalling. A.E.R.E.A., proposed practice.

(3) A.E.R.E.A. Standard. Aspects in three-position signaling (see Fig. 14).

(4) A.E.R.E.A. Miscellaneous Methods and Practices. Aspects in two-position signaling (see Fig. 15).

ASPECTS A	ASPECTS B	A-High B-Dwarf INDICATIONS	NAME
		1 Stop	Stop Signal
		2 Stop-Then proceed at slow speed with caution	Stop and Proceed Signal
		3 Proceed at slow speed prepared to stop	Slow Speed Signal
		4 Proceed with caution prepared to stop short of train or obstruction	Permissive Signal
		5 Proceed at slow speed with caution prepared to stop short of train or obstruction	Caution Slow Speed Signal
		6 Proceed at restricted speed	Clear Restricting Signal
		7 Approach next signal prepared to stop	Approach Signal
		8 Approach next signal at restricted speed	Approach Restricting Signal
		9 Proceed	Clear Signal
		10 Proceed at slow speed	Clear Slow Speed Signal
		11 Approach home signal with caution	Caution Signal
		12 Take siding	Take Siding Indicator

FIG. 16.—Aspects in position light signaling.

(5) Aspects in position light signaling. Fig. 16 shows the various indications with their respective interpretations in one of the most recent developments in modern signal practice. The position light signal is installed on the Philadelphia-Paoli and the Chestnut Hill branch of the Penna. R. R. and is growing in favor throughout the country.

Spectacle. Fig. 17 shows the design of spectacle for left-hand upper quadrant 90 deg. semaphore having 6 in. openings and 6½ in. lenses for 3-position semaphore signals and for use with 3 ft. 6 in. pointed blade, adopted as standard by the A.E.R.E.A., 1914.

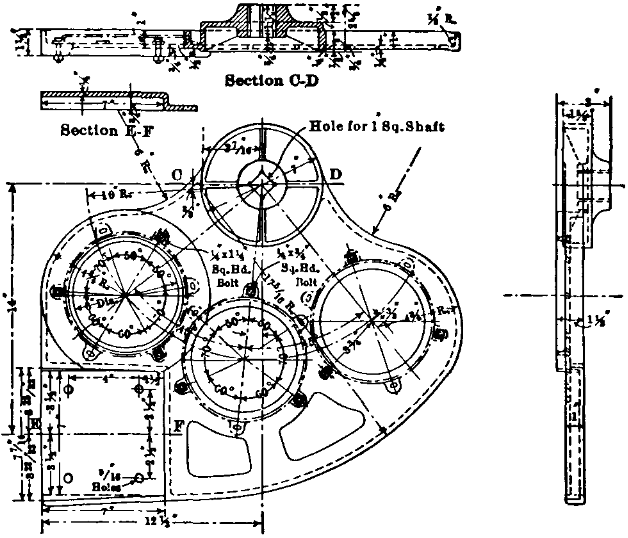


FIG. 17.—Spectacle design for three-position semaphore. A.E.R.E.A.

Clearances for Block Signal. Figs. 18 and 19 show A.E.R.E.A. standard clearance diagrams for semaphore signals, for use on electric roads where steam road equipment is operated.

Discernibleness of Light Signal in Sunlight. *There is a difference* of opinion as to what can be claimed for the range of the various types of light signals; this is largely due to the fact that there never has been adopted a standard constituting a basis of comparison. Manufacturers of daylight signals consider that there are two general classifications, long range and medium range, and it is generally considered that the long range signals provide a positive and distinct indication under the most unfavorable sunlight conditions at a distance of 3500 ft. While the medium range signals provide a positive indication at a distance ranging from 1500 to 2000 ft., the long range signals are employed where heavy trains

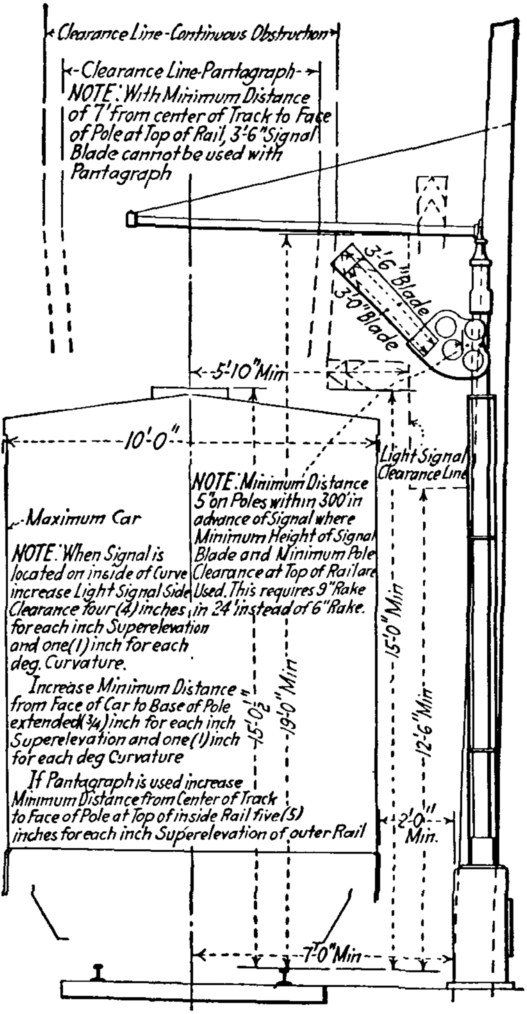


FIG. 18.—Standard signal clearance diagram.

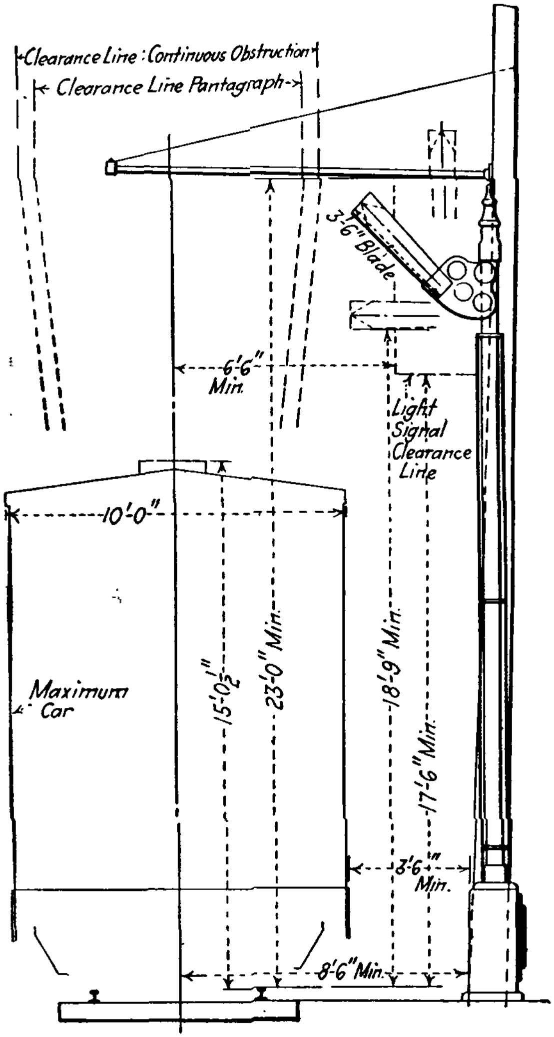


FIG. 19.—Standard signal clearance diagram.

are operated at high speeds, such as over the entire electrification of the Chicago, Milwaukee & St. Paul. The medium range signals have been found to be adequate for electric railways and interurban lines where comparatively light trains are involved, even though operated at high speeds.

Manually-operated Lamp Signal. (Fig. 20.) This is one of the simplest types of signals and is in very common use. The lamps are grouped to operate on trolley potential and but one additional signal wire extending through the block is used. A member of the crew of a train about to enter a block throws a switch, thereby displaying a light at the distant end of that block with a caution signal at the entering end. When the train reaches the end of the block, a member of the crew throws a switch at that leaving end, thereby removing the signals set by him at both ends of that block. If, however, two trains running in the same direction occupy one block at the same time, the signal should not

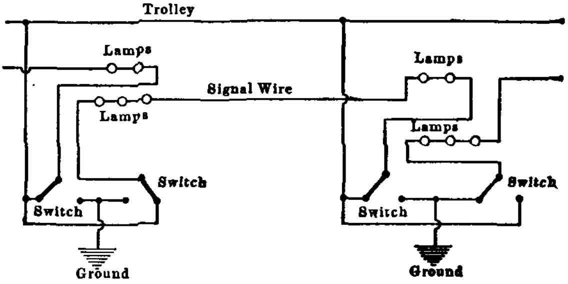


FIG. 20.—Manually-operated lamp signal.

be removed until the last train leaves the block. In some cases, one lamp only is used in the signal box at each end of the circuit while the remaining lamps of the series are installed at appropriate places along the section as an assurance to the motorman that his protection remains as set.

The signal should be arranged to prevent unauthorized manipulation. This is commonly done by padlocking the operating handles. The following scheme developed by the Rhode Island Co. for the protection of hand-operated signals also insures the removal of the reverse handle from the car while the signal is being set: Snap switches are used instead of knife switches. A round piece of metal having a square hole in one end is substituted for the ordinary rubber button on the snap switch. The switch is mounted so that this square hole is slightly above an opening in the bottom of the signal box. The motorman's reverse handle, having one end squared for the purpose, is used as a key to turn the switch.

Manual Block System. The manual block system is one in which trains are controlled by signals operated manually, upon information received by telegraph, telephone or other means of communication. It consists of a series of block sections with a block control station at each end of each section, at which there are placed signals to be used by the operator to control the trains entering and using said block. In the operation of this system, information concerning the condition of a given block is obtained by communication between the operators at the ends of said block, and trains are instructed as to movement by means of the signals. The safety of operation under the manual block system depends upon the accuracy with which orders are received and transmitted, there being no check, either electrical or mechanical, to prevent the display of a wrong signal.

Controlled Manual Block System. The controlled manual block system is the same as the manual block system just described except that the signals at the ends of each block are electrically interconnected in such manner that cooperation of the signalmen at both ends of the block is required to display a clear signal. It is a great improvement over the manual block signal, but still does not prevent two operators from making simultaneous error. This is overcome by the continuous track circuit, so arranged that a train having entered a block will automatically put the proper signals to stop and hold them there until the train is entirely out of the section, regardless of any attempt on the part of the operators to clear them. Its use on many electric lines is prohibited on account of the large expense involved in maintaining operators at such frequent intervals.

Automatic Block System. The automatic block system is a system in which the signals govern the entrance to each block and are automatically controlled by the train as they proceed, in such a manner that head-on and rear-end protection is afforded. A number of forms of the automatic block systems are being used, their essential differences appearing in the manner in which the control of the signals by the train is effected. These different forms of automatic systems include the following methods of control: (1) Trolley or auxiliary rail contacts, which are operated by passage of trolley or third rail shoe and thus operate the signal circuits when a train passes; (2) continuous track circuit, based on a fundamental principle of a train controlling, at all times, the signals governing the specific portion of the track occupied, and utilizing the track rails as a signal circuit; (3) short track circuit, which utilizes short insulated sections of the track at the ends of the block for setting and restoring signals.

Trolley Operated Signals. Trolley operated signals are generally operated by electrical contact through the current collector of the car or by electrical contact through a switch which is opened or closed mechanically by the current collector in passing.

Contact Type Trolley Operated Signal. Fig. 21 shows the relation of the essential parts of a typical trolley contact signal (the Nachod). The mechanism of the signal itself may be divided into three parts: (1) The indicating, consisting of the lamps, color lenses and opaque color disks; (2) the intermediate, represented by the relay, which converts the impulses of current caused by the passage of the car into signal indications; (3) the actuating, consisting of the overhead trolley contact switches, which by the passage of the car determine the setting and the clearing of the signals. The trolley switch consists of a light angle-iron frame supporting at its ends through insulating blocks, two flexible inclined contact strips, the trolley wire being withdrawn during the length of the contact strips, which are formed to receive the wheel without shock. One strip is connected to trolley, the other to ground through the relay. The wheel, being a metallic conductor, bridges them and sets the signal whether the car is taking power or not at the instant of passing the switch. One signal wire in addition to the trolley wire connects the apparatus at the ends of the block. Fig. 21 shows the parts in position for no train in the block, in which case each end of the signal wire is grounded through a red lamp *R*. With no current in the coils. the armatures drop. The first entering train sends proper current through magnet *A* in the relay at the entering end, operating a two-way revolving switch, which transfers that end of the signal wire to trolley through a white light *W* and a magnet *D*. This light and the red one at the

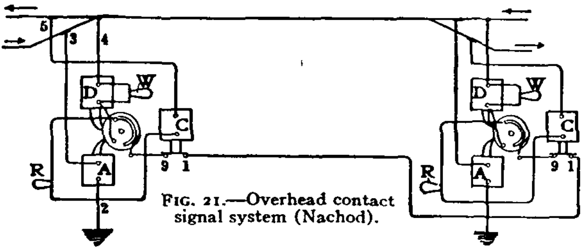


FIG. 21.—Overhead contact signal system (Nachod).

other end now burn in series through the signal wire. Successive following trains turn the revolving switch so that the contacts overlap further, but make no change in the electrical circuit. Each leaving train energizes magnet *C* to break temporarily the signal circuit at 9–1, permitting magnet *D* in the first relay to drop its armature and revolve the switch in the reverse direction. When the same number of impulses has been made on magnet *C* as on *A*, that is, when all the trains that have entered the block have left it, the signals are cleared and the connections are as shown. The color disks are brought to an indicating position by magnet *D* and one in shunt with *R*, not shown. A no-voltage magnet, not shown, is interlocked with magnet *D* to prevent a motion of the armature of the latter should the power fail with trains on the block. This is a signal of the absolute, permissive type. To illustrate the permissive feature, suppose a train has entered the block and set the signals, and before it leaves, a following train arrives at the end of the block showing white The motorman will understand by this that there is at least one train ahead going in the same direction. As his train runs under the contact he notices the flash of the white light that his train causes. This is an indication that his train is counted in or registered in the signal relay; and so on for a number of trains following each other, each receiving its signal that there are other trains ahead of it and that it has counted in. When the first train arrives at the end of the block, in running under the contact switch, it causes a blinking of the

