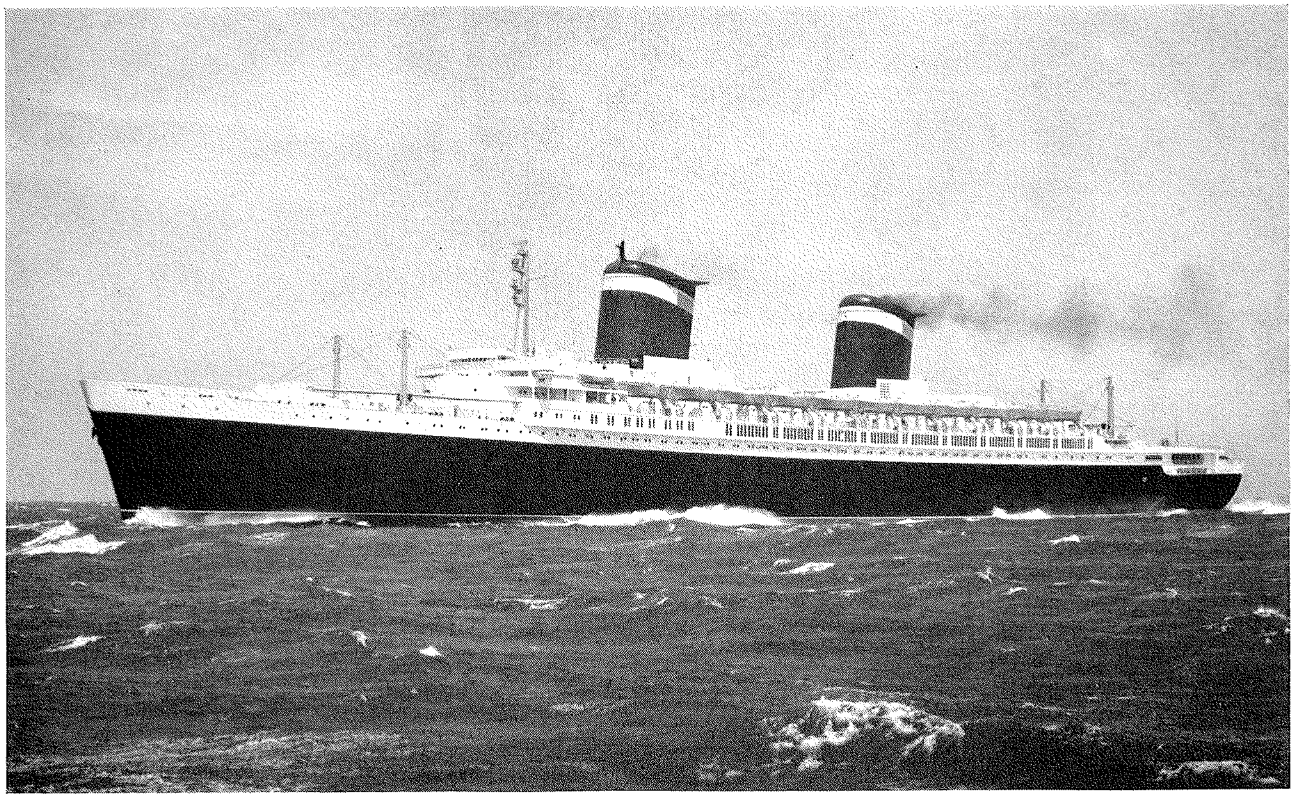
KINGSBURY BEARINGS

for Marine Propulsion

Bulletin M

KINGSBURY



S. S. "United States," queen of the American Merchant Marine. Kingsbury Thrust Bearings installed in main propulsion machinery and in many of the auxiliaries.

Courtesy of Gibbs & Cox, Inc., Naval Architects, and Newport News Shipbuilding & Dry Dock Co., Builders.

KINGSBURY

Thrust and Journal

BEARINGS

for

MARINE PROPULSION

also

KINGSBURY

THRUST METERS

BULLETIN M

KINGSBURY

KINGSBURY MACHINE WORKS, INC.

Main Office and Works

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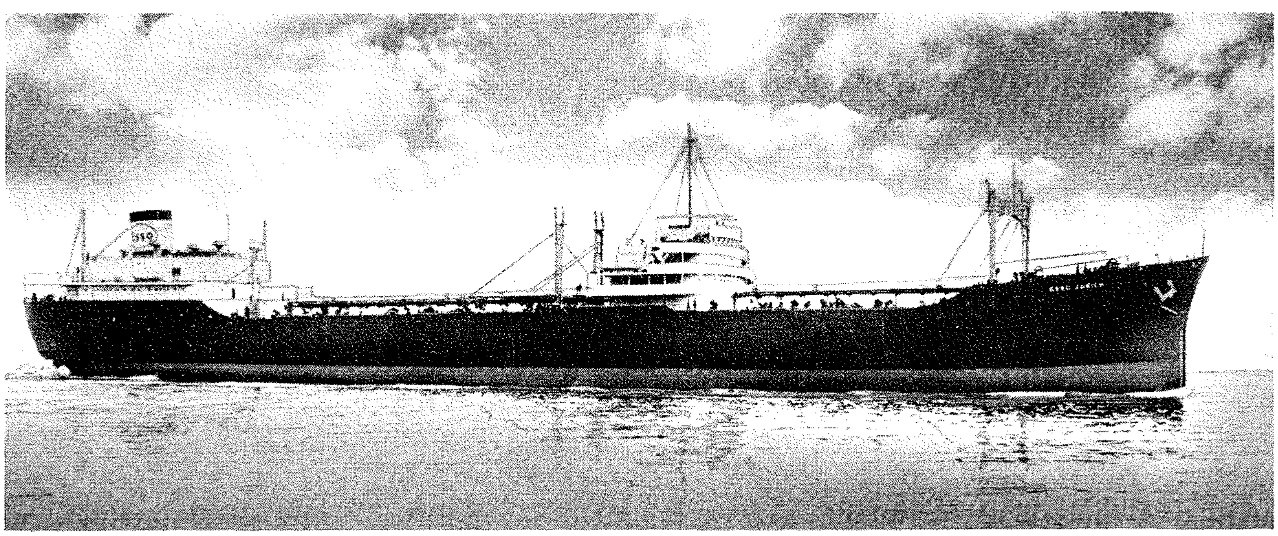


S.S. "Esso Zurich." One of numerous single-screw tankers built in 1948-49, equipped with Kingsbury Thrust Bearings.

FOREWORD

This Bulletin deals with Kingsbury Propeller Thrust Bearings for ships. The principal designs are described, also methods of mounting which have been found appropriate to various common types of propelling machinery.

The basic Kingsbury principle is explained on an introductory page.

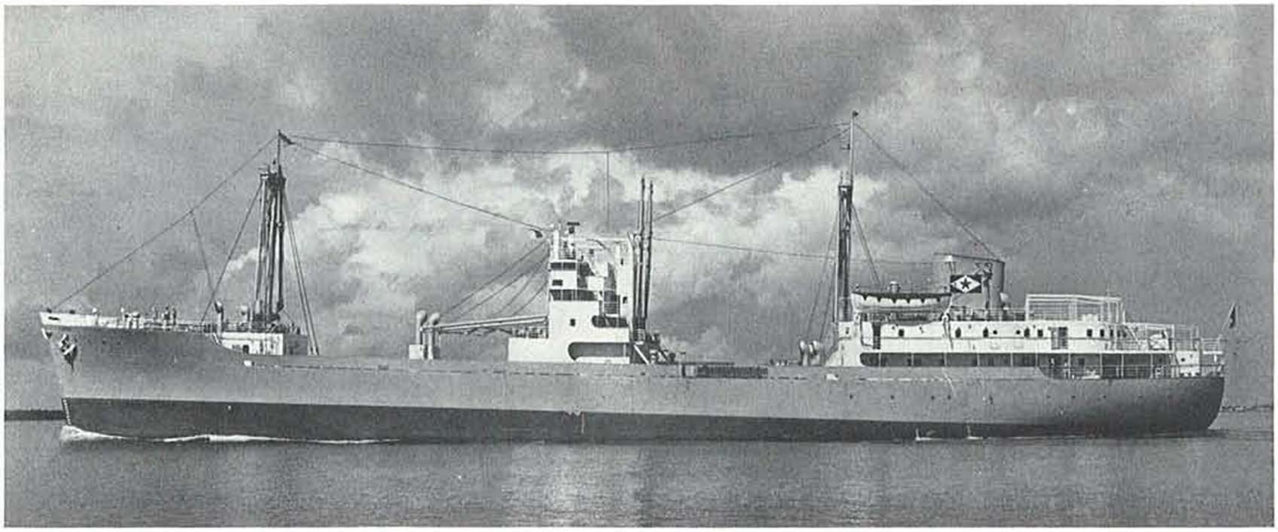
Matters of bearing selection, lubrication and cooling, foundation design, installation and operation are considered in some detail.

Kingsbury Line Shaft Bearings are included, with emphasis on their complete dependability, resulting from automatic lubrication and self-aligning construction.

Kingsbury Thrust Meters, permitting accurate measurement of propeller thrust, are described and illustrated.

Separate booklets, furnished on request, give full data on dimensions and load capacities of standard bearings.

Marine uses of Kingsbury Bearings not covered herein are mainly propulsion turbines and important auxiliaries. Those auxiliaries include the boiler feed pumps, circulating, hot well and booster pumps, and lighting sets. All those applications are covered by other literature, available on request.



Motorship "Algerie," Tampa Shipbuilding Company, builders, equipped with Kingsbury Bearings on twin shafts.

How It Started

The idea of "floating" a bearing load on a film of oil, which separates the bearing surfaces and eliminates wear, grew out of experiments made by Beauchamp Tower in England, the results of which were published in 1883 and 1884. On reading Tower's findings, it occurred to Osborne Reynolds that, if the oil films were sufficiently thick, Tower's results would be deducible from the equations of hydrodynamics. This led to Reynolds' paper, published in 1886, in which the idea of the wedge-shaped oil film was advanced.

Working with Professor Thurston in 1889, Albert Kingsbury, then an undergraduate at Cornell University, discovered that under comparable conditions differences in bearing metals had no effect on journal friction, if the bearings were geometrically identical and copiously lubricated. Later he built an air-lubricated bearing, and, in addition to determining film pressures, was able actually to measure the tapering film of air supporting the weight of the rotating journal. This work was published in 1897 and led to Kingsbury's invention of the sector shaped pivoted-shoe thrust bearing, which was tested at the University of New Hampshire during the years 1898-1899.

Working on the other side of the world, an Australian inventor, A. G. M. Michell, arrived independently at the same conclusions and a very similar thrust bearing design, patented in 1905.

Acceptance of the pivoted-shoe thrust bearing in the United States was rapid, and there followed a spectacular growth in the size and power of vertical hydroelectric generators, until then handicapped by lack of thrust bearings capable of supporting loads of the order of hundreds of tons. The U. S. Navy immediately adopted Kingsbury thrust bearings for the combat ships used in the first World War. Their acceptance by the Merchant Marine followed; and, during World War II, practically all vessels, both combat and merchant, were equipped with Kingsbury bearings.

One effect of the Kingsbury bearing was to make obsolete the always-troublesome horseshoe propeller shaft thrust, till then tolerated because there was nothing better. That type of bearing, with its multiple collars, was inherently unreliable because the "horseshoes," with their bearing surfaces parallel to the collar faces, gave no opportunity for the oil films to assume the natural taper which would permit self-renewal under load. Since the load-carrying capacity of parallel films is small, unit pressures had to be kept very low, resulting in large, cumbersome bearings. Even so, there frequently was actual metallic rubbing and heating. In comparative tests between a horseshoe bearing and a Kingsbury bearing under equal loads and speeds, the single-collar Kingsbury bearing carried the load with one-tenth the bearing area and one-tenth the friction.

S.S. "Independence." Kingsbury Thrust Bearings in main gear housings of this and sister ship "Constitution."
Courtesy American Export Lines.

Basic Kingsbury Principle

Every Kingsbury Thrust Bearing has a single *thrust collar*, which acts against stationary segments called *shoes*, usually steel or bronze castings, babbitt-faced. Each "shoe" contains a hardened pivot, which bears against a hardened backing surface.

When collar and shoes are flooded with oil, tilting of the shoes forms wedge-shaped films, slightly thicker on the entering side. Though the actual tilt is small, oil adhering to the collar keeps the films full. As long as the shaft turns, oil is supplied. There is then no metallic contact between shoes and collar, hence no wear, and the only friction is that of oil film shear.

Due to this principle of automatic lubrication, Kingsbury bearings carry heavy loads on small areas. When loads and speeds are sufficiently great it becomes necessary to cool the oil heated by shear between the shoes and collar.