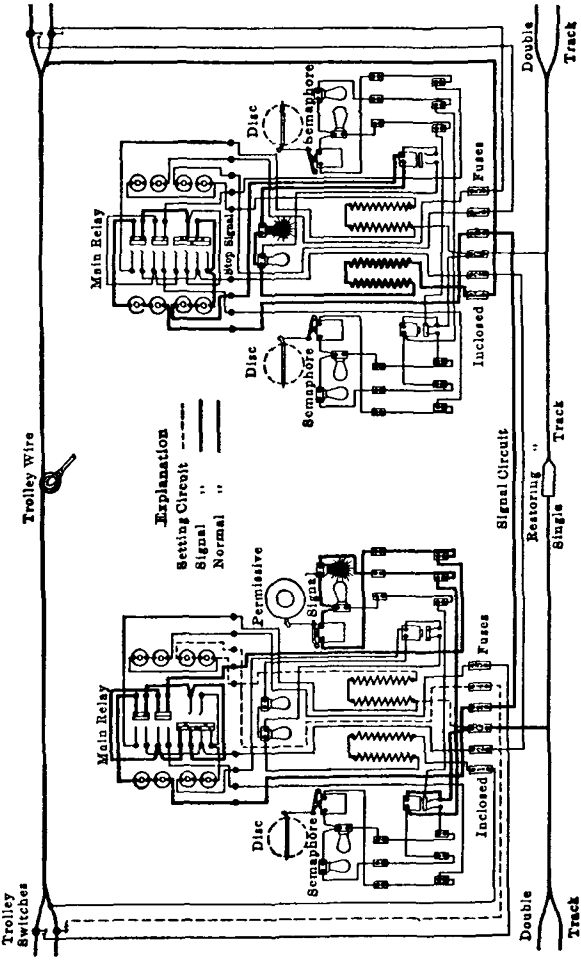


FIG. 22.—Overhead contact signal system (U.S.).

light, but the signals are set as before. They so remain until the last train clears them, leaving the block ready for trains in either direction. If a train should enter the block from one end and back out by the other track at the same end, using the single track merely as a Y or cross-over, the signals will be cleared when the train leaves the block. This makes a very flexible operation, as in the complicated case where a number of trains enter the block from one end, and are continuously entering and leaving, some leaving by one end and some by the other, but the signals protect all the trains in the block and are cleared only when they all leave it. The signal may also be operated as an absolute block system by leaving the hand-clearing switch open, which is equivalent to opening the line wire. For instance, the motorman of a work train, entering the block closes the block at both ends by leaving the hand-clearing switch open. Then no permissive signals may be obtained from either end. To leave the signals for normal operation again, he must close the switch when leaving the block. In case of failure of line voltage, the signals reappear on its resumption with the same indications.

Fig. 22 shows the circuits of another typical trolley contact signal (the United States).

The right-hand front magnet is the setting magnet which rotates the registering wheel one step for each energization. The left-hand front magnet is the restoring magnet which at each energization rotates the registering wheel one step in the direction opposite to the above. The front or main contact bar is in its right-hand position when the block is clear, but is thrown to its left-hand position when the first car is registered into the block on the registering wheel and remains in the left-hand position until the last car is registered out. The middle contact bar or alternating switch is operated by a star cam on the registering wheel so that it stands in its right-hand position when each odd car is registered in, and in its left-hand position when each even car is registered in. When it is in its right-hand position, it causes the right-hand green and white semaphore to be displayed, and when it is in its left-hand position it causes the left-hand green and white semaphore to be displayed so that the proceed indication will be given only when the registering wheel operates. The rear contact bar, normally in its right-hand position, is momentarily thrown to its left-hand position when the rear right-hand magnet is energized, but as soon as the latter is de-energized falls to its normal position. This magnet is energized by the restoring trolley contactor.

The trolley wheel of a car entering an unoccupied block from the left strikes the setting contactor, thus completing a circuit from trolley wire, through fuse 4, through setting (right-hand lower) magnet, resistance and fuse *G* to ground. This moves the registering wheel one notch and moves the main (front) contact bar to the left-hand position. As soon as the trolley passes beyond the contactor the circuit previously made is opened and the armature of the setting magnet falls back by gravity into its normal position. A circuit is then completed from the trolley wire at the right-hand end of the block, through the wire indicated by the heavy line, fuse 1 in the right-hand signal, resistance, red lamp, pick-up magnet, front left-hand magnet, the two right-hand contacts and main contact bar, the contacts and the rear contact bar, rear left-hand locking magnet, fuse 3, signal circuit line wire, fuse 3 in the left-hand signal, rear left-hand locking magnet in the left-hand signal, rear contact bar, upper contacts and main contact bar, around through contacts and middle contact bar, right-hand permissive semaphore magnet, green light, pick-up magnet, thence to ground. If a second car passes over the contactor, setting magnet will be energized in the manner above outlined for the first car and the registering wheel will be rotated one more notch, the middle contact bar or alternating switch will be thrown to the left-hand position. The circuit through the signal at the right-hand end of the block will then be completed as above outlined, but the circuit through the left-hand signal will be completed as follows: From fuse 3, through rear left-hand locking magnet, rear contact bar, upper left-hand contacts, main contact bar, left-hand contacts and middle contact bar, left-hand semaphore magnet, left-hand green light, pick-up magnet, thence to ground. This will cause the opposite green semaphore disk to show. When the trolley wheel passes the restoring contactor at the right-hand end of the block, a circuit is completed through fuse 5, rear right-hand magnet, resistance, thence to ground. This moves the rear contact bar to its left-hand position, opening the signal circuit previously described and completing a circuit across the left-hand pair of contacts. This action completes the restoring circuit as follows: From the trolley wire at the left-hand end of the block, through fuse 1, resistance, red lamp, pick-up magnet, restoring magnet, lower left-hand contacts and main contact bar, fuse 2, restoring circuit line wire, fuse 2 in the right-hand signal, left-hand pair of contacts and rear contact bar, thence to ground. This energizes the restoring magnet in the left-hand signal and rotates the registering wheel one step in the direction opposite to that in which it was rotated by the setting magnet. As the trolley leaves the restoring contactor the circuit just described will be opened, rear right-hand magnet in the right-hand signal will be de-energized and the rear contact bar will fall back to the right-hand pair of contacts. This places the signal circuit in its normal condition. When the last car is counted out of the block, the main contact bar in the left-hand signal is thrown to its right-hand position. Both signals are then neutral. If a car enters the block from the right-hand end, the circuits will be completed in the same manner except they will be fed in the opposite direction, that is, from the left-hand end of the block.

Single-rail Track Circuit Alternating Current Signal System. Fig. 23 shows the relation of the essential parts of a track circuit alternating-current signal system in which one rail (the "block rail") is insulated in sections and is used to carry only current of the signal system. The other rail is used to conduct the train propulsion direct current and the current of the signal system. Alternating current whose potential is reduced from that of the signal mains by transformers at the block is used to operate the signals. A non-inductive resistance is placed in series with the secondary of the transformer which supplies the current for the track signal circuit. A non-inductive resistance is likewise placed in series with the signal relay and an inductance is shunted across the signal relay. The purposes of these resistances and the inductance are to reduce to a negligible value the direct current which, due to the drop in potential of the train propulsion current in the block, flows in the relay and the track transformer, also to limit the flow of alternating current through the track transformer when the rails are bridged by wheels and axles of a train in the block. The magnetic circuits

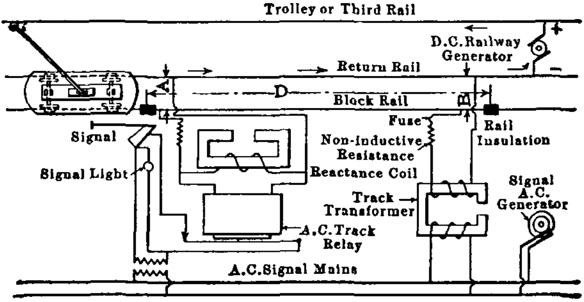


FIG. 23.—Single-rail alternating current block signal.

of inductance and transformer each include an air gap to diminish the magnetic effect of the small direct current which passes through their windings. Wheels and axles of a train in the block shunt the signal current, thus causing the relay to release its armature and set the signals.

Double-rail Track Circuit Alternating Current Signal System. Fig. 24 shows the relation of the essential parts of a track circuit

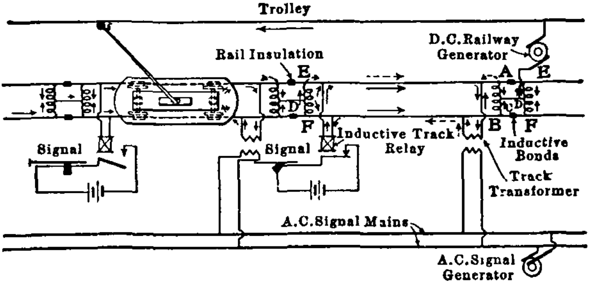


FIG. 24.—Double rail alternating current block signal.

alternating current signal system in which both rails are used in common by the signal alternating current and the train propulsion direct current. This common use is made possible by the inductive bonds which, while offering little resistance to the train propulsion current, maintain a considerable alternating current potential difference between the rails when there is no train in the block supplied by the track transformer. While this considerable alternating current potential difference continues the armature of the relay holds the signals at the normal indication, but when, as by the bridging action of the wheels and axles of a train in the block, this alternating current potential is greatly reduced, the armature of the relay is released and the signals are set. The inductive bond consists of a single winding of heavy conductor on a laminated iron core and having a tap brought out at its middle point. (The magnetic circuit for use with direct current for propulsion includes an air gap.) The bond is connected across the rails at one end of an insulated section of track and the middle tap is connected to the middle tap of a similar bond connected across the rails at the adjacent end of the adjoining insulated track section. To limit the flow of current supplied by it, the track transformer is provided with an adjustable leakage path between its primary and secondary windings.

This system is also used where alternating current is used for train propulsion. For this service the frequency of the current in the signal circuit is greater than that of the train propulsion current. The impedance bonds have no air gaps. They offer little impedance to the train propulsion current, but they offer sufficient impedance to the signal circuit current. The relay is constructed to operate only with current of the higher frequency.

Short Track Circuit Signal System. In this system only short insulated sections of track at the ends of the block are used in the signal operation. There have been a few signal installations employing the short track circuit principle, or what is more commonly known as the "trap circuit" arrangement, but this has been resorted to only in such cases as where the roadbed was such as not to permit the installation of continuous track circuits. While it represents a possibility, it has a small place in modern signaling. The following outline is based on the Kinsman system. Special provision for signal operation is made between adjacent rails of but one side of the electric railway track, thus the other rail is left continuous as if it were to serve only the needs of train propulsion. The track circuit includes two adjacent insulated sections, each two rail-lengths long. The sections are insulated by three insulated joints. The section farther from the center of the block controls the setting of the signals, while the second section controls the release of the signals. The return circuit of the train propulsion current is carried around these insulated sections by an auxiliary conductor, either rail or cable. The relays are operated by current from a primary battery. When the block is clear the normal position of the signal arm is vertical. The train in entering the block first passes over the setting section and operates the setting relay. This cuts out a release relay and makes both insulated sections a setting section. If a train is leaving a block it first passes over the release section, cutting the setting relay out of circuit and making both sections a release section. A step-by-step controller at each signal registers the movement of the trains into and out of the block and controls the circuit operating the signal. The controller switch is not thrown back to its normal position until the last train leaves the block.

The position of the signals is controlled by the position of the controller switch. When the block is unoccupied this switch is at normal position and the signals at both ends are in vertical position. As a train enters a block the setting relay operates the controller at the entering end and the controller switches open. This breaks the circuit, de-energizing the signal at the distant end of the block, allowing it to fall by gravity to the horizontal or stop position. As a signal assumes the stop position it opens the circuit. When an entering train leaves the insulated section, the signal at the entering end is de-energized and falls by gravity to the 45-deg. or caution position and is held in this position by a slot magnet which is operated by the switch on the controller. As each succeeding train enters the block it passes onto the insulated section and the signal moves from the 45-deg. position to the vertical position; then after the train has passed over the insulated section the signal falls back to the 45-deg. position, and the signal at opposite end of block goes to horizontal position. As each train passes out of the block the release relay operates and in turn actuates the controller at the entering end of the block one step back toward its normal position. As the last train passes out the controller switch is operated which completes the circuit through the signal at both ends of the block, causing both to operate to the vertical position, which indicates that all trains are clear of the block.

Insulating Joints. Where, as at the end of a block, it is necessary to insulate adjacent rail ends from each other, a piece of insulating fiber is placed between the rail ends, insulating bushings are placed around the bolts and insulating fiber is clamped between the rail and the joint plates, the latter having been planed down or specially made for the purpose.

Effect of Zinc-treated Ties on Two-rail Track Circuits. The following is the consensus of opinion obtained in letter replies. It was given by the Committee on Signals and Interlocking, 1913, A.R.E.A., and accepted as information by that association: (1) Track circuits a mile in length are rendered inoperative by the extensive use of zinc-treated ties. (2) Track circuits 2000 ft. in length may be operated successfully, even with 50 per cent or more of ties so treated. (3) 10 per cent to 15 per cent renewals per year will not materially affect such length circuits. (4) Where renewals are made of 15 or 20 adjacent ties, the leakage is much greater than where they are made singly at uniform distances, *i.e.*, with 15 per cent renewals (every sixth or seventh tie). (5) While the surface salts are present, more leakage occurs during wet weather than with untreated ties, as these wet salts form a better conductor than ordinary wet wood. (6) In dry hot weather, the salts are drawn to the surface and constitute a more or less perfect conductor. (7) After a period varying from 3 months to a year, these salts disappear and subsequently no interference is noticeable.

Interlocking. Interlocking plants are assemblages of switches, switch-operating and switch-locking devices, and signals so interconnected that their movements must succeed one another in a predetermined order. They are used at crossings or junctions of one line with another, at drawbridges, at cross-overs, or in yards where the number of switches or the frequency of their movement is so great that hand operation would not be sufficiently rapid nor safe. At railroad crossings or drawbridges, signals are frequently interlocked with derailing devices, this protection being required by the laws of several states where trains are permitted to pass over without stopping. Interlocking plants are either mechanical or power-operated. In those of the former type, pipes or wires connect the switches, locks, signals, or other operated units with the levers that operate them. In the latter type, pneumatic cylinders or electric motors are connected to suitable lock and switch movements or signal mechanisms. Electric circuits controlled by the levers in the interlocking machine regulate the operation of these switch and signal mechanisms.

In all interlocking machines the desired sequence of operations is secured by means of the dogs, bars, or tappets which constitute the mechanical locking. These parts are so interrelated that if any lever in the machine is reversed (that is, moved from its normal position), the act of the signalman in unlatching this lever will cause parts of the locking so to operate that no other lever in the frame can be moved which would allow a train movement conflicting with the train movement controlled by the first lever. The mechanical locking between the levers of the machine also provides for the movement of the levers in proper sequence when the route for a train is being set up, by assuring that the switches must first be properly set and must then be locked in the proper position before the signal governing the route can be cleared.

In power-operated interlocking machines where the only physical connections between the operated units and the levers in the machine consists of insulated conductors which carry small currents, there is no direct assurance, when a switch lever is operated, that the switch itself has moved. To insure that it is moved properly and will be locked in a position corresponding to that of the lever, all power-interlocking systems provide for what is called a return indication. When the lever in the interlocking machine is moved to cause a switch or signal to be thrown, it cannot at first be moved through its complete stroke nor through sufficient length of stroke to release the mechanical locking between itself and conflicting levers. The lever can only be moved far enough to close the circuit which will cause the desired motion of the switch that is to be operated. The switch mechanism itself must complete the throwing of the switch, must completely lock it in the proper position, and must then close contacts that will cause current to flow back to the lever in the tower and energize a magnet to release a dog, which has heretofore limited the motion of the lever. After the dog has been released the operator is permitted to complete the stroke of the lever, which will in turn release the mechanical locking between the switch lever and others in the frame, allowing movement of the latter to be made in proper sequence.

Long bars of steel, called detector bars, are held by clips along the outside of the head of the rail at switches to prevent operation of the switch while a train is passing. They are of a length greater than the maximum distances between any of the adjacent wheels of a train, and are so mounted that they must move upward above the level of the top of the rail before the switch may be unlocked. They generally are mechanically connected to, and operated simultaneously with, the lock plungers which pass through holes or notches in the lock rods of switches to lock them in proper position. This method of protection may prove unsatisfactory owing to the increasing width of rail heads, and especially on electric roads, the relatively narrow wheel treads of which may fail to engage the top of the detector bar while the vehicle is passing over it. Electric track circuits controlling electric locks mounted on the levers of the interlocking machine itself, are becoming quite generally used as a substitute for detector bars in the prevention of switch movement while trains are passing.

Dispatchers' Signal Systems. These are systems in which provision is made whereby a dispatcher may set signals at desired points along the track. Telephone communication between train crew and dispatcher is also generally provided for. The main point of difference between dispatchers' signal systems which set fixed signals is the method whereby the desired fixed signal is selected.

Impulse Systems. In the Blake signal system several signals are connected to the same circuit and each signal contains a pendulum of a length differing from the lengths of all other pendulums in the signals on that circuit. In the dispatcher's office there is a pendulum of the same length as each of the pendulums in the signals. When the dispatcher sets one of his pendulums vibrating it opens and closes an electric circuit, thus sending impulses of current out on the signal circuit. These impulses of current pass through electromagnets in the signals and, in a few seconds, the pendulum of the length of the one started by the dispatcher is made to vibrate by the electromagnet to such an extent that it mechanically trips the release which sets the signal. As the signal sets it closes a sounder circuit to the dispatcher's office, thus giving indication that the signal has been set. There are other signal systems in which a spring or weight-driven selector in the dispatcher's office sends out impulses which will operate a desired signal. The selector signals are generally reset manually at the signal or by cooperation of the dispatcher and someone at the signal.

Crossing Protection. Automatic highway crossing protection may be afforded by an audible or a visible signal, or by a combination of these, to warn highway traffic. Which should be used at a given crossing depends upon the traffic conditions on both the railway and the highway, also upon the particular surroundings of the situation. The audible signal is given by either a gong having a frequency of about 200 strokes per minute or by a horn. The visible signal may be given in the daytime by a semaphore or swinging arm which is painted so that it is conspicuous against its background and at night by electric light or oil lamp—the latter being arranged with screens operating to expose the proper danger signal. To avoid unnecessary stops or slowdowns at a crossing, a signal operating with the highway protection device is sometimes placed to indicate to the motorman whether or not the crossing device has operated. Such a signal is commonly arranged to indicate "danger" normally and "clear" when the crossing device is operating. Crossing protection signals may be operated by any of the means used for ordinary block signals with the exception that in any particular installation the means employed must be such as not to interfere with the proper working of the block signal system.

The three-aspect "automatic flagman" of the Union Switch and Signal Co. is shown by Fig. 25. This signal indicates the approach of a train by swinging a red banner on which appears the word "*Stop*," and displaying a red light attached to the banner. When no train is approaching the banner is held to one side as shown by the dotted lines, between two screens upon which is painted "*Look and Listen*," and the lamp suspended from the banner is not lighted. If the circuit through the holding magnets is broken, but the apparatus is otherwise in good condition, the banner will swing irrespective of the approach of a train. If the circuit is broken through the operating magnet but not through the holding magnet the banner will be retained in its extreme position between the screens until a train approaches, when it will be released and ultimately assume a vertical position with the banner stationary but fully displayed. If the current is totally cut off the mechanism or if the operating parts become disconnected, the banner will also assume the vertical position and be fully displayed. The operating mechanism is enclosed in a waterproof case and consists essentially of operating magnets for driving the swinging banner and holding magnets for retaining the banner at one end of its arc of travel. A circuit controller provides for the selection between the pairs of operating magnets. The signal is designed to operate normally at 10 volts D.C. It requires an average of 0.4 amp. for swinging the banner, and 0.4 amp. for the 5-watt 12-volt lamp, making an average of 0.8 amp. drawn from the battery while the banner is swinging. The holding magnets are of 1000 ohms resistance and require normally 10 mil-amp. when the flagman is latched in the clear position. The control is so arranged that it is unnecessary to break the operating circuit through relay contacts; the only current passing through these contacts is that of 10 mil-amp. required for holding the mechanism in its latched position. The signal can be equipped with a bell and with either an oscillating or fixed lamp. If desired, the fixed lamp can be arranged to burn oil and to give flashes of light as the banner swings to and fro, as shown at *B*, Fig. 25.

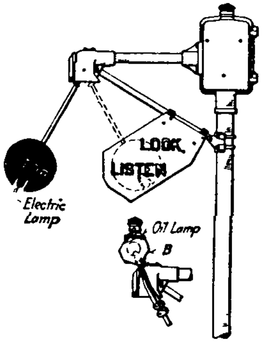


FIG. 25.

The signal as described above is controlled by the continuous track circuit principle, which may be considered as the most reliable, where it is feasible to use it. Several schemes using other methods of control are described below.

Trolley Switch Contact. A setting switch is placed in the trolley wire at the approach to the crossing and a restoring switch is placed in the trolley wire at the crossing. The trolley wheel of a car passing the setting switch closes a momentary contact to energize the setting magnet of a relay in the crossing mechanism, causes its armature to be attracted and mechanically latched in position. This operation closes a permanent feed from trolley across contact points on the relay to the bell and through resistance to ground. Upon reaching the crossing, the trolley wheel on the car closes a momentary contact in the restoring switch, which, being connected to the restoring magnet of the relay, causes its armature to be attracted; which operation unlocks the mechanical latch above mentioned, allowing the armature of the setting magnet to fall by gravity and opening the trolley feed, thus restoring the signal to the inoperative condition. The circuits may be arranged to illuminate a danger sign at the crossing or for other lamps or light arrangement such as the use of red lights on the highway each side of the crossing or such other arrangement as will best suit the particular case.

Third Rail Contact. (Sedwick.) A section of third rail is laid on the side of the track opposite to that of the regular third rail which supplies the train propulsion current. When a car passes this point, the contact shoes of the car, being all four in parallel, make contact between the regular third rail and the above-noted insulated section on the other side of the track. There is a circuit from the insulated section through a relay and thence to ground. The relay has two sets of coils. When a car is approaching the crossing it passes over the section of rail which makes the contact so as to energize one pair of coils in the relay. The armature of the relay is then pulled over so as to make contact and establish a circuit from the regular third rail through a bank of five lamps to ground. An alarm bell is connected in shunt with one of these lamps. The lamps are lighted and the bell is rung until the car passes a second section of third rail which is beyond the crossing. In passing this section a circuit is established through the other pair of coils on the relay, which results in opening the alarm bell circuit and putting the signal out of operation until again energized by a car approaching the crossing.

Plunger Contactor. (Sauer and Johnson.) In this signal, the car wheels, in passing, depress the plunger of a track instrument, located close beside the rail. This plunger is held up by a stiff spring and can be depressed only by the car wheels. The car wheels, in depressing this plunger, operate a lever which pushes an armature up in a solenoid. This armature, when at the upper limit of its travel, makes contact between two fingers, thus completing the circuit from the trolley or third rail through the solenoid coil. The coil thus being energized, holds the plunger up of itself. The same circuit passes through the alarm bell and bank of lamps. The alarm bell circuit is broken by the passing of the car wheels over a similar track instrument on the other side of the crossing, which momentarily breaks the circuit and causes the solenoid of the track instrument on the other side of the crossing to let go.

Oscillator. (Protective Signal Manufacturing Co.) The oscillator is a contacting device enclosed in a waterproof iron casing, so arranged that it may be fastened mechanically, in a rigid way, to the base of the running rail. It consists of a spring arm held at one end and carrying at the other end a magnetic coil in series with the line circuit connecting the oscillator and the signal mechanism at the crossing. Normally this contact is broken, but the movement or swing of the arm by vibration causes it to open the circuit momentarily at its outer end and thus start the operation of the crossing bell. The bell is kept ringing during the interval between the time that the train passes over the oscillator and the time when it reaches the crossing by means of a magnetic timer consisting of a ratchet wheel revolved by means of a magnetically operated pawl and retent, connected in series with the bell circuit. This is usually set to operate for 30 seconds after the oscillator has closed the line circuit.

Magneto Mechanical. (Hoeschen Manufacturing Co.) This apparatus operates independently of an external supply of electric current. One end of a lever pivoted near a rail extends under the rail. The other end carries the armature of a pair of induction coils having permanent magnets for cores. The passage of a train over the outer end of this lever depresses it, thus lifting the armature from the magnets. This induces a momentary electromotive force in the induction coils and the resulting current is conducted through line wires to the release magnets which control the bell motor at the crossing. The bell motor consists of a gear movement of three wheels used in connection with three motor springs. These springs are secured to the main driving shaft of the motor and are wound by a rod connected directly with a lever resting against the underside of the rail at a point opposite the motor. When the rail at that point is depressed by the wheel of each car of a passing train, a reciprocating motion of the winding rod (motor lever) is obtained, thus winding the springs and restoring the releasing lever which controls the motor. The connecting rods, extending from the winding rod to the pawl plates on which are carried the dogs for turning the ratchet wheel are arranged to wind the ratchet wheel on both the upward and downward stroke of the winding rod. The passing of one car will wind the ratchet more than 4 in. The motor gearing is so arranged that less than this amount of winding is needed to ring the bell for the passing of two following cars. The motor stores sufficient energy to ring the bell for about 100 cars before rewinding is necessary.

Automatic Train Control

The necessity for an automatic device to stop a train increases as train speed is increased and as headway is shortened, becoming greatest where the possibility of failure to promptly see and obey a danger signal is greatest. Automatic train stops are generally so arranged that the train may "key by" in an emergency. That is, a member of the train crew may arrest the operation of the stop *in order to let the train pass into the block protected. Aside from* the location of regular stopping places, local conditions of traffic and right of way and the type and method of signaling in use, the climatic conditions under which the train stop is to operate must be given serious consideration in deciding upon the proper train stop for a required service.

The Joint Committee on Automatic Train Stops, American Railway Association, Nov., 1913, called attention to the necessity of studying an automatic train control system with a view to avoiding the introduction of new elements of danger which might be greater than those which it is the purpose of the installation to overcome. This was because no automatic train control device, so far as was known, could be universally applied without adding elements of danger in train operation. The following requisites of construction, installation and operation of an automatic train controlling system are from the report of that committee:

Failure of any essential part will cause the application of the brakes.

Proper operation under all conditions of speed, weather, wear, oscillation and shock.

A train traveling at a speed conforming to restrictions may pass a trip in the tripping condition without the brakes being applied.

Prevention of the release of brakes after application has been made until the train has been stopped or its speed reduced to a predetermined rate.

Stop or speed control accomplished within a predetermined distance.

Operation without interference with the application of the brakes by the motorman's valve and without a reduction in the efficiency of the brake system.

Operation without interference with fixed signals.

In 1924 automatic train control is being actually installed by the steam railroads and there is no question but that in the very near future large installations of automatic train control will be in service. The fact that the Interstate Commerce Commission has ordered 49 of the principal railroads to install this system of control, serves to show the importance which is being placed upon this kind of protection for traffic.

Union Switch and Signal Automatic Train Control from Continuous Track Circuit. This system provides three fixed speed limits—*high, medium and low, corresponding closely to the three indications* "proceed," "caution" and "stop" of the automatic block signaling system. The cab equipment may include a speed limit indicator showing the permissible speed limit established by the train control system, and a speedometer to show the speed at which the train is running. The control of the speed is by means of a centrifugal speed governor, connected by a propellor shaft type of drive to an axle of the locomotive. The control of the brakes by the governor is purely pneumatic, by means of the same type of valves as are used in existing air brake equipment. The governor is controlled by two magnets, which are, in turn, controlled by a three-position a-c. relay on the locomotive, designated the "train control" relay. This relay is of the same type and functions in the same manner as the track relay of the block signaling system. The system involves no clearance difficulties, as no apparatus is required on the trackway and the engine-carried apparatus is within the equipment clearance lines. Brakes are applied if the train passes a signal indicating caution. However, there is also a valve in the engine cab, which can be operated by the engineman to prevent the train speed being unduly restricted by a brake application, provided he operates it *just before* passing a signal indicating caution; thus, at each caution signal the engineman is given the opportunity to show that he is awake and alert, and, by so doing, retains control of the train so long as he functions properly. If the engineman fails to acknowledge a caution signal by the operation of the valve, the train is automatically brought to a stop. Inasmuch as the information is conveyed to the engineman at the caution signal, a service brake application only is required and given. This system does not make use of automatic emergency brake applications under any condition, but it does not prevent the engineman himself making such an application in the regular manner. The use of the closed circuit principle gives a foundation for obtaining maximum safety. The section of track which governs the train includes the track rails directly in advance of the train, which are used to conduct the signaling current which operates the train control apparatus located on the engine.