However widely the conditions of size, load, speed and mounting may vary, the combination of single collar, tilting shoes and cooled, circulating oil is found in all Kingsbury Thrust Bearings. These bearings might indeed be described as a system of a few basic elements, so designed as to "float" the load on moving films of oil constantly renewed.

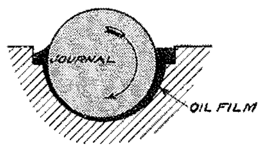


Figure 1
The wedge film in journal bearings.

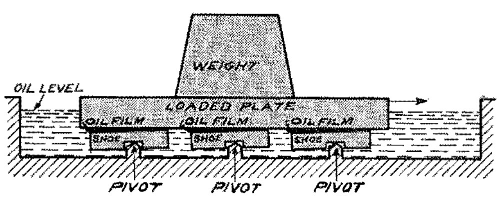


Figure 2
Action of pivoted shoes supporting a loaded movable plate in an oil bath.

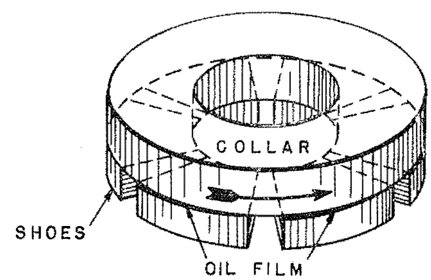
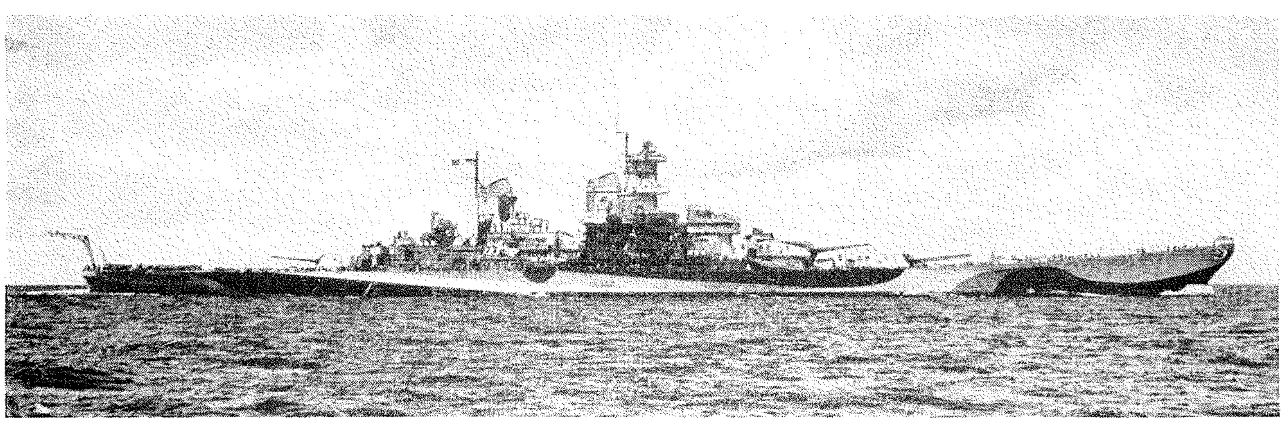


Figure 3
Rotating thrust collar, bearing against pivoted shoes. (Flooded lubrication is assumed.)



U.S.S. "MISSOURI." Displacement 45,000 tons. Each of the five battleships of this class is equipped with thirty-six Kingsbury Thrust Bearings. Four of these, of size 49 inches, are on the propeller shafts.

Underwood-Stratton. Official U. S. Navy photo.

Application of Kingsbury Thrust Bearings To Principal Types of Drive

The Naval Architect selects the type of propelling machinery without much regard to the thrust bearing. His choice, however, largely determines the location and type of the Kingsbury Bearing.

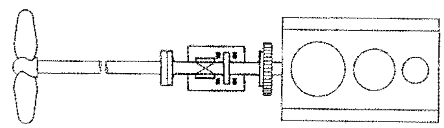
Seven major types of drive are considered in the following paragraphs. Figures 4 to 9 show these types, with alternate arrangements for each type.

1. Steam Reciprocating Engine.—The thrust bearing is always located abaft the engine.

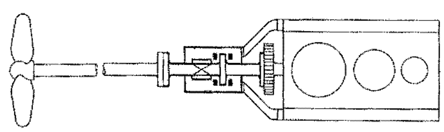
2. Direct-Connected Diesel Engine.—The thrust bearing is usually located abaft the engine, Figures 5 and 13 to 15, etc.

3. Geared Diesel Engine.—The thrust bearing may be placed aft of the gear, or built into the gear case. Figures 6, 12 and 24.

4. Diesel-Electric Drive.—The thrust bearing may be mounted on a separate foundation aft of the electric motor, or it may be built into the motor, sometimes with its own journal bearing supporting one end of the rotor. Figures 8 and 29.

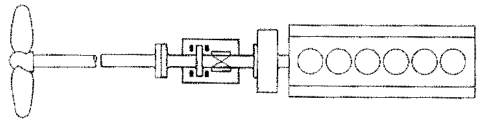


(a) Pedestal type thrust bearing.

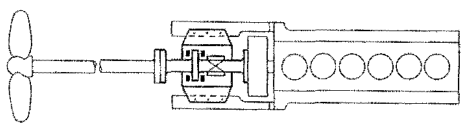


(b) Thrust bearing connected to engine by struts.

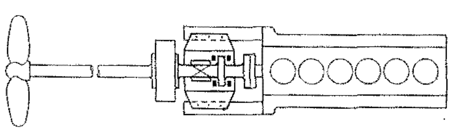
Figure 4
Steam Engine.



(a) Pedestal type bearing aft of flywheel.

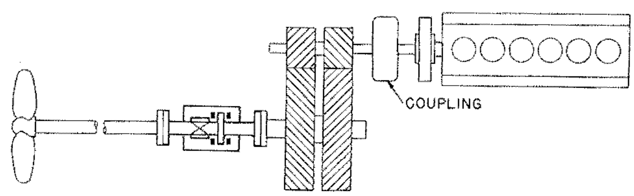


(b) High-flange type bearing aft of flywheel.

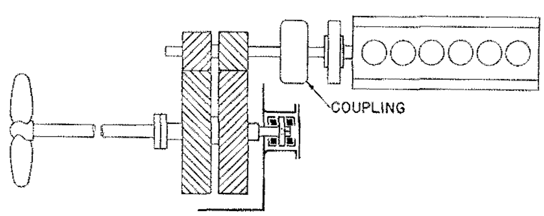


(c) High-flange type bearing between engine and flywheel.

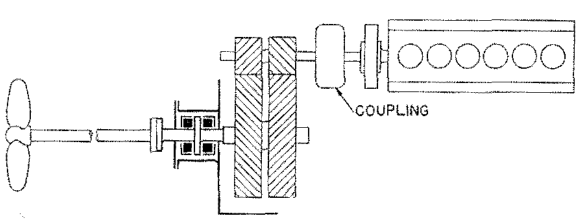
Figure 5
Diesel Engine.



(a) Pedestal type bearing.



(b) Bearing in forward end of gear case.



(c) Bearing in aft end of gear case.

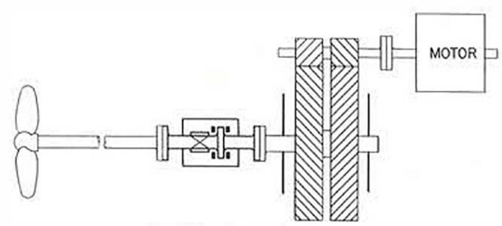
Figure 6
Diesel Engine, geared drive.

5. ***Diesel-Electric, Geared.***—The alternative bearing arrangements are similar to those for geared Diesel drives. Figure 7.

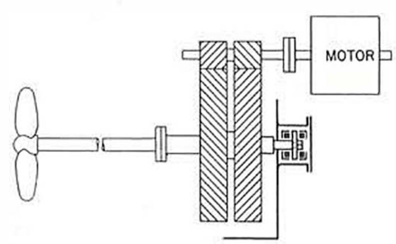
6. ***Turbine-Electric Drive.***—The alternative bearing arrangements are similar to those for Diesel-electric drives.

7. ***Geared Turbines.***—The usual location for the thrust bearing is on the forward end of the bull gear shaft, in a housing forming part of the gear housing. However, the thrust bearing may be mounted in the after end of the gear housing; or it may be a separate unit mounted aft of the gears in any preferred location on the line shaft. Figures 9, 25-28.

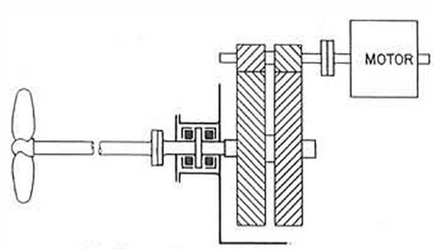
With the general arrangement settled, for any given type of drive, the next step is to select the most suitable type of bearing. The two principal types are Two-Shoe and Six-Shoe. These types and certain variants are described in the following pages, and the principles governing their selection are discussed.



(a) Pedestal type bearing.

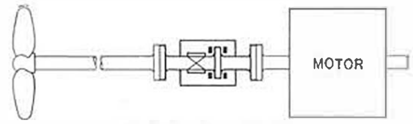


(b) Bearing in forward end of gear case.

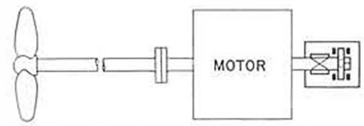


(c) Bearing in after end of gear case.

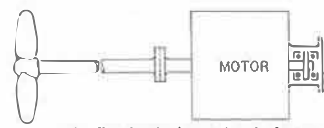
Figure 7
Diesel-Electric, geared drive.



(a) Pedestal type bearing.

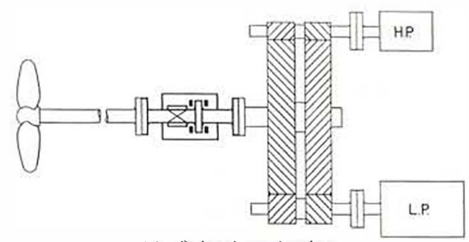


(b) Pedestal type bearing at forward end of motor.

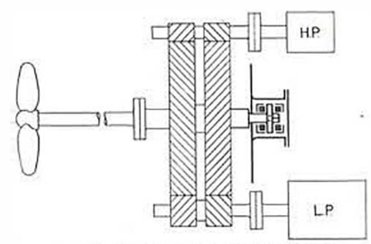


(c) Bearing in forward end of motor.

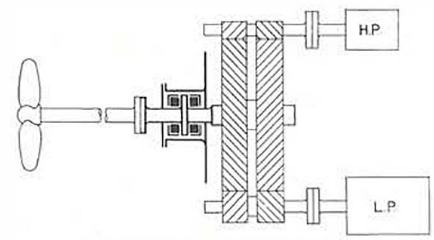
Figure 8
Diesel-Electric drive.



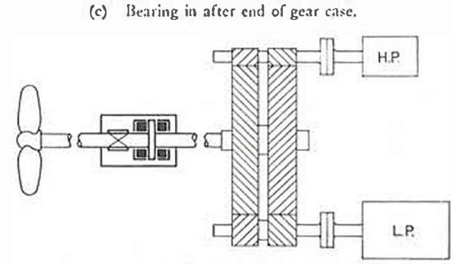
(a) Pedestal type bearing.



(b) Bearing in forward end of gear case.



(c) Bearing in after end of gear case.



(d) Pedestal type bearing aft of machinery space.

Figure 9
Geared Turbine drive.

Suiting the Bearing Type to the Location

Two principal types of thrust bearings account for perhaps 95 per cent of all Kingsbury installations on marine propeller shafts—*Two-Shoe* and *Six-Shoe*. The choice of type is influenced chiefly by bearing location, thrust load, shaft diameter and shaft speed.

Two-shoe bearings are a quite distinct group. In standard forms they have a self-contained housing, usually with automatic lubrication, and a journal bearing adjoining the thrust collar. Standard housings are of pedestal type, but some modifications are available. Two-shoe bearings are most often employed aft of the drive. They are the usual choice for direct engine drive and medium service generally, for which their extreme simplicity commends them.

Six-shoe bearings usually are built into the forward end of a reduction gear housing, on the end of the bull gear shaft. They may also be mounted at the forward end of direct-drive electric motors. However, they are sometimes mounted at the after end of a gear housing; or they may be still farther aft, in self-contained pedestal housings, located anywhere on the length of the line shaft.

These two types are described in some detail on pages 18 to 26. Since the type of drive usually governs the location of the thrust bearing, and both the magnitude of the thrust load and the location affect the choice of bearing type, the following discussion is based on the bearing location.