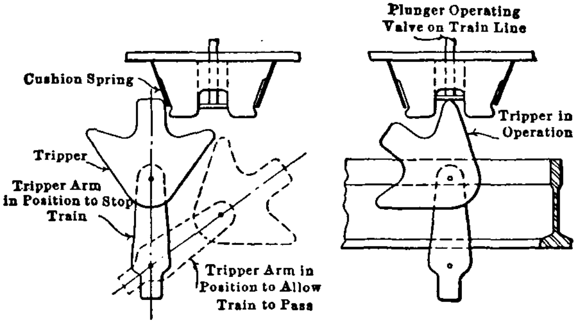


FIG. 26.—Automatic train stop, Pennsylvania tunnel.

Union Switch and Signal Automatic Train Stop. Fig. 26 shows the type of automatic train stop used on the Pennsylvania Tunnel & Terminal installation in New York. This stop consists of a tripping arrangement on the roadway and a valve under the car which can be operated by the tripper to set the brakes. The tripper arm carrying the tripper, held in position by two helical springs, is mounted on a rocker arm and is operated in connection with the signal so that when the latter is at "clear" the tripper arm is depressed, by a solenoid or compressed air, against the ties and thus holds the tripper out of the path of the train mechanism. When, however, the signal is at "danger" the tripper arm takes the vertical position and upon being struck by the cushion spring on the train mechanism the tripper revolves about its axis on the tripper arm and operates the valve in the air brake system as indicated in the figure. The tripper is of light and flexible construction in order that it may not be broken by fast trains. Where weather or other conditions prevent satisfactory operation under the train the stop is placed overhead.

Suggested Clearances for Automatic Train Stop. Fig. 27 shows the limiting clearances for automatic train stop as suggested by the 1911 Committee on Heavy Electric Traction, A.E.R.E.A.

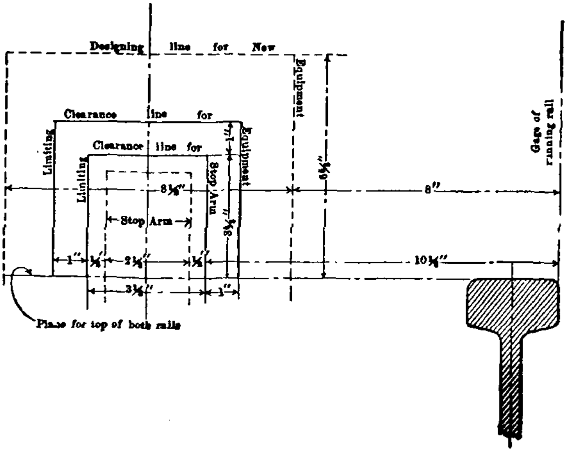


Fig. 27.—Clearances for automatic train stops.

Telephone

General Methods of Dispatching by Telephone. Where train dispatching is done by telephone, the telephone is either contained in a booth beside the track or it is carried on the train. In the latter case, connection with the line is made to terminals situated beside the track or by the connections provided for the cab signal. Where the telephone is the only method of signaling, the motorman or conductor of a train calls the dispatcher from whom he then receives his running orders. Where a dispatcher's signal system (see p. 777) in addition to the telephone is used, the dispatcher sets a signal at which the train stops and the crew communicates with the dispatcher. The motorman or conductor may receive the order and repeat it to the dispatcher directly or the motorman or conductor may receive the order, write it, meanwhile making a carbon copy which he gives to the other who reads it back to the dispatcher.

The latter method insures the least chance for error in receiving the order.

Telephone Dispatching in City Operation. The following is a general outline of the system of telephone dispatching used in Rochester, N. Y., where, due to congestion, traffic handling is unusually complex: The chief dispatcher and several assistants are stationed at a special dispatching telephone switchboard located in the railway company's office building. This switchboard is connected by means of wires leased from the telephone company to telephone sets located at every line terminal and at main intersections. Telephone sets are also located at canal lift bridges, grade crossings and other points where delays are liable to occur. Each motorman is required to report immediately upon his arrival at a terminal. The act of taking the receiver off the hook at the terminal telephone immediately illuminates the line lamp at the dispatcher's switchboard. The dispatcher answers the call by pressing a key corresponding with the lamp, which places him in direct connection with the motorman. After receiving the call the dispatcher gives the motorman the time for his departure from that terminal and any special instructions that may be needed The system in this way gives the dispatcher the exact location of all cars and advises him of any gaps in the service. In case of gaps he can order out trippers to fill in until service again becomes normal. The switchboard used at the dispatcher's office is made up in the form of a flat-top desk having mounted on it three turret apparatus cabinets. Two turrets include straight dispatching line equipments, while the third is a combination dispatcher's and commercial board. During the rush hours three dispatchers are required, but ordinarily one or two are sufficient. Two of the turrets contain forty dispatching lines each, while the third carries twenty dispatching lines and also a multiple of sixty commercial lines appearing in the railway company's private branch exchange. This arrangement permits the night dispatcher, whose duties are light, to care for the commercial business, thereby eliminating a night operator for the commercial or regular private branch exchange board. Space is provided on the top of the desk for schedules, train sheets, etc. A tier of drawers accommodates past records, special schedules and a supply of dispatching stationery. The train sheets, arranged by train number and also in chronological order, are always before each dispatcher so that the whole traffic situation at any moment can be read at a glance. Not only is the best of service under normal operating conditions assured by this system, but a means for rapidly straightening out traffic tangles caused by delays is also provided. The system gives the railway company a permanent record of every train movement, late cars and times of accidents. There is also available in cases of court actions much exact testimony which would otherwise be based upon the inaccurate memory of witnesses. Furthermore, train crews promptly report trouble with cars and give many details which they would not bother to make out in written form.

Local Battery Telephones. Local battery telephones may be divided into two general classes, namely, series and bridging.

Series Telephone. (Fig. 28.) Series telephones give good service where there is only one telephone per line. They have been used where there are several telephones per line by connecting them in series with the line, but for such service bridging telephones are more satisfactory. The hook-switch, *H*, is shown in its raised position, so as to connect the receiver, *R*, and the secondary, *S*, in eries in a circuit between the binding posts, *LL*, of the instrument; nd at the same time the transmitter, *T*, the primary, *P*, of the nduction coil and the battery, *B*, are connected in a local circuit by hemselves. This is the condition for receiving or transmitting peech. When the hook is depressed, as when the telephone is not n use, the bell or ringer, *R'*, is connected across the binding posts, *L*. This circuit would also include the generator, *G*, but for the act that it is normally shunted out by the springs, *g* and *g'*. The ontact between these springs is automatically opened, however,

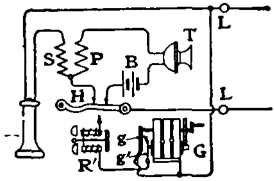


FIG. 28.—Series telephone.

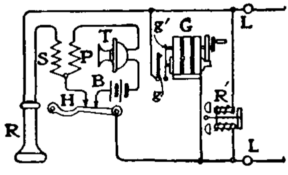


FIG. 29.—Bridging telephone.

when the generator is operated and then there is current from the generator to the line through the bell, *R'*. The springs, *gg'*, form what is called an automatic shunt for the generator, their function being to remove the resistance of the generator armature from the circuit at all times save when the generator is in use.

Bridging Telephone. (Fig. 29.) Bridging telephones give good service where it is necessary to have several telephones per line. For this service these instruments are bridged across the line, that is, they are connected in parallel with each other. In the bridging telephone the arrangement of the receiver, induction coil, transmitter and battery is identical with that of the series telephone (Fig. 28). The ringer, however, is bridged permanently across the line and the generator is placed in a circuit across the line which is normally open, but which is closed automatically by the spring, *g'*, when the generator is operated.

Telephone Disturbance. Annoying inductive effects of adjacent alternating currents may be sufficiently reduced by transposing the telephone line wires relatively to each other so that the influences on one side of the line shall oppose and nearly equal the influence on the other.

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THE FOLLOWING PAGES ARE REPRINTED FROM THE 1915 EDITION

Standard General Electric Railway Motors

NON-COMMUTATING POLE TYPE, DIRECT CURRENT 500 VOLTS

Motor		No motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
Name	H.p.				
54[1]	25	2	K-10	1,900	4,760
		4	K-12		8,865
60[1]	25	2	K-10	1,700	4,360
		4	K-12		8,065
800	25	2	K-10	1,950	4,860
		4	K-12		9,065
81	30	2	K-10	2,020	5,000
		4	K-12		9,345
78	35	2	K-10	2,580	6,080
		4	K-28		11,630
1000[1]	35	2	K-10	2,185	5,330
		4	K-28		10,200
*58[1]	37	2	K-10	2,225	5,410
		4	K-28		10,360
67[1]	40	2	K-10	2,450	5,860
		4	K-28		11,260
*80	40	2	K-10	2,850	6,660
		4	K-28		12,860
88	40	2	K-10	3,070	7,080
		4	K-28		13,630
70[1]	40	2	K-10	2,750	6,460
		4	K-28		12,460
53[1]	45	2	K-11	2,850	6,750
		4	K-14		13,665
57[1]	50	2	K-11	3,030	7,110
		4	K-14		14,385
		4	Mult. unit		14,745
90[1]	50	2	K-36	2,875	6,765
		4	K-35		13,750
		4	Mult. unit		14,171
*98	50	2	K-36	3,290	7,815
		4	K-35		14,910
		4	Mult. unit		15,865
*87	60	2	K-28	3,380	7,783
		4	K-34		15,796
		2	Mult. unit		8,690
		4	Mult. unit		16,207
74[1]	65	2	K-28	3,535	8,530
		4	K-34		15,600
		2	Mult unit		9,360
		4	Mult. unit		16,295

[1] These motors have been superseded by later types, but owing to their widespread use, data concerning them are included for comparison.

* Characteristic curves, Figs. 6 to 9.

GE 500-Volt Non-commutating Pole Motors.—*Continued*

Motor Name	H.p.	No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight total equipment, pounds
*73	75	2	K-28	4,137	9,740
		4	K-34		18,760
		2	Mult. unit		10,580
		4	Mult. unit		19,670
*66	125	2	Mult. unit	4,400	11,515
		4	Mult. unit		21,270
55	160	2	Mult. unit	5,420	14,085
		4	Mult. unit		27,520
76	160	2	Mult. unit	5,170	13,585
		4	Mult. unit		26,520
69†	200	2	Mult. unit	6,250	15,820
		4	Mult. unit		31,000

GE 600-Volt D.C. Commutating Pole Type Railway Motors

Motor Name	H.p.	No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight total equipment, pounds
200	40	2	K-36	2,120	5,490
		4	K-35		9,880
		4	Mult. unit		10,410
226	45	2	K-36	2,180	5,620
		4	K-35		10,430
		4	Mult. unit		10,670
203	50	2	K-36	2,150	5,550
		4	K-35		10,270
		4	Mult. unit		10,550
*203	50	2	K-36	2,600	6,450
		4	K-35		12,100
		4	Mult. unit		12,350
216	50	2	K-36	2,875	7,000
		4	K-35		13,200
		4	Mult. unit		13,450
219	50	2	K-36	2,887	6,833
		4	K-35		13,140
		4	Mult. unit		13,390
201	65	2	K-36	2,735	6,710
		4	K-35		12,610
		2	Mult. unit		7,490
		4	Mult. unit		13,100
201	65	2	K-36	2,870	7,000
		4	K-35		13,180
		2	Mult. unit		7,690
		4	Mult. unit		13,280

* Characteristic curves, Fig. 10, 11, and 12.
† This motor has been superseded by later types, but owing to its widespread use, data concerning it is included for comparison.

GE 600-Volt D.C. Commutating Pole Type Railway Motors.—*Continued*

Motor Name	Motor H.p.	No. motor	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
*210	70	2	K-36	3,380	7,750
		4	K-34		15,710
		2	Mult. unit		8,690
		4	Mult. unit		15,680
218	70	2	K-36	3,200	8,100
		4	K-34		15,000
		2	Mult. unit		8,330
		4	Mult. unit		14,960
204	75	2	K-35	3,400	8,500
		4	K-34		15,800
		2	Mult. unit		8,730
		4	Mult. unit		15,760
*214	80	2	K-35	3,806	9,282
		4	K-34		17,414
		2	Mult. unit		9,430
		4	Mult. unit		17,240
*205	110	2	K-35	3,925	9,200
		4	K-44		18,330
		2	Mult. unit		9,800
		4	Mult. unit		18,300
225	115	2	Mult. unit	3,860	10,200
		4	Mult. unit		18,760
222	140	2	Mult. unit	4,100	10,710
		4	Mult. unit		19,900
207	165	2	Mult. unit	5,160	13,050
		4	Mult. unit		25,950
212	235	2	Mult. unit	6,000	14,800
		4	Mult. unit		29,780
209	275	4	Mult. unit	10,400	50,930
GE 600/1200-Volt D.C. Commutating Pole Type Railway Motors					
217	50	2	K-42	3,250	8,650
		4	K-47		15,150
		4	Mult. unit		17,150
*205-A	80	2	K-42	3,850	9,850
		4	K-47		17,600
		4	Mult. unit		19,730
225	105	4	Mult. unit	3,860	20,980
205	110	2	Mult. unit	3,930	10,400
		4	Mult. unit		18,250
207	110	4	Mult. unit	5,160	26,220
222	130	4	Mult. unit	4,100	22,380
*207-A	140	4	Mult. unit	5,160	26,870
GE 1200-Volt D.C. Commutating Pole Type Railway Motors					
205	80	2	K-42	3,850	9,850
			Mult. unit		20,000

*Characteristic curves, Figs. 13 to 17.

Standard Westinghouse Railway Motors

DIRECT-CURRENT NON-COMMUTATING POLE TYPE
(Designed prior to 1904)

Motor		No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
Name	H.p.				
*12-A	25	2	K-10	2,200	5,400
		4	K-12		10,140
12-A	30	2	K-10	....	5,400
		4	K-12		10,140
*69	30	2	K-10	1,950	4,900
		4	K-12		9,140
*49	35	2	K-10	1,925	4,900
		4	K-28		9,300
92-A	35	2	K-10	2,265	5,580
		4	K-28		10,500
68-C	40	2	K-10	2,280	5,600
		4	K-28		10,560
101	40	2	K-10	2,645	6,340
		4	K-28		12,020
101-C	40	2	K-10	2,780	6,700
		4	K-28		12,700
38-B	50	2	K-10	2,380	5,810
		4	K-28		10,960
*93-A	60	2	K-11	3,440	8,080
		4	K-14		16,360
56	60	2	K-11	3,000	7,200
		4	K-14		14,400
112	75	2	Unit switch	3,440	8,775
		4	Unit switch		15,820
76	75	2	Unit switch	3,840	9,575
		4	Unit switch		17,420
85	75	2	Unit switch	4,500	10,895
		4	Unit switch		20,060
121	90	2	Unit switch	4,300	10,530
		4	Unit switch		20,015
*119	125	2	Unit switch	4,680	11,425
		4	Unit switch		21,610
*113	195	2	Unit switch	6,550	15,795
		4	Unit switch		Special
*101-B-2	40	2	K-10-A	2,780	6,635
		4	K-28-B		12,620
		2	Unit switch		7,225
		4	Unit switch		13,065
101-D-2	50	2	K-36-J	2,780	6,705
		4	K-35-G		12,695
		2	Unit switch		7,245
		4	Unit switch		13,100

* Characteristic curves, Figs. 18 to 24.

WESTINGHOUSE D.C. NON-COMMUTATING POLE TYPE RAILWAY MOTORS.—*Continued*
(Designed since 1904)

Motor Name	Motor H.p.	No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
93-A-2	60	2	K-36-J	3,440	8,100
		4	K-34-D		15,900
		2	Unit switch		8,585
		4	Unit switch		15,815
*112-B	75	2	K-35-G	3,485	8,505
		4	K-34-D		16,245
		2	Unit switch		8,860
		4	Unit switch		16,000
121-A	90	2	K-35-G	4,300	10,130
		4	K-34-D		19,500
		2	Unit switch		10,530
		4	Unit switch		19,815
*114	160	2	Unit switch	5,300	13,080
		4	Unit switch		Special

WESTINGHOUSE D.C. COMMUTATING POLE RAILWAY MOTORS

Motor Name	H.p. 600 V.	H.p. 500 V.	No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
328	37	30	2	K-11-M	1,680	4,400
			4	K-28-U		8,155
			2	Unit switch		4,480
			4	Unit switch		8,365
323-A	40	33	2	K-11-M	1,890	4,820
			4	K-28-U		8,995
			2	Unit switch		5,445
			4	Unit switch		9,505
312	50	40	2	K-11-M	2,630	6,300
			4	K-28-U		11,955
			2	Unit switch		6,945
			4	Unit switch		12,505
307	50	40	2	K-36-J	2,850	6,815
			4	K-35-G		12,995
			2	Unit switch		7,385
			4	Unit switch		13,390
†319-B	50	40	2	K-36-J	2,865	6,845
			4	K-35-G		13,055
			2	Unit switch		7,415
			4	Unit switch		13,450
306	60	50	2	K-36-J	2,850	6,815
			4	K-35-G		12,995
			2	Unit switch		7,400
			4	Unit switch		13,455
†316	60	50	2	K-36-J	3,050	7,215
			4	K-35-G		14,190
			2	Unit switch		7,800
			4	Unit switch		14,255

* Characteristic curves, Figs. 25 and 26.
† Characteristic curves, Figs. 27 and 28.

WESTINGHOUSE D.C. COMMUTATING POLE RAILWAY MOTORS.—*Continued*

Motor Name	H.p. 600 V.	H.p. 500 V.	No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
305	75	60	2	K-36-J	3,550	8,215
			4	K-34-D		16,370
			2	Unit switch		9,000
			4	Unit switch		16,360
310	75	60	2	K-36-J	3,440	7,995
			4	K-34-D		15,930
			2	Unit switch		8,780
			4	Unit switch		15,920
304	90	75	2	K-35-G	3,550	8,655
			4	K-34-D		16,525
			2	Unit switch		9,000
			4	Unit switch		16,360
*317	90	75	2	K-35-G	3,660	8,875
			4	K-34-D		16,965
			2	Unit switch		9,220
			4	Unit switch		16,800
303-A	110	......	2	Unit switch	4,150	10,230
			4	Unit switch		19,220
*302-A	140	......	2	Unit switch	4,685	11,430
			4	Unit switch		21,630
*301-B	175	......	2	Unit switch	5,510	13,500
			4	Unit switch		Special
300-B	220	......	2	Unit switch	6,400	15,500
			4	Unit switch		Special

WESTINGHOUSE 1200- AND 1500-VOLT D.C. RAILWAY MOTORS

Motor Name	H.p. 600 V.	H.p. 500 V.	No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
*333	115	100	2	Unit switch	3,900	Special
			4	Unit switch		Special
	750 V.	600 V.				
*321	110	90	2	Unit switch	4,150	Special
	110	90	4	Unit switch		Special
*322	140	115	2	Unit switch	4,685	Special
	140	115	4	Unit switch		Special
*308-D-3	260	225	2	Unit switch	6,740	Special
	260	225	4	Unit switch		Special

* Characteristic curves, Figs. 29 to 35.

Westinghouse Single-phase A.C. Railway Motors

For Motor Cars

Motor		No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
Name	H.p.				
*135	75	4	Unit switch	4,500	Special
132	100	4	Unit switch	5,100	Special
132-A	100	4	Unit switch	5,575	Special
*132-C	100	4	Unit switch	5,575	Special
132-F	100	4	Unit switch	5,700	Special
*148-A	125	4	Unit switch	6,100	Special
*133	135	4	Unit switch	6,000	Special
*156	150	4	Unit switch	7,700	Special
409	175	4	Unit switch	7,846[1]	Special
409-B	175	2	Unit switch	6,700	Special
409-D	175	4	Unit switch	7,846[1]	Special
For Locomotives					
410	125	4	Unit switch	13,365[1]	Special
*409-C[2]	175	4 pairs	Unit switch	16,500[1]	Special
151	175	4	Unit switch	10,600	Special
137	240	3	Unit switch	15,700	Special
*130	250	4	Unit switch	16,400	Special
403	310	4	Unit switch	21,220[1]	Special
*403-A	310	4	Unit switch	19,800[1]	Special
403-B	310	4	Unit switch	19,800[1]	Special
406	675	2	Unit switch	41,200	Special

[1] Motors arranged for quill drive. Weights here include drive details.
[2] 409-C motor is arranged to operate in pairs. Weight given is for one pair of motors complete with drive details.

Allis Chalmers Direct-current Railway Motors

Motor		No. motors	Type control	Weight of motor including gear and gear case, pounds	Weight, total equipment, pounds
Name	H.p.				
*301	40	2	S-3	2,630	6,550
		4	S-4		12,300
*501	50	2	S-3	2,610	6,450
		4	S-4		12,100
*302	55	2	S-3	3,060	7,640
		4	S-4		14,030

* Characteristic curves, Figs. 36 to 46.

Motors for Low Floor Cars. Motors commonly known as "Low Floor," "Dachshund" and "Baby," for use on low floor cars built with small wheels, are of smaller diameter than but do not differ in general electrical design from motors ordinarily used with 33-in. wheels.

Pressed Steel Motor. Certain railway motor parts formerly made of cast steel are now made of pressed steel, for some of the motor capacities of most common application. The weight of a 40-h.p. pressed steel motor is about 12 per cent. less than a cast steel motor of the same capacity.

Railway Motor Characteristic Curves. Figs. 6 to 46, inclusive, give the operating characteristics of some of the commercial motors when operating at particular voltages with particular gear ratios and driving-wheel diameters. The ordinates for the *tractive effort* curve for a series motor with any gear ratio or driving-wheel diameter may be derived from the ordinates of the original *tractive effort curve for that motor as follows:*

$$T^1 = T\left(\frac{G^1 \times D}{G \times D^1}\right)$$

in which T^1 = tractive effort ordinate for any current value on the derived curve

T = tractive effort ordinate for the same current value on the original curve

G^1 = gear ratio for derived tractive effort curve

G = gear ratio for original tractive effort curve

D^1 = driving-wheel diameter for derived tractive effort curve

D = driving-wheel diameter for original tractive effort curve.

The approximate values of the ordinates of the *speed* curve for a series motor with any gear ratio or driving-wheel diameter operating at any other electromotive force, differing slightly from the electromotive force at which the original speed curve was obtained, may be derived from the original *speed* curve as follows:

$$S^1 = S\left(\frac{E^1 \times G \times D^1}{E \times G^1 \times D}\right)$$

in which S^1 = speed ordinate for any current value on the derived curve

S = speed ordinate for the same current value on the original curve

E^1 = e.m.f. for derived speed curve

E = e.m.f. for original speed curve.

(NOTE: When the electromotive force is not changed, that is, when $E^1 = E$, the value given for S^1 is exact.)

Example of the Derivation of Tractive Effort and Speed Curves. Fig. 48 (p. 265) shows tractive effort and speed curves (dotted) derived from the original curves by the above process as follows: The original gear ratio, wheel diameter and voltage are 2.78, 33, and 500, respectively, and it is desired to derive the tractive effort and approximate speed curves for a gear ratio, wheel diameter and voltage of 3.83, 36, and 550, respectively. In this case, $G^1 = 3.83$, $G = 2.78$, $D^1 = 36$, $D = 33$, $E^1 = 550$ and $E = 500$.

Therefore

$$T^1 = T\left(\frac{3.83 \times 33}{2.78 \times 36}\right)$$
$$= 1.26\ T$$

and

$$S^1 = S\left(\frac{550 \times 2.78 \times 36}{500 \times 3.83 \times 33}\right)$$
$$= 0.87\ S$$

Thus in Fig. 48 the dotted tractive effort curve is the locus of all points the ordinate of each of which is equal to 1.26 times the corresponding ordinate on the original tractive effort curve, and the dotted speed curve is the locus of all points the ordinate of each of which is equal to 0.87 times the corresponding ordinate on the original speed curve.

(NOTE: Where, as for preliminary investigation, only one or very few points on the derived characteristic may be desired, it may be advantageous to obtain the desired derived values by the above method of calculation directly without actually plotting the derived curve.)

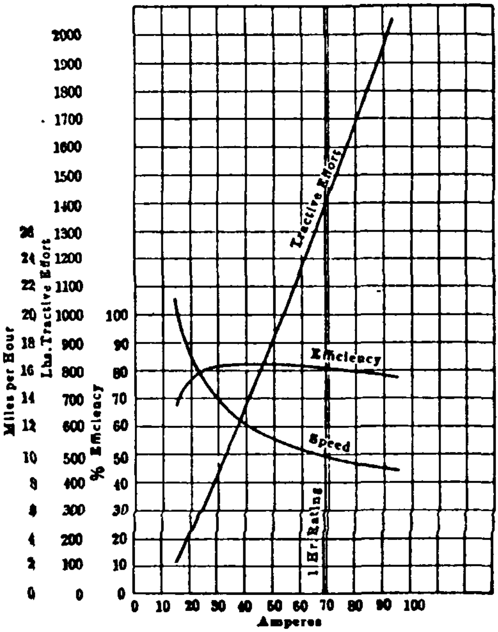

FIG. 6.—General Electric No. 58 railway motor. 37 h p., 500 volts, pinion 15, gear 69, ratio 4.60, wheels 30 in.

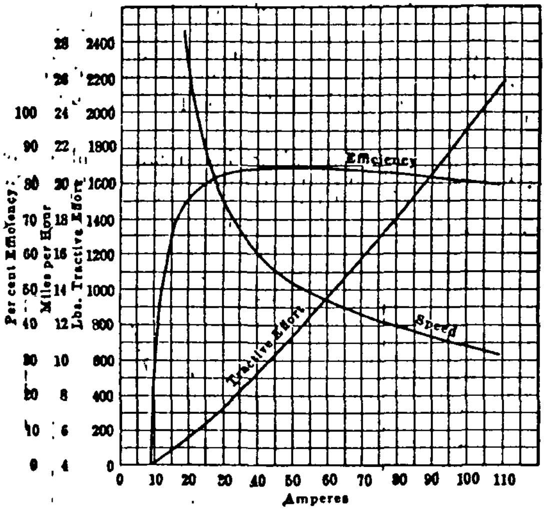

FIG. 7.—General Electric No. 80 railway motor. 40 h.p., 500 volts, pinion 17, gear 69, ratio 4.06, wheels 33 in.

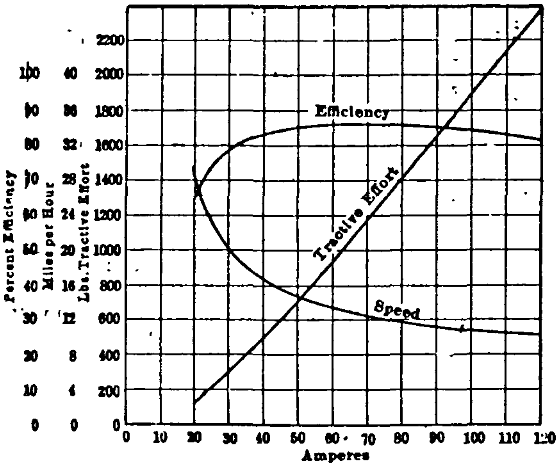

FIG. 8.—General Electric No. 98 railway motor. 50 h.p., 500 volts, pinion 16, gear 71, ratio 4.43, wheels 33 in.

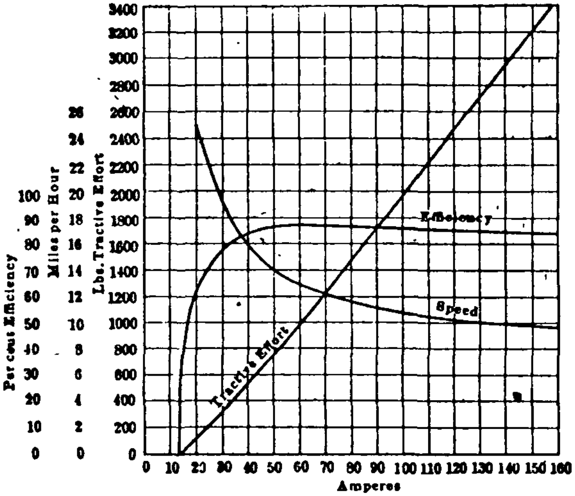

FIG. 9.—General Electric No. 87 railway motor. 60 h.p., 500 volts, pinion 16, gear 71, ratio 4.44, wheels 33 in.

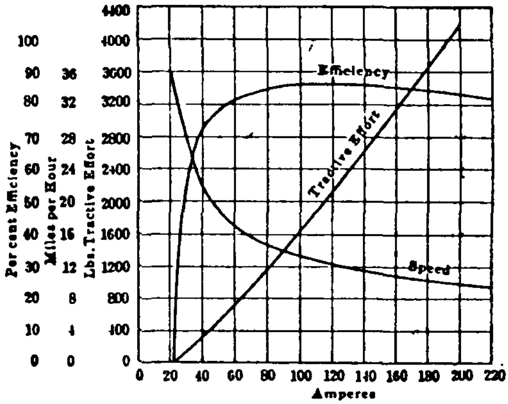

FIG. 10.—General Electric No. 73 railway motor. 75 h.p., 500 volts, pinion 17, gear 73, ratio 4.30, wheels 33 in.