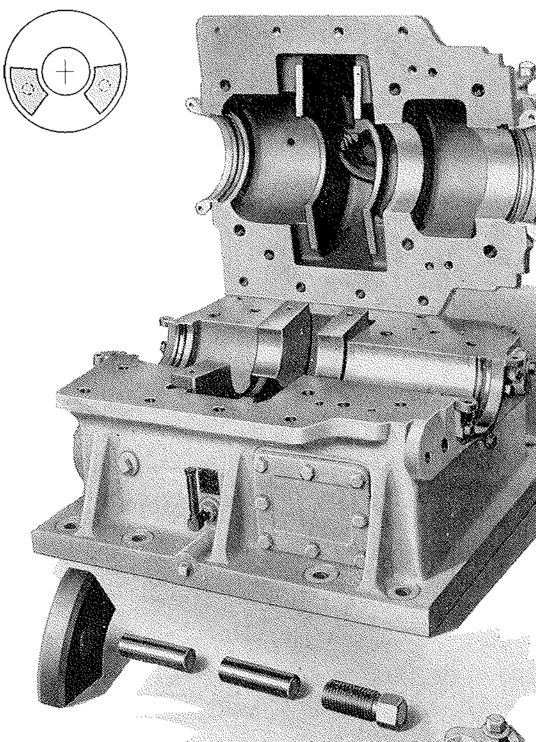


Figure 10
Two-shoe thrust bearing with cover lifted. One shoe with thrust pins, jack screw, and locking wrench shown separately. Shaft with thrust collar not shown.

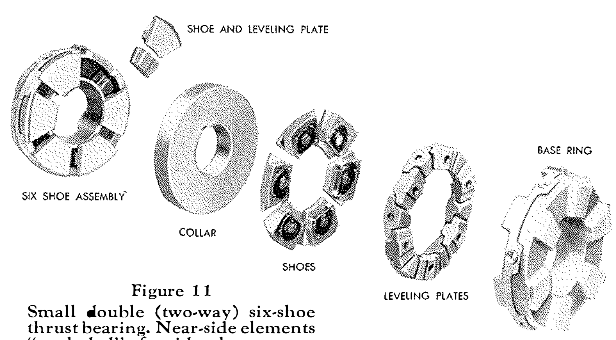


Figure 11
Small double (two-way) six-shoe thrust bearing. Near-side elements "exploded"; far-side elements assembled.

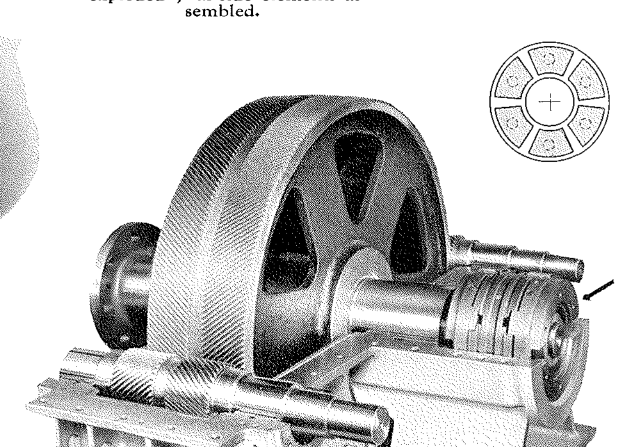


Figure 12
Typical reduction gears for geared Diesel drive. Six-shoe bearing on forward end of main gear shaft.

Courtesy Universal Gear Corp.

Location Abaft the Drive

Commonest example is with a direct-drive steam or Diesel engine. The thrust requirement is usually satisfied by a two-shoe bearing, with thrust collar diameter about twice the diameter of the shaft. The housing is usually of pedestal type. It is preferably bolted on an extension of the engine bedplate; but it may be attached to the engine base by struts, or separately mounted farther aft on a rigid foundation. Generally lubrication is automatic, from an oil bath in the housing base.

Of two-shoe pedestal type bearings there are three styles available, differing chiefly in minor points, described on pages 18 to 23. In one style, high flanges are optional in certain sizes for mounting on a bedplate at about the height of the shaft center (Figure 45).

Occasionally the aft location is preferred with a direct-drive electric motor.

In higher powers, a two-shoe bearing might require too large a thrust collar. In that case, a somewhat smaller collar may be used with a six-shoe bearing of pedestal type. See Figures 56 to 59. When a six-shoe bearing is placed abaft the drive, usually it is because some circumstance makes it undesirable to locate it in the gear housing. See page 26.

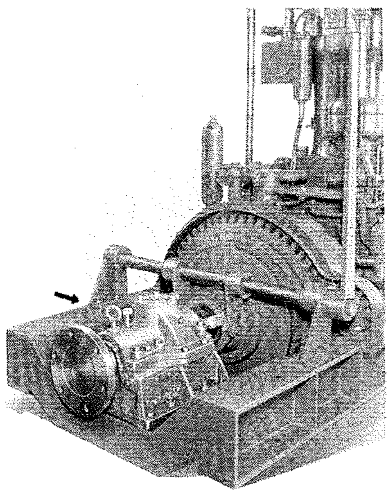


Figure 13
Standard two-shoe thrust bearing, pedestal type, on base of Diesel engine.
Courtesy Union Diesel Engine Co.

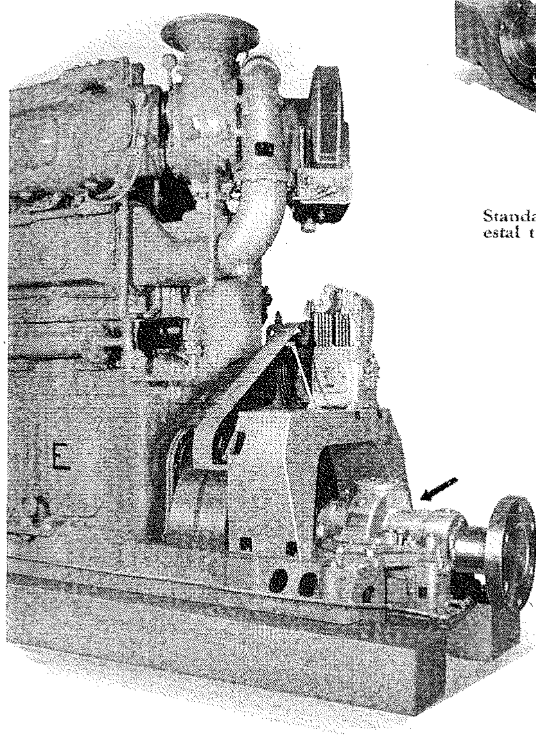


Figure 14
Special two-shoe high-flange thrust bearing on base of Diesel engine.
Courtesy Enterprise Division, General Metals Corp.

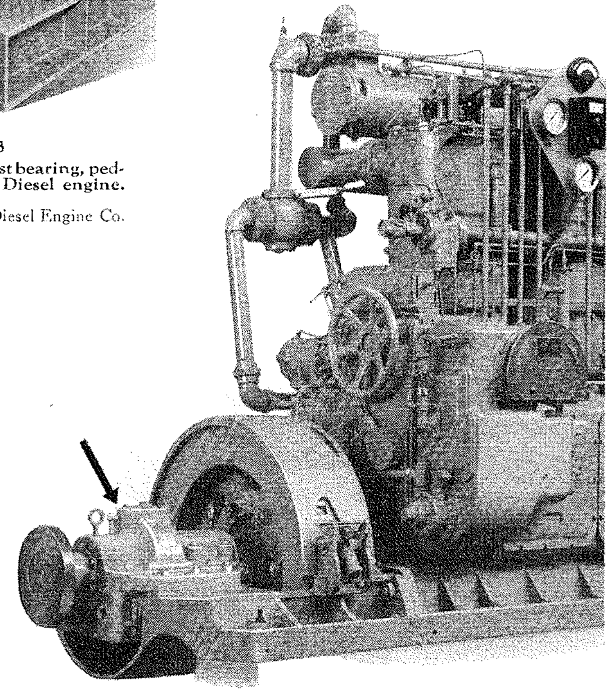


Figure 15
Special two-shoe bearing, high-flange type, on base of Diesel engine.
Courtesy Atlas Diesel Div., National Supply Co.

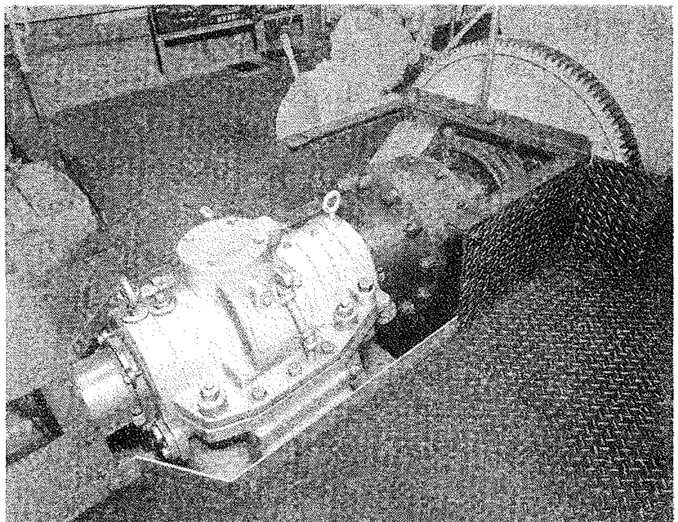


Figure 16
Two-shoe bearing on U.S. Engineers' Tug "Stephen S. Austin."

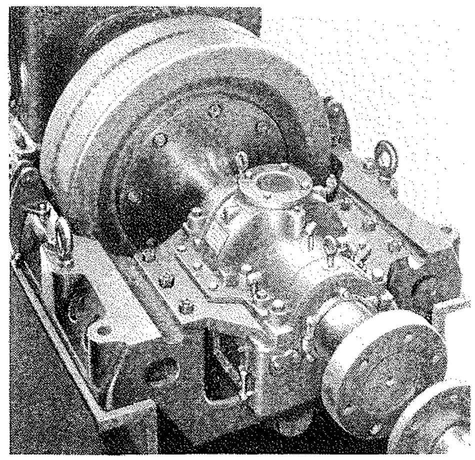


Figure 17
Special GH, high-flange type, on tuna clipper "Paramount."

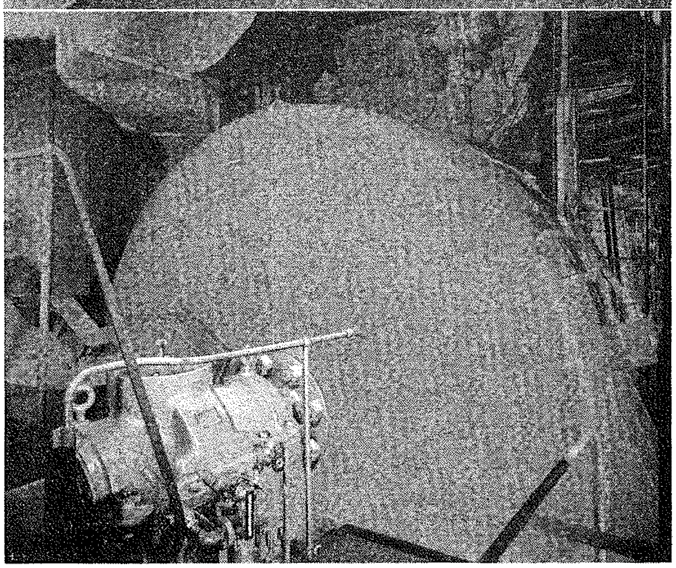


Figure 18
Special three-shoe thrust bearing with Elliott-Bowes Drive on M.S. "Costal Liberator" (see page 23 for description).

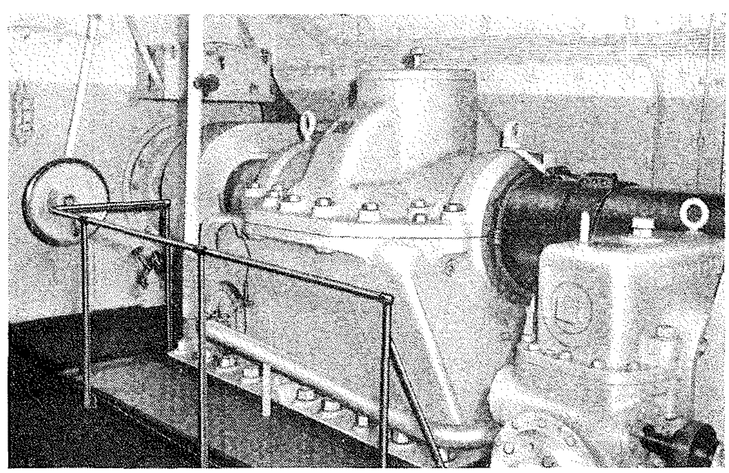


Figure 19
Two-shoe thrust bearing of twin-screw S.S. "President Wilson."