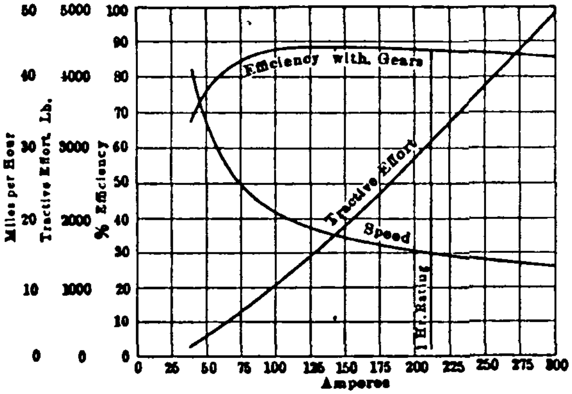

FIG. 11.—General Electric No. 66 railway motor. 125 h.p., 500 volts, pinion 17, gear 61, ratio 3.59, wheels 33 in.

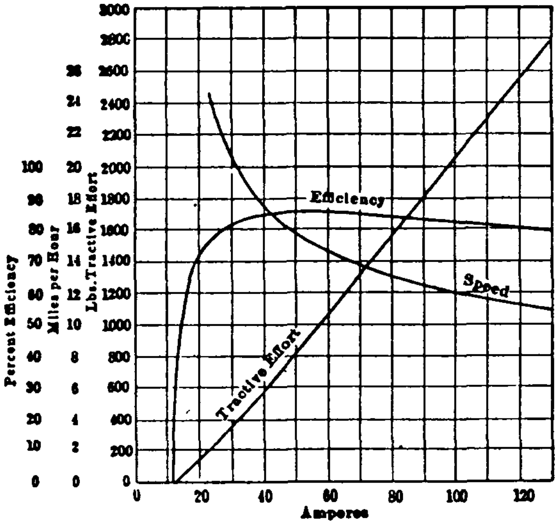

FIG. 12.—General Electric No. 203 railway motor. 50 h.p., 600 volts, pinion 15, gear 69, ratio 4.60 wheels 33 in.

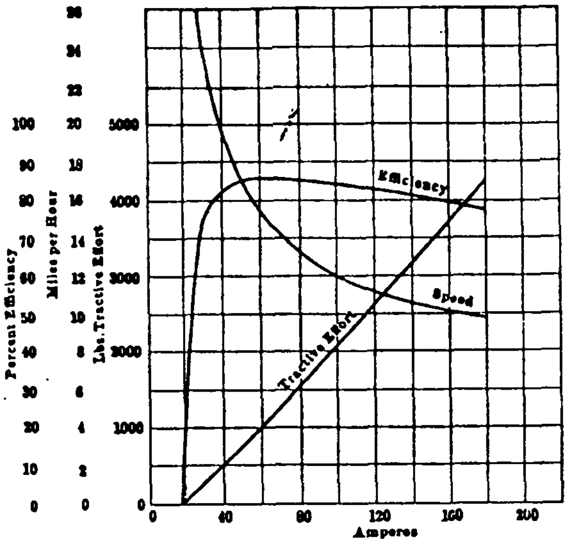

FIG. 13.—General Electric No. 210 railway motor. 70 h.p., 600 volts, pinion 16, gear 71, ratio 4.43, wheels 33 in.

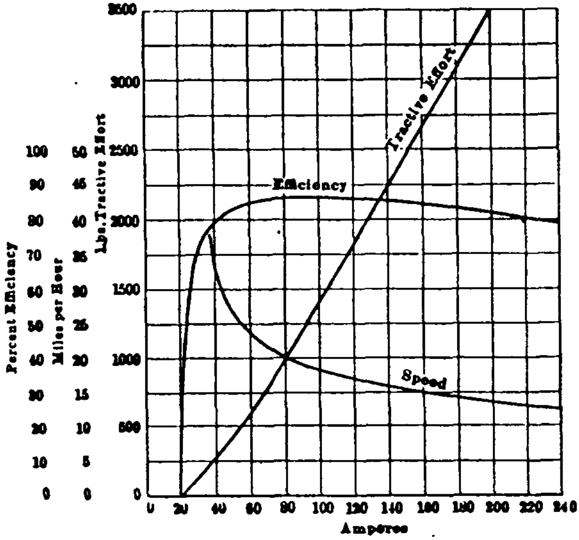

FIG. 14.—General Electric No. 214 railway motor. 75 h.p., 600 volts, pinion 17, gear 60, ratio 3.53, wheels 33 in.

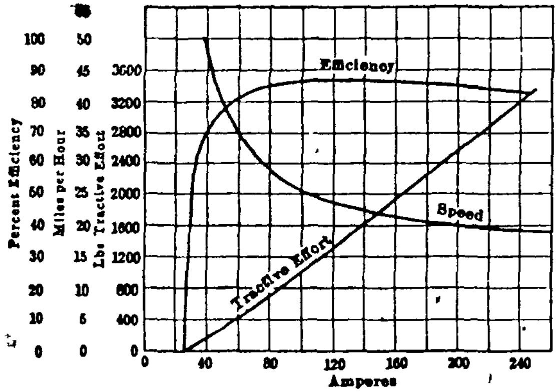

FIG 15.—General Electric No. 205-*B* railway motor. 100 h.p., 600 volts, pinion 17, gear 57, ratio 3 35, wheels 33 in.

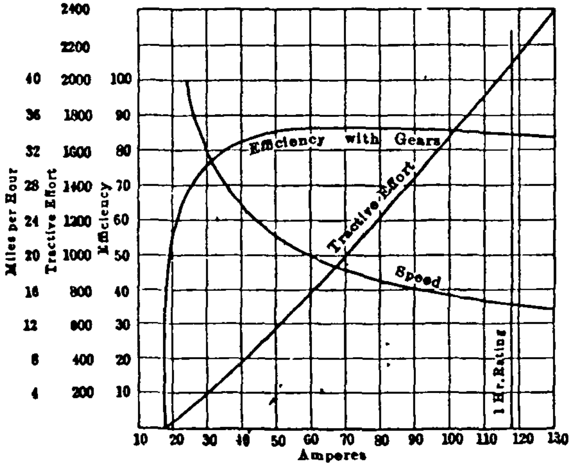

FIG 16.—General Electric No 205-*A* railway motor 80 h p 600 volts pinion 17, gear 67, ratio 3 35, wheels 33 in Wound for operating two in series on 1200 volts

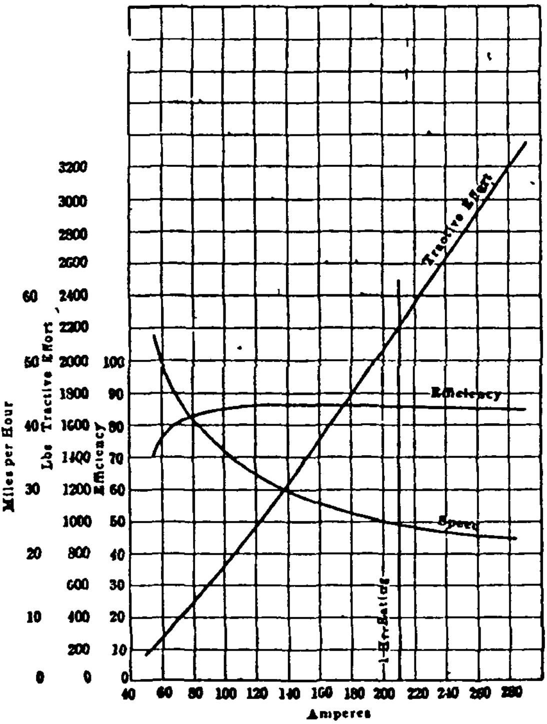

FIG 17 —General Electric No. 207-*A* railway motor 140 h p . 600 volts, pinion 22, gear 59, ratio 2.68, wheels 36 in. Wound for operation two in series on 1200 volts.

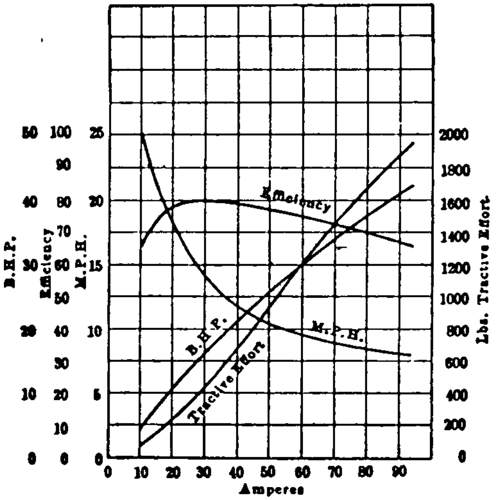

FIG. 18.—Westinghouse No. 12-A-25 railway motor. 25 h.p., 500 volts, pinion 14, gear 68, ratio 4.86, wheels 33 in.

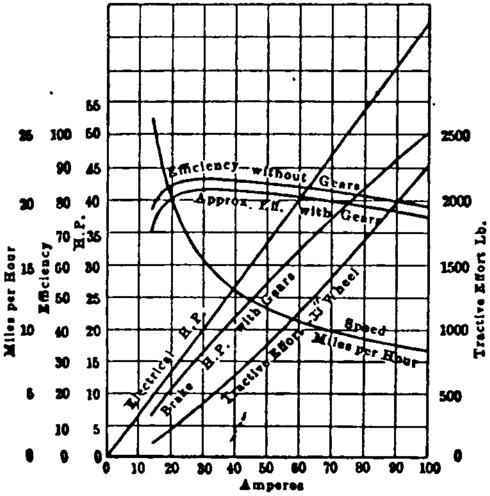

FIG. 19.—Westinghouse No. 69 railway motor. 30 h.p., 500 volts, pinion 14, gear 68, ratio 4.86, wheels 33 in. Continuous capacity 25 amp. at 300 volts, 23 amp. at 400 volts.

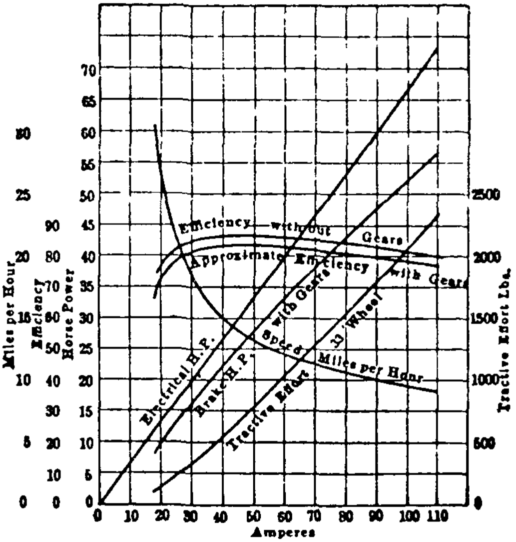

FIG. 20.—Westinghouse No. 49 railway motor. 35 h.p., 500 volts, pinion 14, gear 68, ratio 4 86, wheels 33 in. Continuous capacity, 30 amp. at 300 volts, 28 amp. at 400 volts.

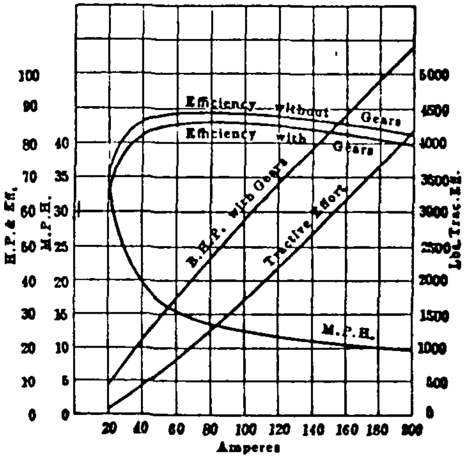

FIG. 21.—Westinghouse No. 93-*A* railway motor. 60 h p., 500 volts, pinion 17, gear 70, ratio 4.12, wheels 33 in.

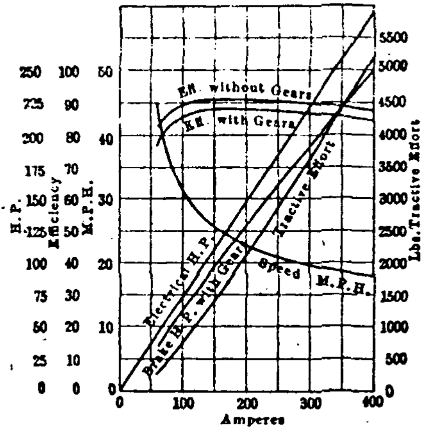

FIG. 22.—Westinghouse No. 119 railway motor. 125 h.p., 550 volts pinion 20, gear 55, ratio 2.75, wheels 33 in. Continuous capacity, 95 amp. at 300 volts, 85 amp. at 400 volts.

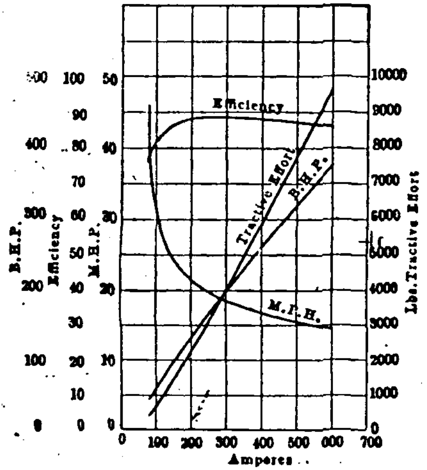

FIG. 23.—Westinghouse No. 113 railway motor. 195 h.p., 550 volts, pinion 19, gear 64, ratio 3.37, wheels, 36 in.

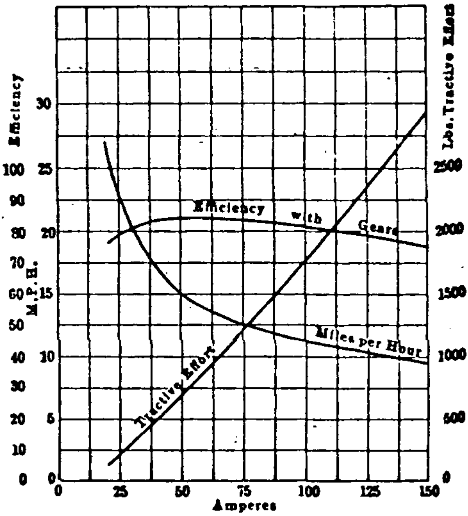

FIG. 24.—Westinghouse No. 101-*B* railway motor. 40 h.p., 500 volts pinion 17, gear 67, ratio 3.94, wheels 33 in.

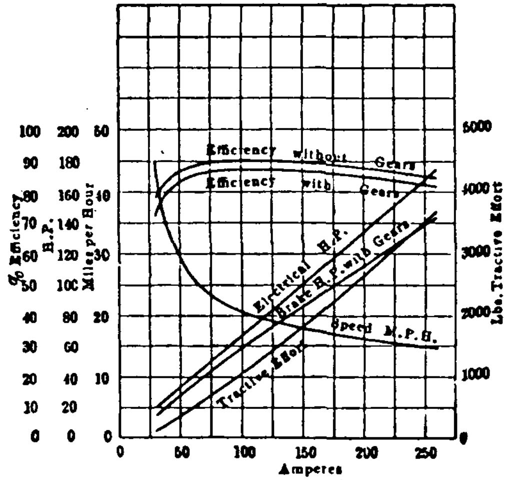

FIG. 25.—Westinghouse No. 112-*B* railway motor. 75 h.p., 500 volts. *pinion 19, gear 70, ratio 3.68, wheels 36 in.* Continuous capacity, 60 amp, at 300 volts, 55 amp. at 400 volts.

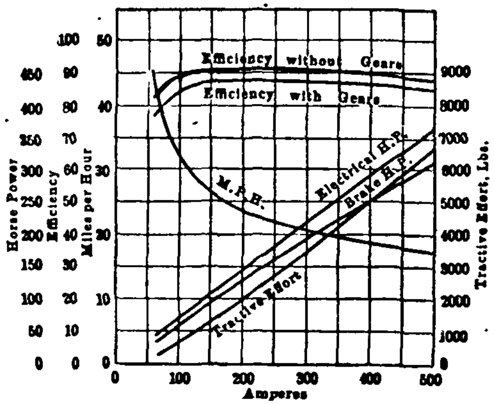

FIG. 26.—Westinghouse No. 114 railway motor. 160 h.p., 550 volts, pinion 20, gear 57, ratio 2.85, wheels 33 in. Continuous capacity, 120 amp. at 300 volts, 110 amp. at 400 volts.

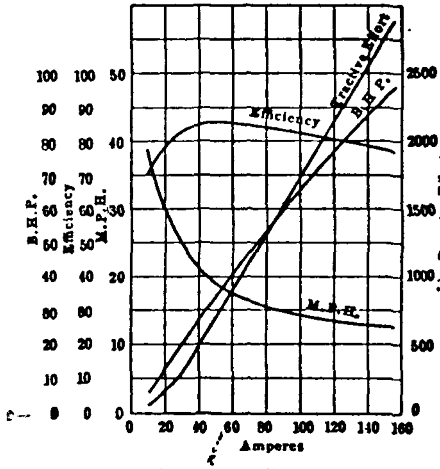

FIG. 27.—Westinghouse No. 319-*B* railway motor. 50 h.p., 600 volts, pinion 17, gear 69, ratio 4.06, wheels 33 in.

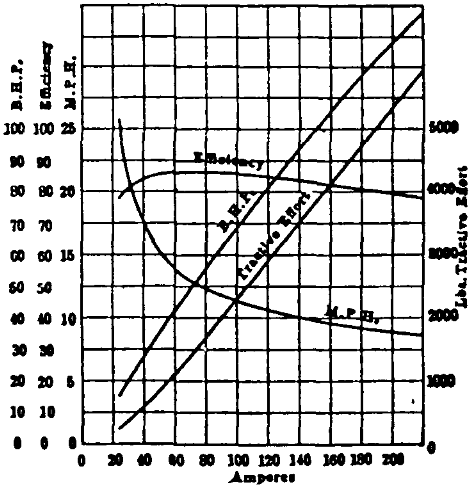

Fig. 28.—Westinghouse *No.* 316 *railway motor.* 75 *h.p., 600 volts, pinion* 15, gear 71, ratio 4.73, wheels 30 in.

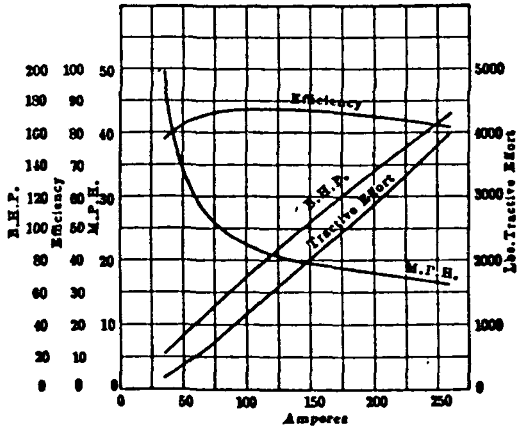

Fig. 29.—Westinghouse No. 317 railway motor. 90 h.p., 600 volts, pinion 19, gear 70, ratio 3.68, wheels 33 in.

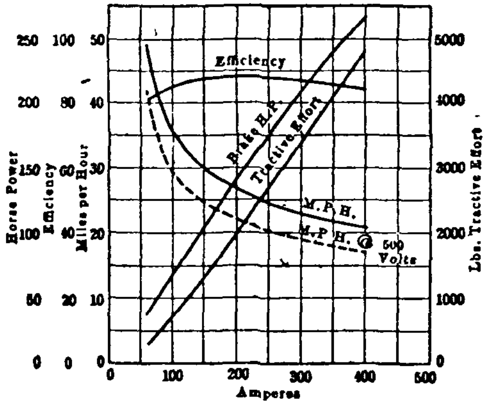

FIG. 30.—Westinghouse No. 302-*A* railway motor. 140 h p., 600 volts, pinion 21, gear 56, ratio 2.67, wheels 36 in.

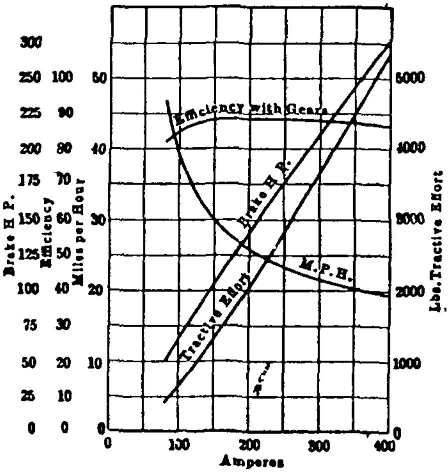

FIG. 31.—Westinghouse No 301 railway motor. 175 h p., 600 volts, pinion 18, gear 59, ratio 3 28, wheels 36 in.

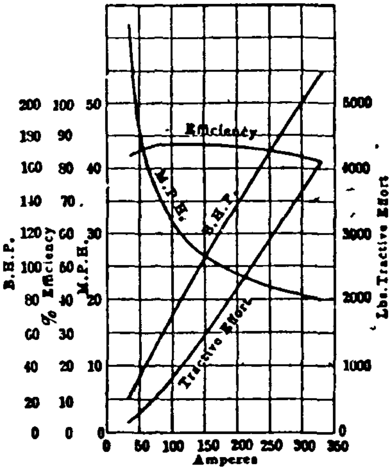

FIG. 32.—Westinghouse No. 333-*B* railway motor. 115 h p., 600 volts, pinion 19, gear 58, ratio 3.05, wheels 36 in. Wound for operation two in series on 1200 volts.

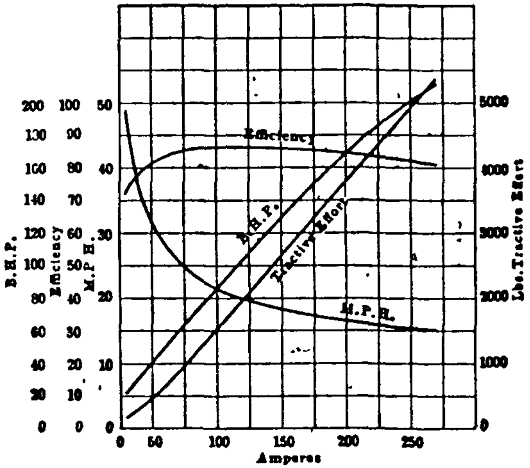

FIG. 33.—Westinghouse No. 321 railway motor. 110 h p., 750 volts, pinion 16, gear 61, ratio 3 81, wheels 36 in. Wound for operation two in series on 1500 volts.

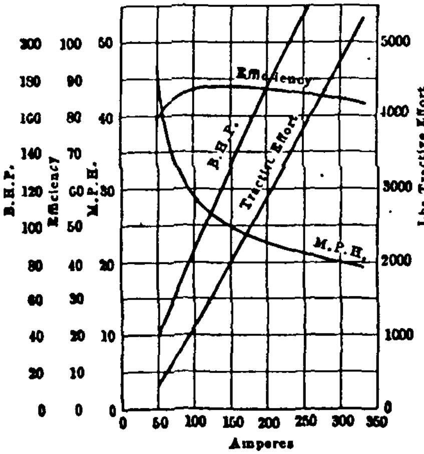

FIG. 34.—Westinghouse No. 322 railway motor. 140 h.p., 750 volts, pinion 16, gear 61, ratio 3.81, wheels 36 in. Wound for operation two in series on 1500 volts.

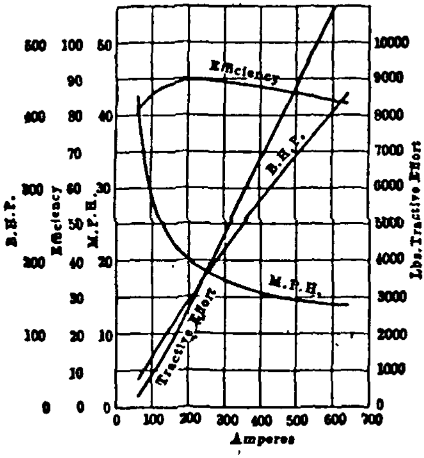

FIG. 35.—Westinghouse No. 308-*D* railway motor, 225 h.p. (250 h.p. with forced ventilation) 600 volts, pinion 19, gear 72, ratio 3.79, wheels 36 in. Wound for operation two in series on 1200 volts.

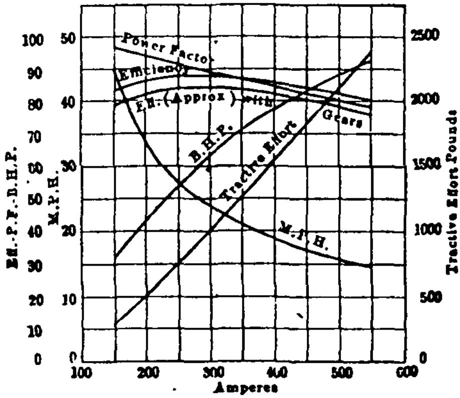

FIG. 36.—Westinghouse No. 135-*B* railway motor. 380 amp., 210 volts, 25-cycle, single-phase, alternating-current, pinion 17, gear 60, ratio 3.53, wheels 36 in. Continuous capacity 170 amp. at 130 volts.

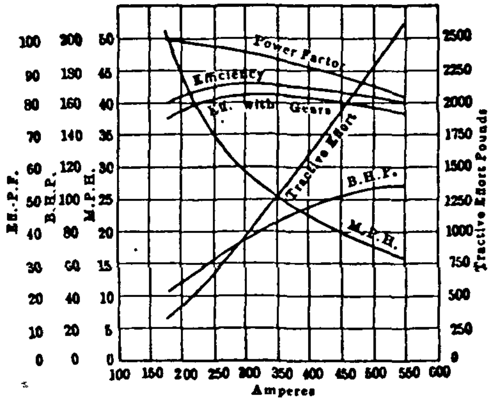

FIG. 37.—Westinghouse No. 132-C railway motor. 430 amp., 235 volts, 25-cycle, single-phase, alternating-current, pinion 20, gear 63, ratio 3.15, wheels 36 in. Continuous capacity 200 amp. at 235 volts.

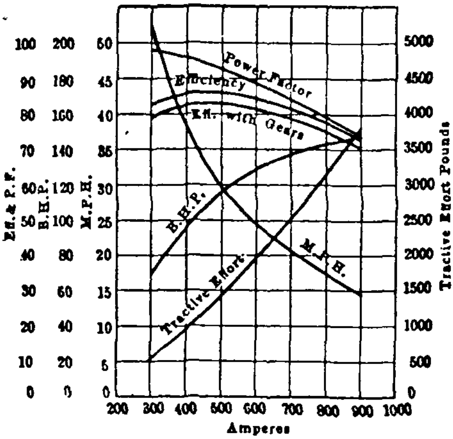

FIG. 38.—Westinghouse No. 148-*A* railway motor. 550 amp. (forced ventilation), 225 volts, 25-cycle, alternating-current, pinion 25, gear 66, ratio 2.64, wheels 38 in. Continuous capacity 325 amp. at 150 volts.

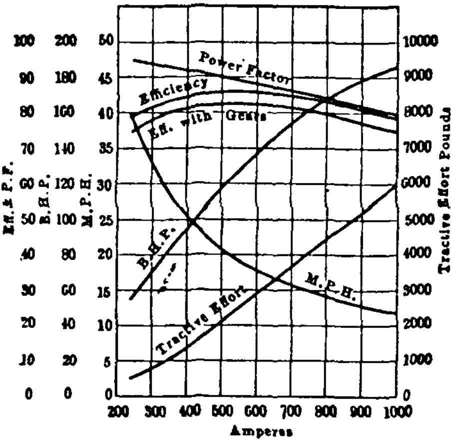

FIG. 39.—Westinghouse No. 133 railway motor. 600 amp. (forced ventilation), 235 volts, 25-cycle, alternating-current, pinion 17, gear 72, ratio 4.24, wheels 38 in. Continuous capacity, 600 amp. at 235 volts.

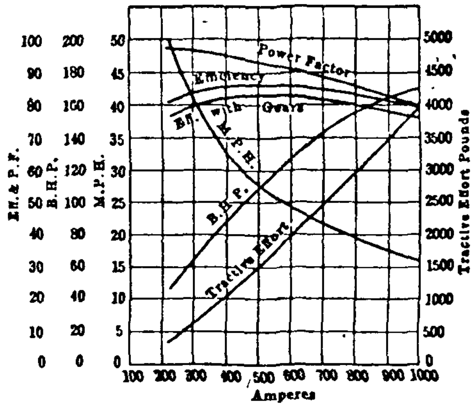

FIG. 40.—Westinghouse No. 156 railway motor. 600 amp. (forced ventilation), 210 volts, 25-cycle, alternating-current, pinion 25, gear 74, ratio 2.96, wheels 42 in. Continuous capacity, 450 amp. at 210 volts.