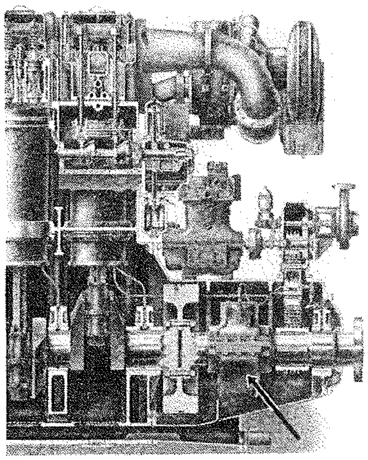


Figure 20
After cylinders of Diesel engine, showing built-in two-shoe thrust bearing.

Courtesy Cooper-Bessemer Corp.

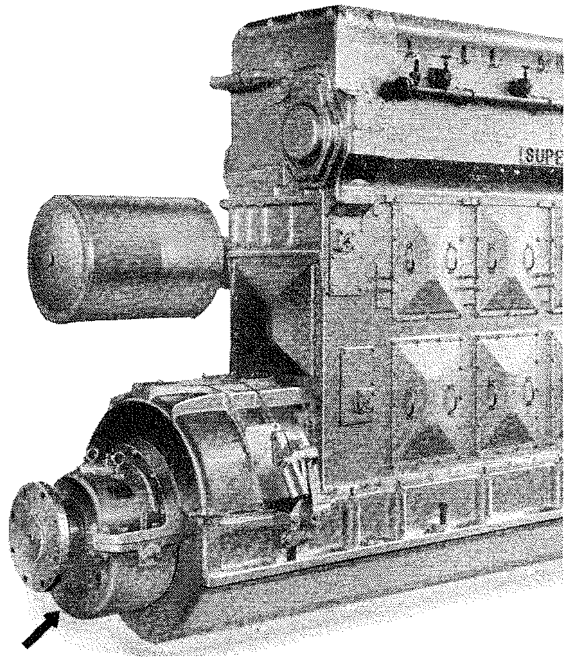


Figure 21
Two-shoe bearing attached to flywheel housing of Diesel engine.

Courtesy Superior Diesel Div., National Supply Co.

Thrust Bearing Attached to Aft End of Crankcase

The compact form of thrust bearing shown in Figure 21 is intended especially for attachment to Diesel engine crankcases. It has self-contained lubrication and cooling.

An alternative arrangement is a thrust bearing (without journal bearing) mounted in a small housing inside the engine crankcase close to the aft end. This is illustrated in Figure 20.

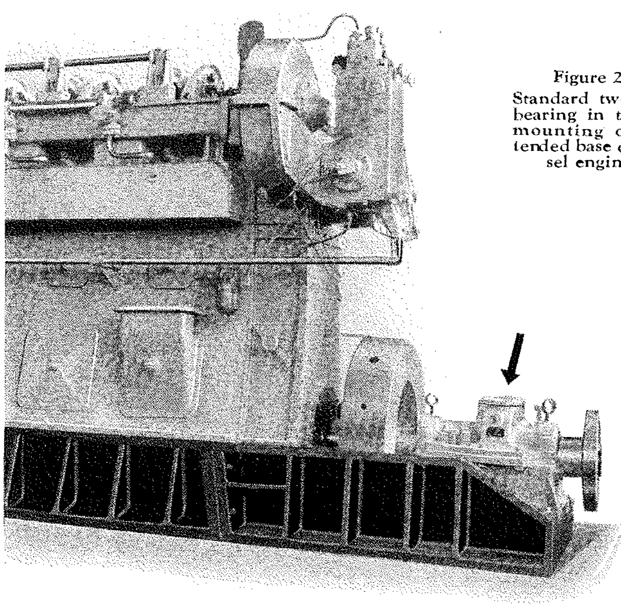


Figure 22
Standard two-shoe bearing in typical mounting on extended base of Diesel engine.

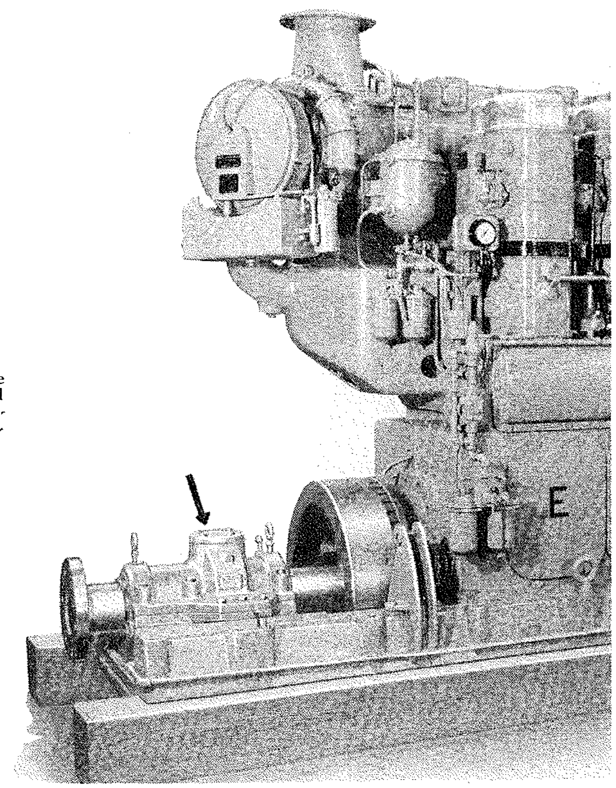


Figure 23
Standard two-shoe bearing on extension of engine base.

Courtesy Enterprise Div., General Metals Corp.

Bearing in Forward End of Reduction Gear Housing (Figures 6b, 7b, 9b, 24-28)

This is the commonest thrust bearing arrangement with turbine drives. Since there are six shoes instead of two, the bearing is adequate even with shaft and collar diameters considerably reduced. The collar is then separate, and is held on the shaft by a key and a massive nut, securely locked. The thrust housing is just large enough to hold the thrust bearing parts, and is constantly flooded with oil. The bearing is lubricated by cooled and filtered oil from the gear system.

A similar arrangement may be used with reduction gearing powered by one or a group of Diesel engines, all working on one set of gears; also with geared Diesel-electric drives.

Bearing in After End of Gear Housing

This arrangement, diagrammed in Figure 9(c), is rarely used, and only when the location at forward end of gear housing cannot be employed. The thrust collar is integral with the shaft. There is sometimes a split facing collar. Or there may be two collars, with a journal bearing between them. All these arrangements are quite special. See Figures 79 and 83.

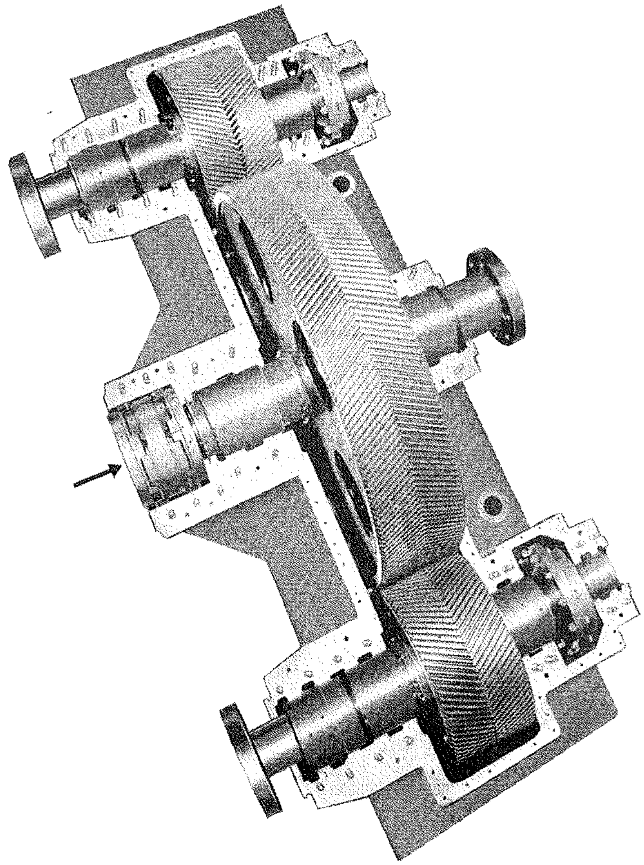


Figure 24
Reduction gear set for marine Diesel engine, with six-shoe thrust bearing.

Courtesy Farrel-Birmingham Co.

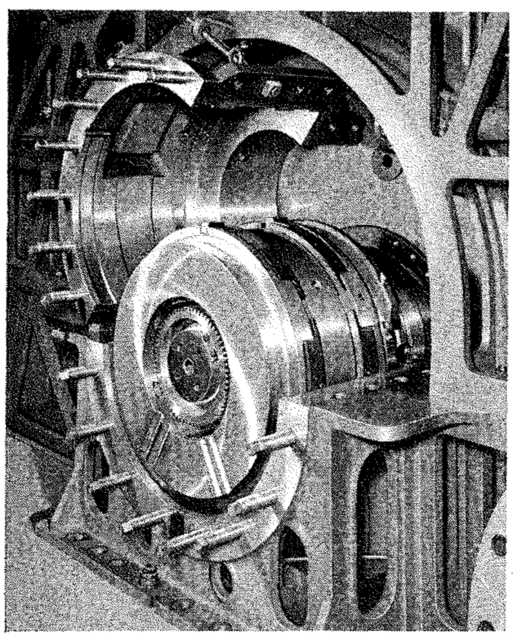


Figure 25
Six-shoe thrust bearing at forward end of reduction gear for steam turbine drive.

Courtesy The Falk Corporation.

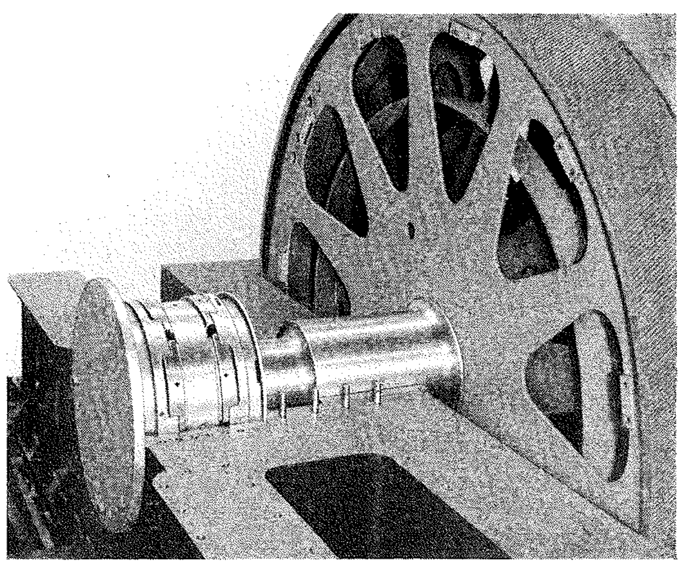


Figure 26
Six-shoe thrust bearing at forward end of reduction gear for steam turbine drive.

Courtesy DeLaval Steam Turbine Co.

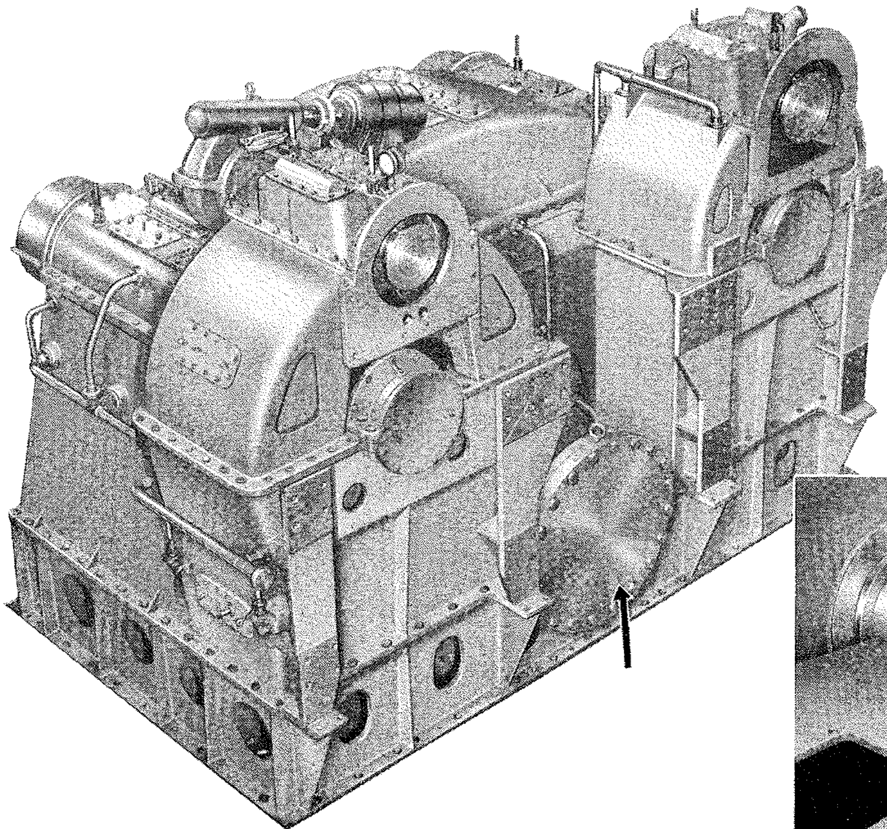
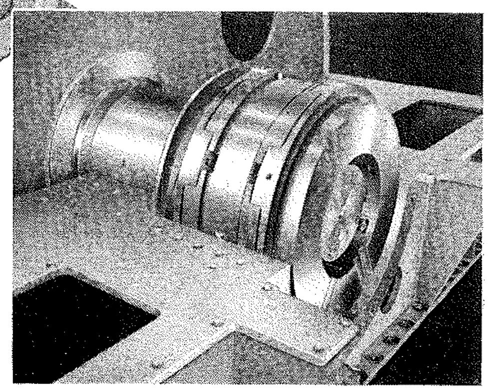
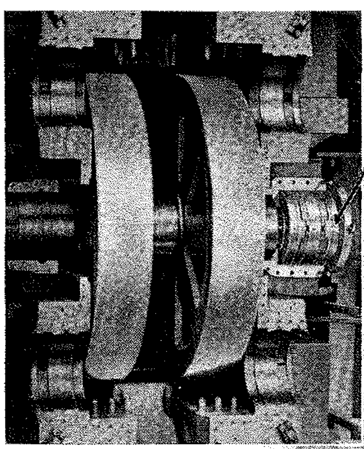


Figure 27
Marine propulsion gear for 12,500 h.p. tankers with turbine drive. Kingsbury six-shoe bearing (see inset) is housed behind the end cover indicated by arrow.

Courtesy General Electric Co.





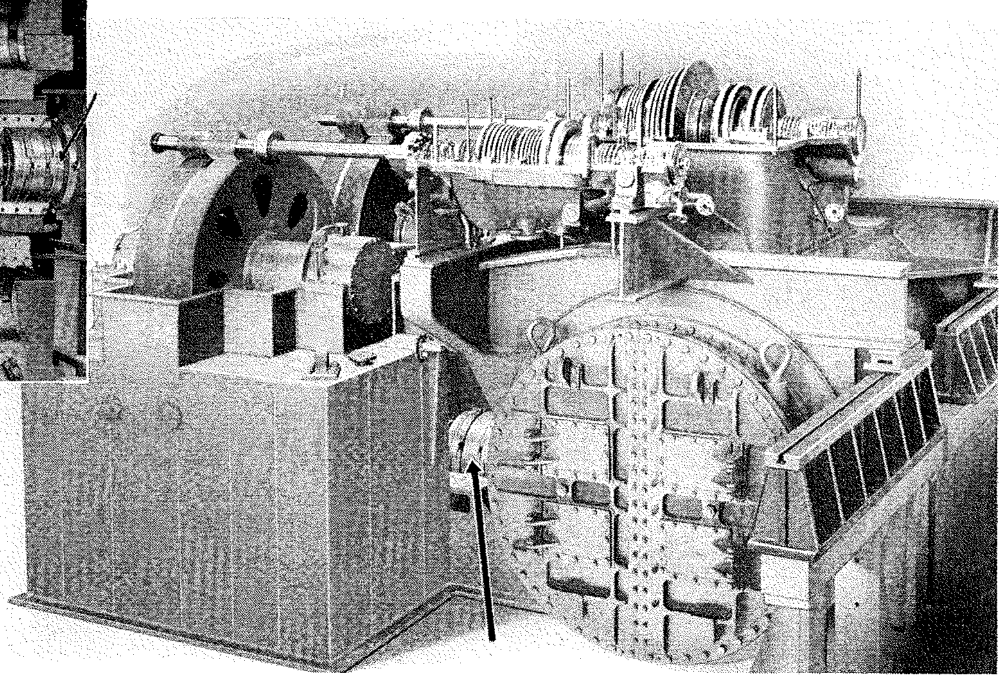


Figure 28
Steam propulsion unit, consisting of high and low pressure turbines, condenser, and reduction gears with six-shoe thrust bearing at forward end of main gearshaft. Inset shows main gears and thrust bearing, looking down.

Courtesy
Westinghouse Electric Corp.

Thrust Bearing at Forward End of Electric M●t●r (Figures 29 to 31)

With direct drive by electric motor, the thrust bearing may be located forward of the motor. Figure 29 shows an electric tugboat motor, in which the thrust-journal bearing supports the rotor.

If the thrust is so great as to make the two-shoe type undesirable, a six-shoe bearing with smaller collar can be used. It may be built into the motor (see Figure 3●) or separately mounted.

Since final choice of bearing type and size may depend partly on the characteristics of the bearings themselves, the various two-shoe and six-shoe types are next described.

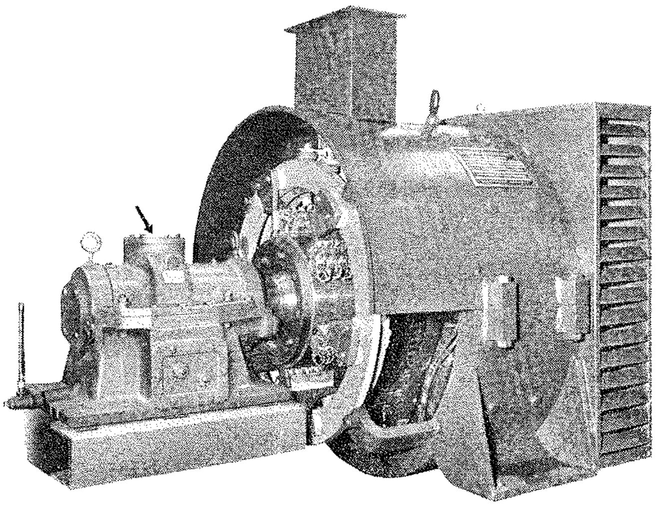


Figure 29
Two-shoe bearing at forward end of propulsion motor for tugboat.

Courtesy General Electric Co.

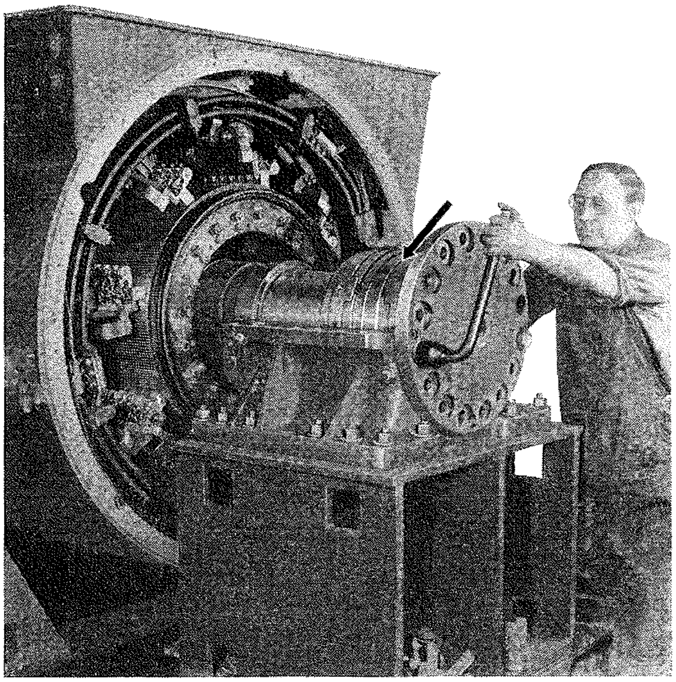


Figure 30
Six-shoe bearing at forward end of 39●0 h.p. propulsion motor.

C●urtesy Allis-Chalmers Mfg. Co.

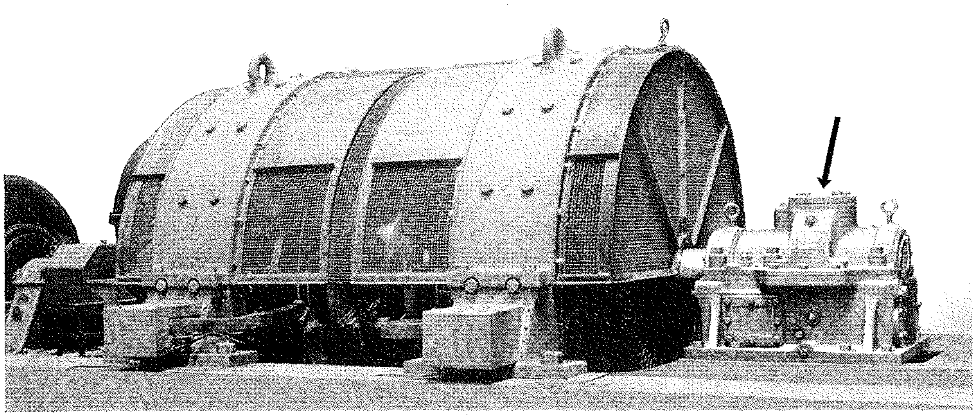


Figure 31
Two-shoe thrust bearing at forward end of electric motor drive.

Courtesy Westinghouse Electric Corp.

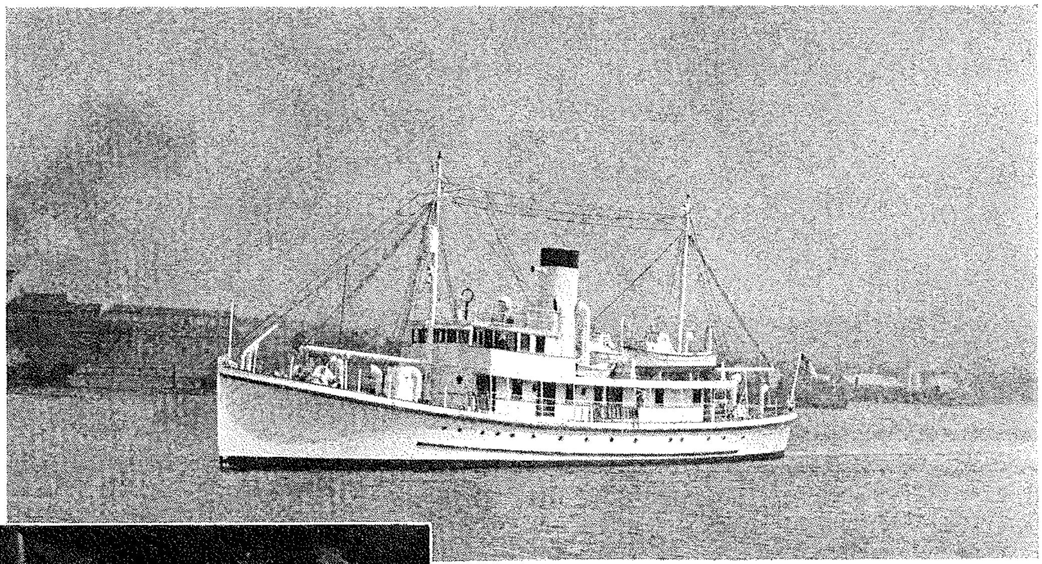


Figure 32
Yacht "Stranger," equipped with two-shoe bearing shown at left.

Photograph by Raymond J. Krantz.

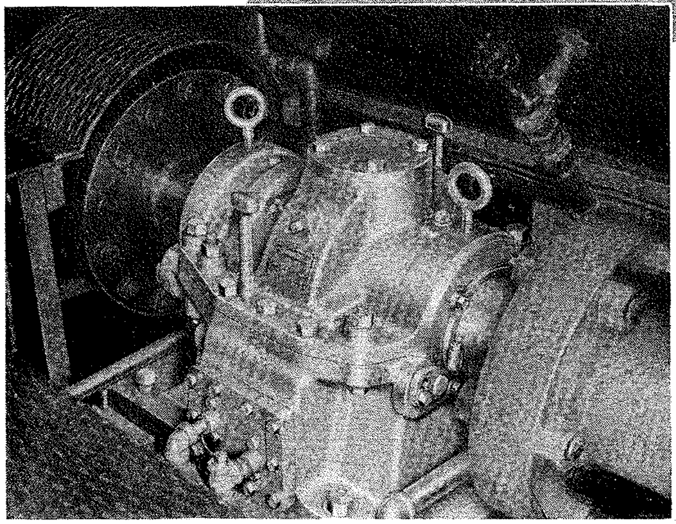


Figure 33
Two-shoe thrust bearing on yacht "Stranger."