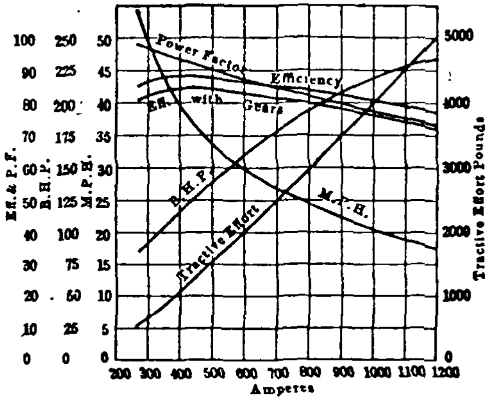

FIG. 41.—Westinghouse No. 409-C locomotive motor. 650 amp. (forced ventilation), 275 volts, 25-cycle, alternating-current, pinion 22, gear 92, ratio 4.18, wheels 63 in. Continuous capacity, 500 amp. at 275 volts.

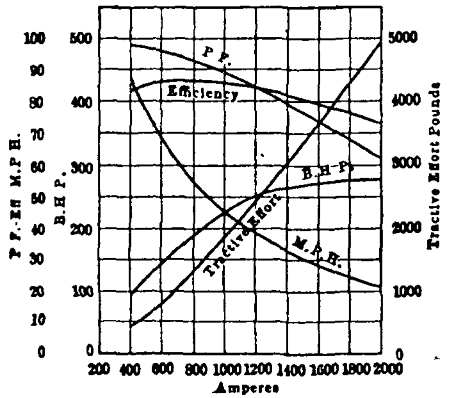

FIG. 42.—*Westinghouse No. 130 locomotive motor. 1130* amp. (*forced* ventilation), 220 volts, 25-cycle, alternating-current, gearless, wheels 62 in. Continuous capacity, 860 amp. at 220 volts.

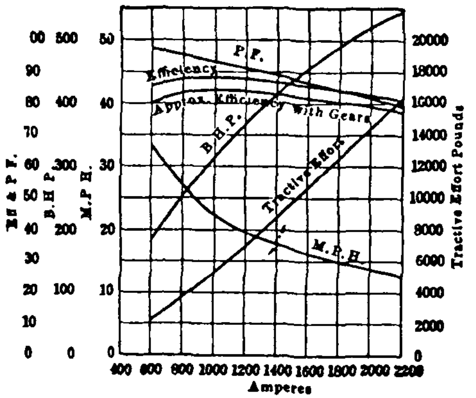

FIG. 43.—Westinghouse No. 403-*A* locomotive motor. 1000 amp. (forced ventilation), 300 volts, 25-cycle, alternating-current, pinion 24, gear 89, ratio 3.71, wheels 63 in. Continuous capacity, 930 amp. at 300 volts.

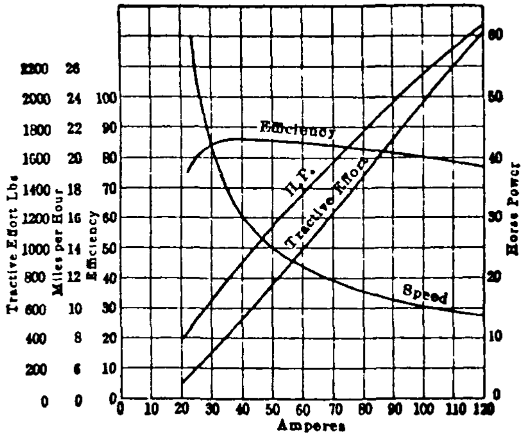

FIG. 44.—Allis-Chalmers No. 301 railway motor. 40 h p., 500 volts, pinion 15, gear 69, ratio 4.60, wheels 33 in.

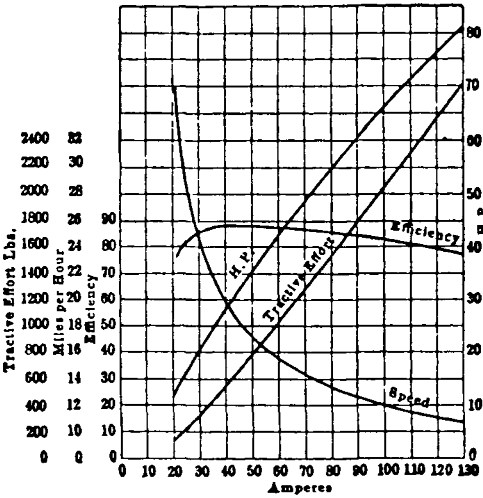

FIG. 45.—Allis-Chalmers No. 501 railway motor. 50 h.p., 600 volts, pinion 15, gear 71, ratio 4.73, wheels 33 in.

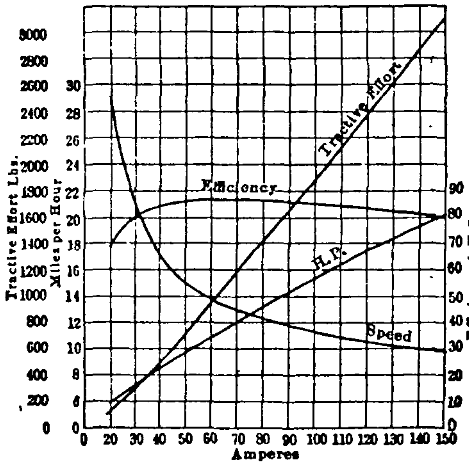

FIG. 46.—Allis-Chalmers No. 302 railway motor. 55 h.p., 500 volts, pinion 16, gear 73, ratio 4.56, wheels 33 in.

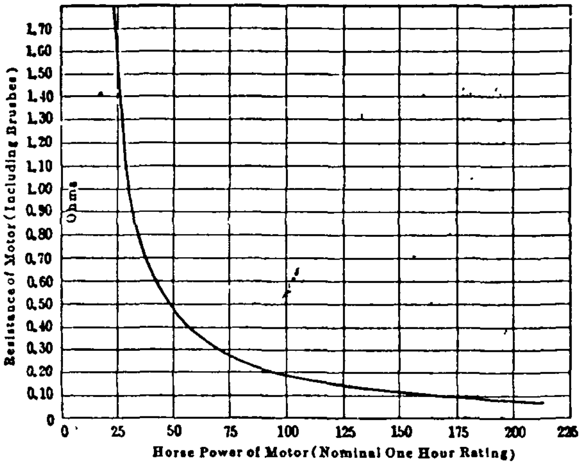

FIG. 47.—Approximate resistance of 500-volt direct-current motors at 75° C.

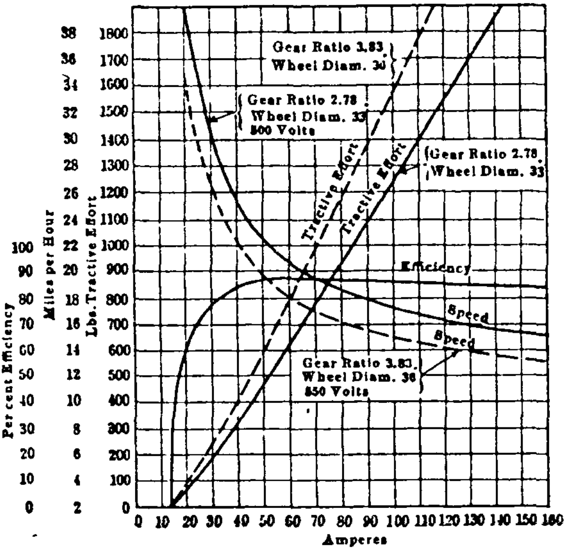

FIG. 48.—Illustrating change of tractive effort and speed curves for a change in gear ratio, driving wheel diameter and voltage. (GE-87 motor, 60 h.p.)

Alternating-current Railway Motors

General Structural Difference between Single-phase Commutating Alternating- and Direct-current Motors. The structural differences between single-phase alternating-current series motors and direct-current motors are summarized by Messrs. R. E. Hellmund and E. W. P. Smith in the Electric Journal, 1912, as follows:

(1) The laminated core of the alternating-current motor;

(2) Comparatively large number of poles of the alternating-current motor;

(3) The use of a distributed auxiliary winding on the alternating-current motor in addition to the main field winding;

(4) The larger armature diameter as compared with that of a direct-current armature;

(5) Short and stubby poles made possible by the small number of field ampere-turns on the alternating-current motor;

(6) The large and wide commutator of the alternating-current motor and the large number of segments as compared with direct-current machines;

(7) Small air gap of the alternating-current motor.

To these should be added preventive leads between the armature winding and commutator in the alternating-current motor.

The use of the laminated field structure, the auxiliary or compensating winding and the preventive leads in the single-phase motor are the differences essential to the single-phase motor.

Laminated Field Structure. The field is laminated in order to reduce the losses and associated heating due to the alternating flux in the iron.

Compensating Winding. The purpose of the compensating winding is to neutralize cross-magnetization, thereby improving commutation and power factor of the motor. Compensation is classified as conductive or inductive, depending upon the process by which the current flowing in the compensating winding is obtained. If the compensating winding is connected to receive current from an external source, as by connecting it in series with the main motor circuit, the compensation is said to be conductive. If the compensating winding is closed on itself so that a current due to a directly induced electromotive force flows within itself, the compensation is said to be inductive. On the New York, Westchester and Boston Railway inductive compensation is used. On the New York, New Haven and Hartford Railroad motors intended only for single-phase operation are inductively compensated. On the same road, motors intended for both single-phase and direct-current operation are conductively compensated. Insulation strain to ground is made negligible in the inductive compensation.

Inductive and Conductive Compensation. When motors operate only on alternating current the difference in results of either inductive or conductive compensation is inappreciable, but where the motor is to be operated with direct current, conductive compensation should be used.

Preventive Leads. Preventive leads in a single-phase commutating motor are added resistance between commutator windings and commutator bars. Their purpose is to restrict the current flowing in the armature coil when short-circuited by the brush. They add their resistance effect to that of the carbon brush bridging the commutator bars. In so restricting the current they improve commutation and they may improve the efficiency of the motor. Whether or not efficiency is improved by the preventive leads depends upon the balance of the following two effects: (*a*) I^2R loss due to the passage of the armature current through the preventive leads on its way to the armature winding. This loss varies directly with the resistance of the leads. (*b*) I^2R loss due to the passage of current set up in the short-circuited armature coil as a result of the electromotive force induced by the field. This loss decreases as the resistance of the preventive leads increases. The preventive leads are of greater value during starting and acceleration.

General Types of Single-phase Commutating Motors. There are three general types of single-phase commutating motors, namely, the series, repulsion and the induction series. Of these the former two are in most general use in America.

Connections of Single-phase Commutator Motors. The conventional diagrams, Figs. 49 to 55, inclusive, show some of the methods of connecting armature, field and compensating coils of single-phase commutator motors. Since the windings on armature and field of the repulsion motors are not electrically connected, operation of this type is possible on comparatively high voltages if the armature be so constructed that a low voltage is induced in it.

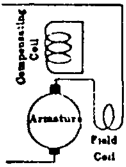

FIG. 49.—Inductively compensated series motor.

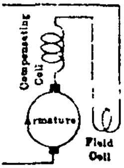

FIG. 50.—Conductively compensated series motor

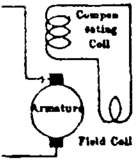

FIG. 51.—Induction series motor.

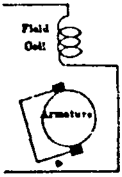

FIG. 52.—Thomson repulsion motor.

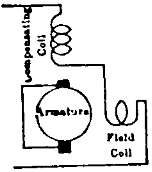

FIG. 53.—Series compensated repulsion motor.

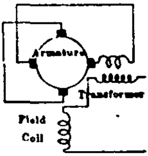

FIG. 54.—Winter-Eichberg repulsion motor.

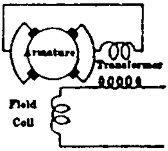

FIG. 55.—Latour repulsion motor.

Three-phase Induction Motor. The three-phase motor is called a constant speed motor (see p. 383 for methods of changing speed). It has the characteristics of a direct-current constant speed motor. Neglecting control, the speed of the three-phase induction motor depends upon the frequency of supply and the speed falls off but little as the load on the motor is increased. The speed of a train driven by three-phase induction motors ordinarily remains nearly constant up grade and down grade. The three-phase induction motor regenerates energy, forcing it back into the line when the motor is driven above its synchronous speed (see p. 213). This process takes place as the train descends a grade, and since this regeneration brings a load on the motor acting as a generator the train is not allowed to accelerate beyond the point at which the force due to gravity is balanced by the forces it is overcoming in driving the motor acting as a generator. Since the three-phase motor is a constant speed motor, in ascending a grade it will make a demand on the power station proportional to the torque required and the losses to the motor. As outlined on p. 214 this demand may, however, be reduced by the energy of regeneration of trains descending. In order to operate on long up grades a three-phase induction motor must have high continuous capacity.

The starting efficiency of three-phase motors is low so the motor is not well suited to service having frequent stops. The tractive effort may be held constant during the starting period, but this is done at the expense of energy lost in the motor-starting resistance

Since there is no electrical connection between the primary winding and the secondary winding of an induction motor, and there is no commutator, this type of motor may be built to operate on voltages which are high compared with those on which motors of other types are operated.

Speed of Induction Motor. The speed of an induction motor is equal to its synchronous speed minus the slip. The value of the *slip increases at practically a constant rate from nearly zero at no* load to about 2 to 5 or 6 per cent. of synchronous speed at full load.

$$S = \frac{f \times 60}{N}$$

in which S = synchronous speed, revolutions per minute
N = number of pairs of poles in the primary winding of the motor
f = frequency of applied electromotive force, cycles per second

$$S_1 = \frac{(100 - s)\,S}{100}$$

in which S_1 = actual speed of motor, revolutions per minute
s = slip, per cent.

TABLE OF SYNCHRONOUS SPEEDS

Number of pairs of poles in motor primary	1	2	3	4	5	6	7
Frequency, cycles per second	Synchronous speed, revolutions per minute						
15	900	450	300	225	180	150	128
25	1500	750	500	375	300	250	214
60	3600	1800	1200	900	720	600	514