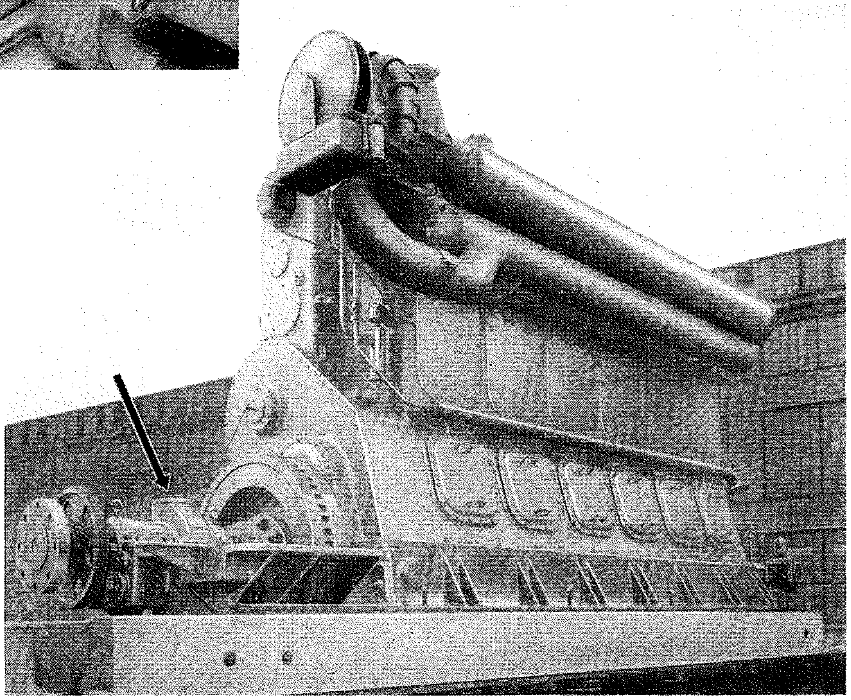
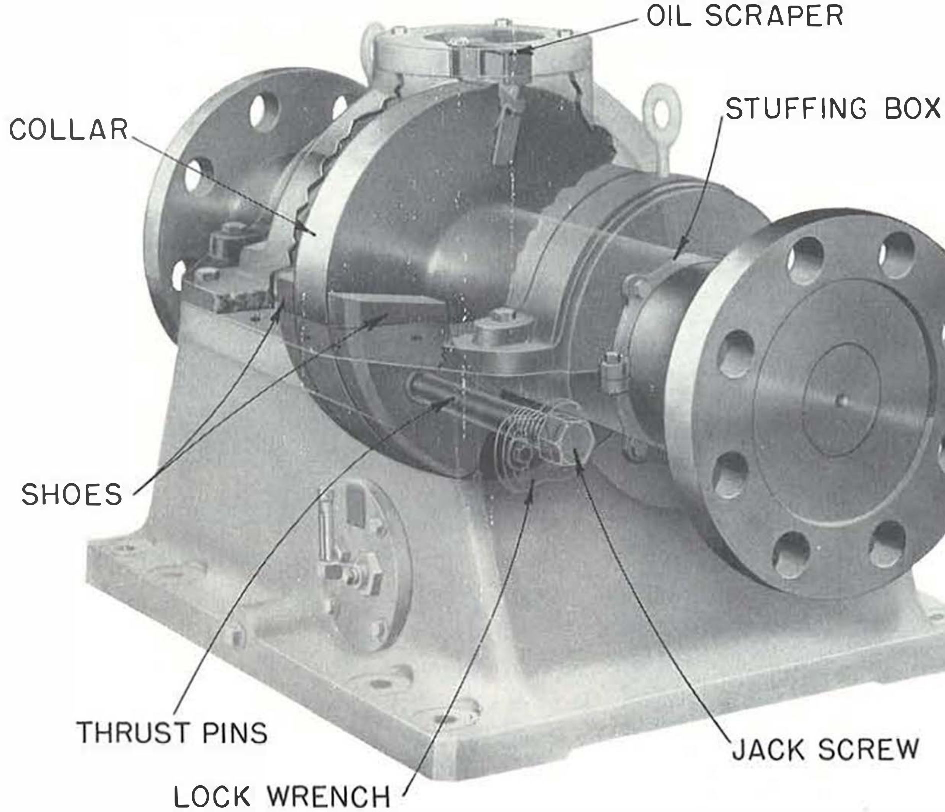


Figure 34
Special high-flange thrust bearing mounted on extension of Diesel engine base. One of six supplied to twin-screw trawlers for the French Government.

Courtesy Baldwin-Southwark Div. of Baldwin-Lima-Hamilton Corp.

Two-Shoe Adjustable Thrust Bearings



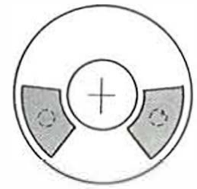


Figure 35
Typical two-shoe bearing in phantom, with part of housing cover broken away. Principal parts are indicated.

The Two-Shoe bearing is the type generally found adequate for use with direct drive engines. It comprises a split *housing*, a *journal bearing*, and a short thrust shaft with an integrally forged *collar* about twice as large as the shaft. Two pivoted *shoes* bear against each face of the thrust collar, and rest loosely in pockets in the housing. They are free to adapt themselves to the position of the collar and to the slight angles created by the wedge-shaped oil films.

The shoes are of cast steel, faced with babbitt metal with radial edges slightly rounded, and scraped to a surface plate. The collar is machined flat and finished very smooth. Each shoe has in its back a hardened insert, or *shoe support*, with a rounded pivot. The pivot bears, usually through a *thrust pin*, against a massive *jack screw*. Each shoe has its own jack screw; and the jack screws can be accurately adjusted and securely locked, so that the thrust load is equally divided between the two shoes and suitable end play is provided. The adjustment is made by hand: the two-shoe bearing contrasts in this respect with the six-shoe bearing described later, in which the load is equalized automatically between the shoes.

The shoes and jack screws are located in the lower housing, a little below the shaft center, so that they are not disturbed by lifting the housing cap for inspection or adjustment.

The standard journal bearing is babbitt-lined like the shoes.

Both thrust and journal elements are flooded with oil from the bath in the housing

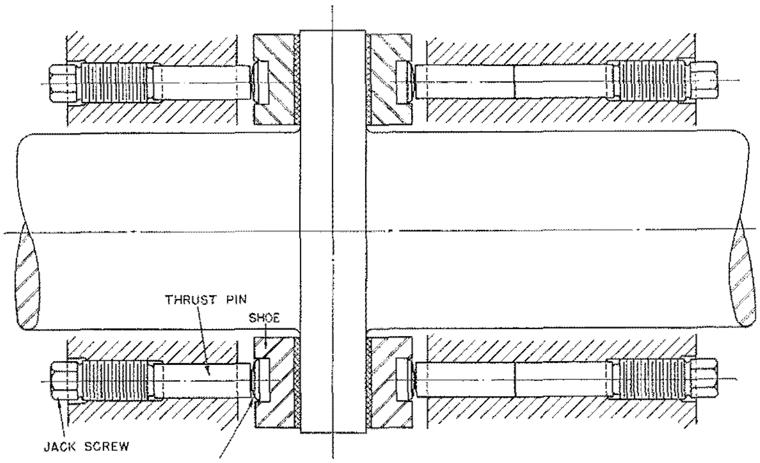


Figure 36
Plan section of two-shoe bearing, showing thrust collar and shaft, shoes, thrust pins and jack screws.

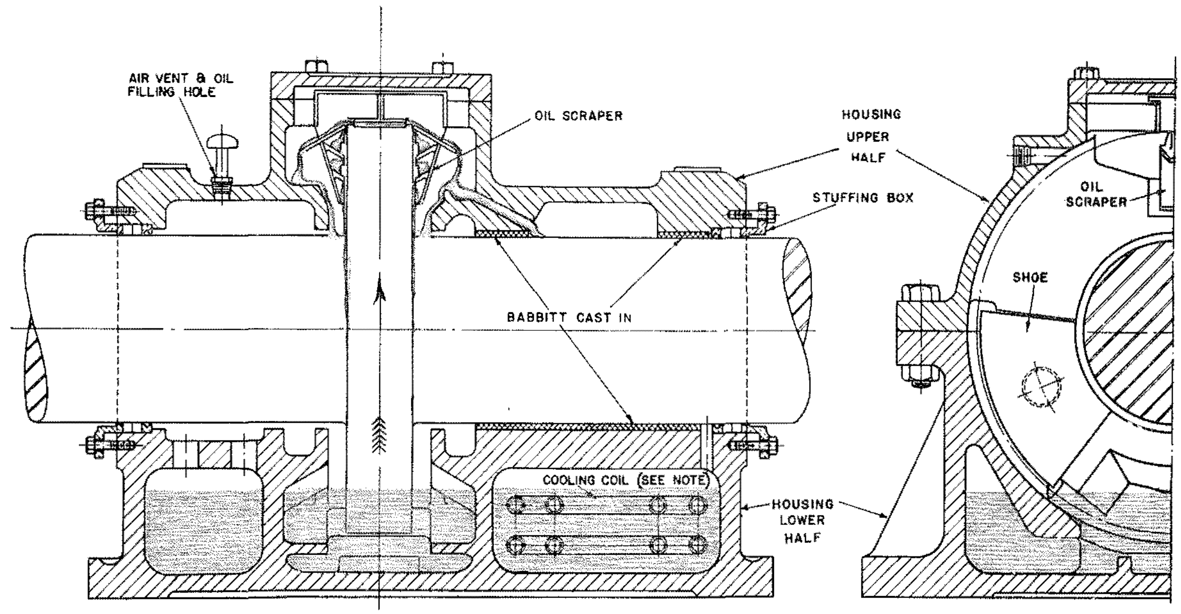


Figure 37
Vertical section of two-shoe bearing, showing oil picked up by thrust collar and distributed to collar faces and journal bearing. The coil is used when the operating speed requires it.

base. The thrust collar dips into the oil bath and carries oil directly to the shoes on the "up" side. It also carries oil up to the bronze *oil scraper* riding on the collar rim. From the scraper, oil flows to the shoes on the "down" side, and also through passages to the journal bearing. Thus all shoes and the journal are lubricated by generous streams of oil.

At moderate speeds the heat generated by oil film shear is carried away by the surrounding air and the foundation. At higher speeds, supplementary cooling is afforded by a water coil immersed in the bath. Because of this (at least) partial reliance on air cooling, the oil used is heavier than in bearings wholly water-cooled.

The bottom flange of the housing is machined flat underside. For details of mounting on the foundations, see "Foundation and Alignment," page 33.

If preferred, any two-shoe bearing may be connected to a general oil-circulating and cooling system. However, such an arrangement should be discussed before ordering.

Standard Two-Shoe Bearing Designs

Standard two-shoe bearings have pedestal-type housings, with the exception of one design, Style GC, which has a circular end flange designed for attachment to the end of a Diesel engine.

The pedestal-type bearings are of three styles, GH, GF and GK, which are similar in most respects, but differ in detail. The main difference concerns the journal bearing arrangement.

In Style GH there is no journal bearing shell, the babbitt being cast directly in the housing. In the upper half, the babbitt is in two end strips, with a large oil pocket between them. The oil scraper on the thrust collar sends a stream of oil into this pocket, thereby assuring ample lubrication.

The GF bearing is intended primarily for use with dredge pumps and is therefore designed to endure upward or possible whirling loads. It has a split full journal bearing shell. Instead of an oil pocket, the shell has oil grooves on both sides at the joint, and receives a stream of oil from the scraper.

The GK bearing has a removable shell in the lower half. The upper half has two end strips of babbitt cast in the housing, the same as for the GH style.

The standard end closures of Style GH are stuffing boxes. However, these stuffing boxes work against no internal pressure, and they are therefore replaced in Style GK by simple felt rings. This contributes to the short over-all length in way of the shaft, which is a feature of Style GK. Style GF bearings aboard ship may have either form of closure, although the stuffing box is considered standard. In dredge pump service, special closures are used to exclude water squirting from the pump stuffing box.

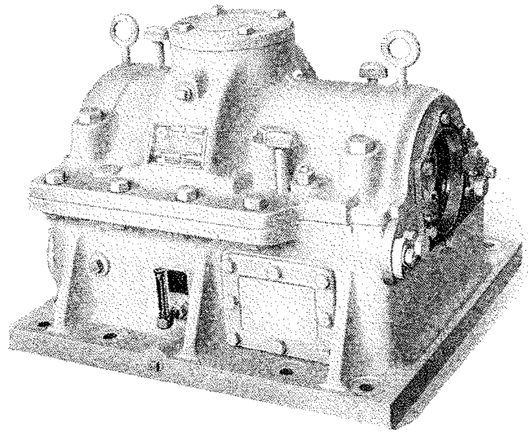


Figure 38
Style GH two-shoe thrust bearing.

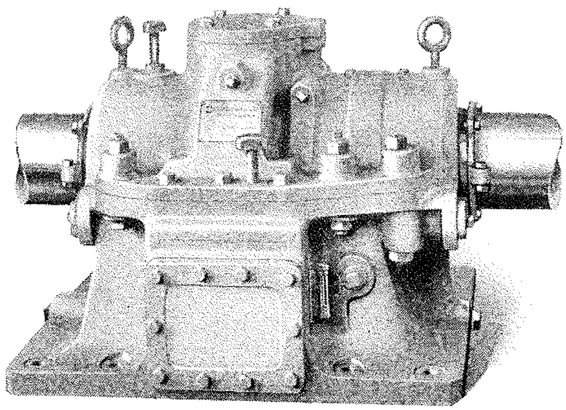


Figure 39
Style GF two-shoe thrust bearing.

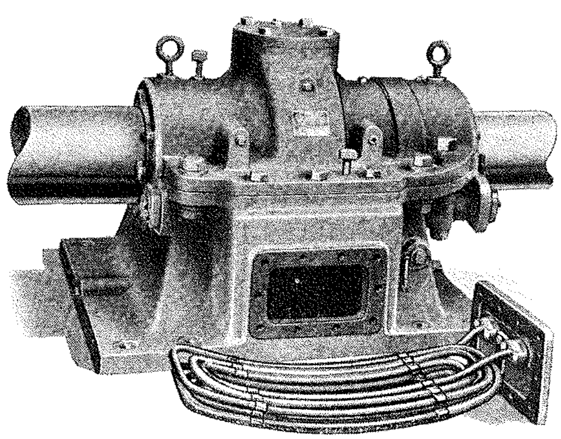


Figure 40
Style GF thrust bearing, showing shape of cooling coil when used.

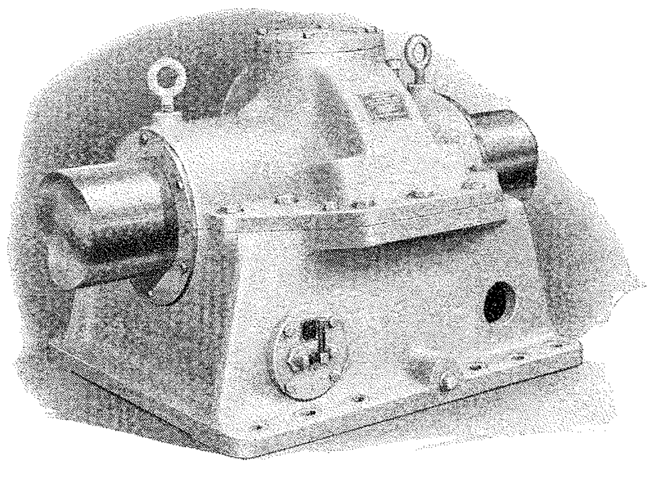


Figure 41
Style GK thrust bearing.

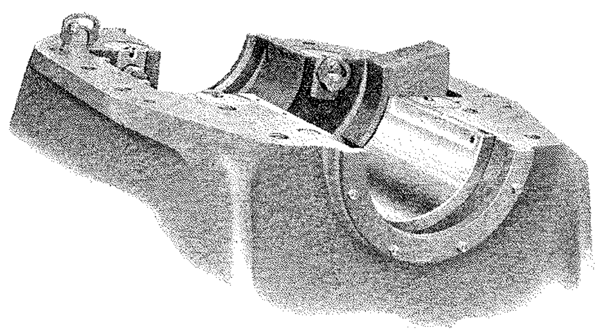


Figure 42
Style GK with cover lifted, showing internal jack screws.

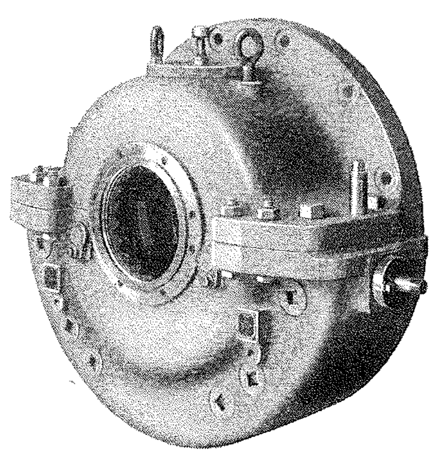


Figure 43
Style GC compact thrust bearing for attachment to Diesel engine crankcase or flywheel case.

The jack screws, for adjusting the thrust shoes, have external heads in Styles GH and GF. In Style GK the heads are internal and are reached by raising the upper half of the housing.

Cooling coils, when used with GH and GF bearings, are of the general form illustrated in Figure 40. Coils for GK bearings are helical in shape, one or two being used according to requirements.

Sometimes it is desirable to modify the standard GH bearing by locating the supporting flange just below the level of the shaft center, instead of at the base of the housing. This "high flange" design is illustrated in Figures 14, 15, 17 and 45.

In general, it is not necessary for the customer to specify any particular "style" of pedestal type two-shoe bearing. We can make the most suitable selection if we have the data listed on page 47.

Style GC (Circular end flange)

This is a compact bearing, combining a two-shoe thrust with journal bearing, designed for Diesel engines. It is bolted to the after end of the crankcase or flywheel case. The journal bearing can carry a share of the flywheel weight.

The shoes are mounted in the lower housing, with jack screws bearing directly against them; and the short journal bearing is just forward of the collar. Being self-aligning, the journal bearing has maximum capacity for its area.

The bearing uses heavy Diesel lubricating oil, and delivers it to its own working parts by a scraper riding on the collar. A cooling coil assures the proper working temperature, since there is usually no communication between the thrust bearing and the crankcase interior.

This bearing is illustrated in Figures 21 and 43.

Special Two- and Three-Shoe Bearings

As previously mentioned, Style GH can be furnished in certain sizes with raised side flanges for attachment to bedplate girders at or near the level of the shaft center, instead of the usual pedestal base flanges. See Figure 45. The internal features are the same as in the standard bearings.

Special lower housings for direct attachment to the engine crankcase or extensions thereof are shown in Figures 44 and 46.

A large two-shoe bearing with a short journal bearing in each end is shown in Figure 47. Many of these have been furnished for large Diesel tankers.

During World War II about 2500 Liberty Ships were equipped with special two-shoe Kingsbury bearings, Figure 48. The "Libertys" had two journal bearings, one each side of the thrust collar; but this same general design is regularly furnished with a single journal bearing, agreeing with our standard practice.

Occasionally the thrust load may be a little greater than can be handled with a two-shoe bearing of the preferred size. Instead of going to the next larger two-shoe bearing, it

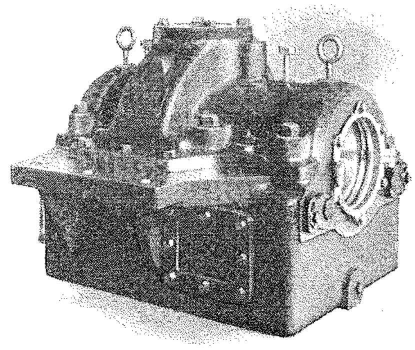


Figure 45
Special two-shoe bearing, high-flange type.

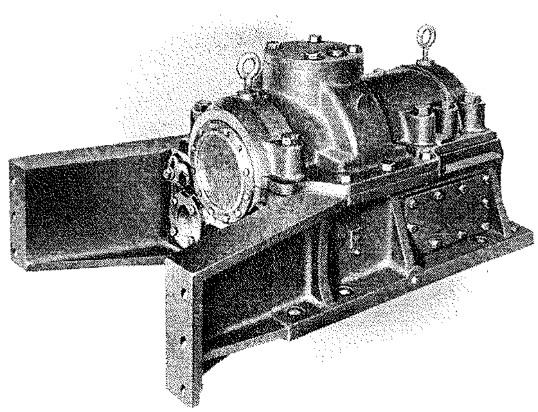


Figure 46
Special two-shoe bearing with struts for direct attachment to engine base.

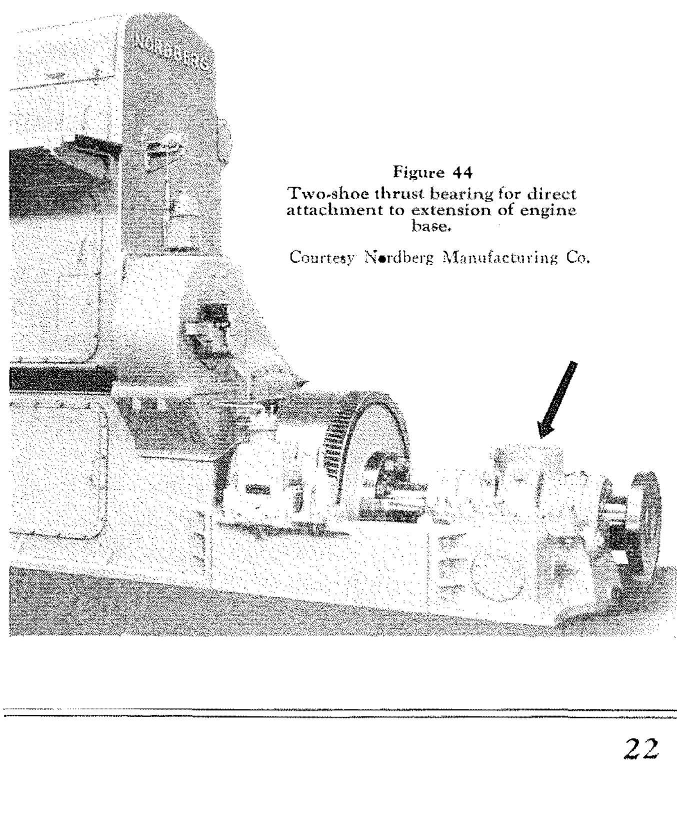


Figure 44
Two-shoe thrust bearing for direct attachment to extension of engine base.

Courtesy Nordberg Manufacturing Co.

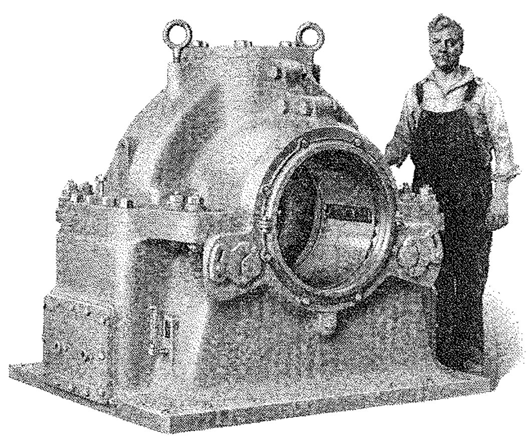


Figure 47
Special large two-shoe bearing with two short journal bearings. Many of these have been supplied to large Diesel-powered tankers.

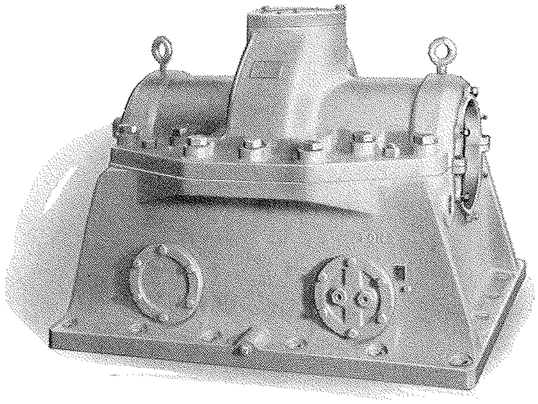


Figure 48
Special two-shoe bearing similar to the Kingsbury Bearings on the Liberty ships.

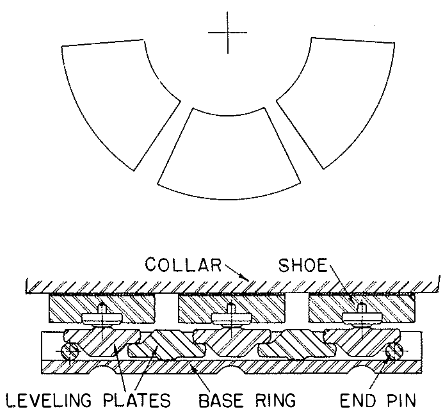


Figure 49
Diagram of three-shoe thrust bearing and arrangement of leveling plates used. (See page 12 for one of these bearings installed with an Elliott-Bowes Drive.)

may seem better, in view of limited shaft diameter or other requirements, to use a special three-shoe arrangement in a housing similar in most respects to the standard pedestal-type two-shoe housings. This is done by arranging the three shoes as shown in the diagram, Figure 49. The shoes are equalized by leveling plates, as in six-shoe bearings (page 24), but the end leveling plates bear against pins or blocks as the diagram indicates. The leveling plates are held in half base rings contained in the lower housing, leaving the housing cap free. This thrust bearing is used with the Elliott-Bowes Drive, shown in Figure 18, page 12. Many destroyer escorts and fleet tugs were equipped with three-shoe equalizing bearings during World War II.

For special purposes, special end closures can be furnished. See End Closures, page 30.

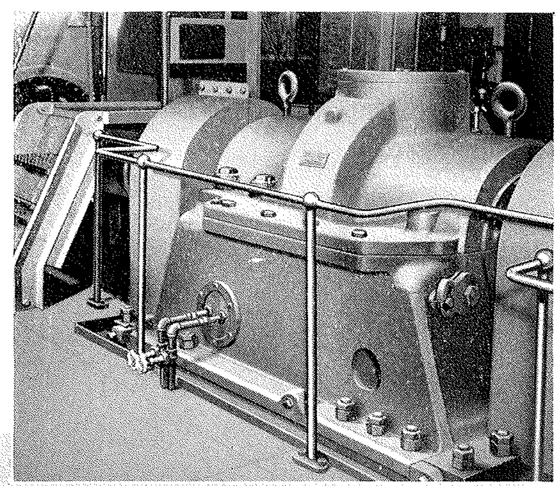
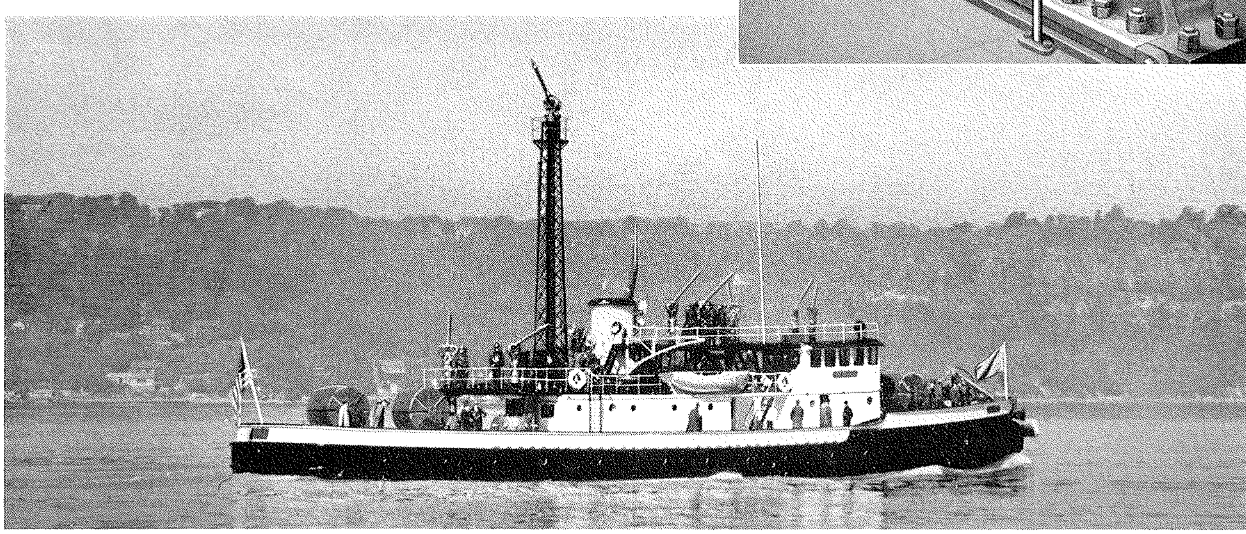


Figure 50
Special two-shoe bearing; one of two on propulsion shafts of U.S. Engineers' seagoing hopper dredge "Essayons."



New York City fireboat "Fire Fighter," equipped with Kingsbury Bearings on twin propulsion shafts.

Six-Shoe Equalizing Thrust Bearings

The Six-Shoe bearing makes maximum use of the thrust collar capacity in a given size. This fact explains its wide use in reduction gearing, where it can be placed at the forward end of the main gear shaft. Since the end of the shaft carries no torque, its diameter can be much reduced, and the thrust collar can be made correspondingly smaller. With six shoes, the capacity is still ample for the usual loads. The thrust housing is then an integral part of the gear case.

The same considerations favor the use of a six-shoe thrust at the forward end of a direct-connected motor drive.

Unlike the two-shoe bearings, the six-shoe type is self-aligning and equalizes the load automatically between the shoes. This is due to the shoes being backed by a series of interlocking *leveling plates*, as diagrammed in Figure 51. The effect is to absorb any slight difference in shoe thickness and error in shaft alignment, so that all shoes are equally loaded and their oil films are of equal thickness.

The shoes have hardened inserts in their backs, and generally resemble those in the two-shoe adjustable bearings. The leveling plates are steel forgings, with hardened bearing surfaces: the larger sizes have hardened inserts against which the shoes bear.

Each group of six shoes and twelve leveling plates is held in a *base ring*, which may be solid for assembling over the end of the shaft, or split for radial assembling. There are always two sets of these elements, for ahead and astern thrust.

For the forward end of reduction gearing (the commonest location) the standard arrangement is to use a separate collar, which is a slip fit on the shaft and is held against a shoulder by a massive nut.

The general form of shoes, leveling plates and split base ring is shown in Figures 52 and 53, which represent a small size of bearing. In Figure 55 is shown a marine-type double six-shoe thrust, one half fully assembled, the other half with shoes removed to show the leveling plates. The base rings in this bearing are solid.

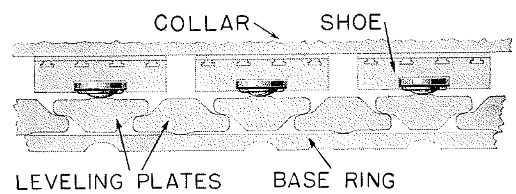


Figure 51
Developed diagram, showing principle of equalized support of shoes by use of leveling plates.

Figures 24-28 and 54 show typical assemblies of double six-shoe thrusts in reduction gear housings.

Correct lubrication is essential. In propeller shaft service these equalizing bearings run entirely submerged in oil, supplied under

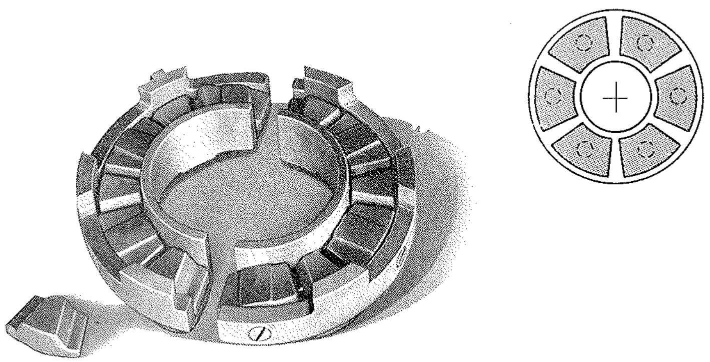


Figure 52
Split base ring and leveling plates of small six-shoe bearing. A "lower" leveling plate is shown separately.

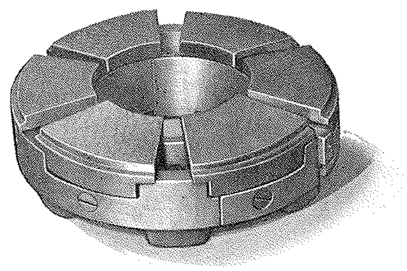


Figure 53
Small six-shoe bearing assembly.

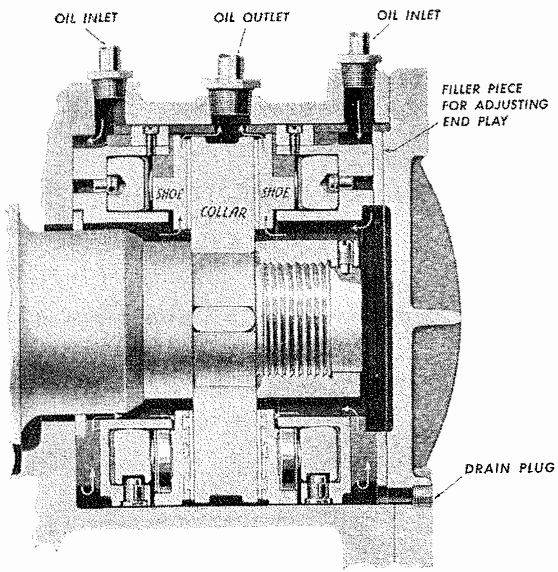


Figure 54
Vertical section of double six-shoe bearing in housing. Arrows show direction of oil flow.

merely nominal pressure. The oil enters passages in the backs of the base rings, flows along the shaft toward the collar on both sides, then turns outward between the shoes, and finally escapes at the top above the collar. See the arrows in the sectional view, Figure 54. In all cases the oil is taken from the general lubricating system and returns to it.

A sight glass and thermometer may be installed in the outlet.

A seal ring is interposed between the thrust cavity and the adjacent journal bearing, and the rate of oil flow is determined by restricting the outlet, so that the cavity is always full.

A drain plug permits checking the oil for dirt or water.

With the oil drawn down and the housing cap or inspection cover removed, the shoes may be lifted out by rotating the base rings, and the collar faces thereby exposed. Other parts may be withdrawn after removing the circular end cover plate.

The end play is fixed by a filler ring, usually located between the forward base ring and the end cover. Since there is no appreciable wear in normal service, the end play seldom needs to be changed, especially since considerable tolerance is permissible. Sometimes, due to rescraping the shoes and perhaps refinishing the collar faces, the allowed end play is exceeded. In that case a thicker filler ring should be substituted. Since the leveling plates automatically compensate for minor inequalities in shoe thickness, the shoes are interchangeable; also the leveling plates.

Six and Eight-Shoe Bearings at Aft End of Gear Housing

Sometimes it is more convenient to locate the thrust bearing aft of the bull gear instead of forward. To avoid too large a collar, this may lead to choosing eight small shoes.

In order to save axial space (which in that location is often limited), two collars are sometimes used, with a split journal bearing between them, and leveling plates with thrust shoes operating on the inner faces of the collars. See Figure 79.

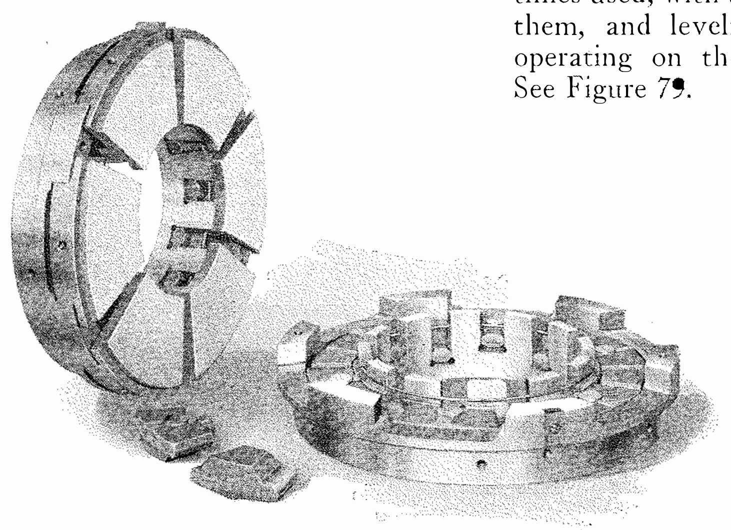


Figure 55
Large six-shoe bearing, without collar. Solid base ring and leveling plates are shown at right. Two leveling plates (upper and lower) shown separately.

Six-Shoe Thrust Bearings in Shaft Tunnels

In large twin or multi-screw ships it is sometimes necessary to locate the six-shoe thrust bearings considerably aft of the drive. For that purpose complete units are available, each with built-in journal bearing. Figure 57 shows such a thrust bearing, one of four built for SS "Manhattan" and "Washington" in the 1930's. On those ships two journal bearings were used with each thrust; but one is usually sufficient.

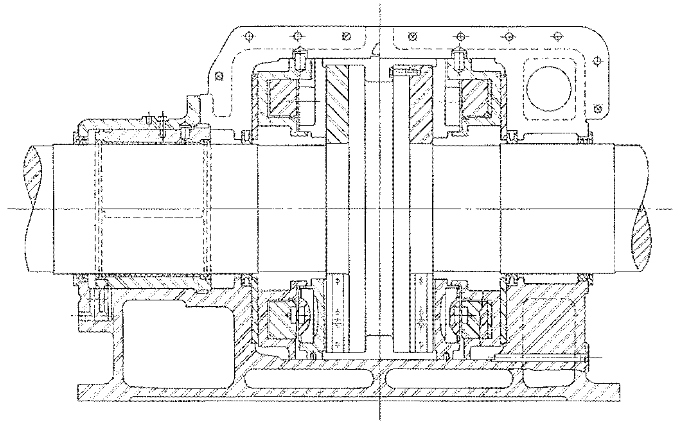


Figure 58
Large six-shoe propeller thrust bearing with single journal bearing. A recent design.

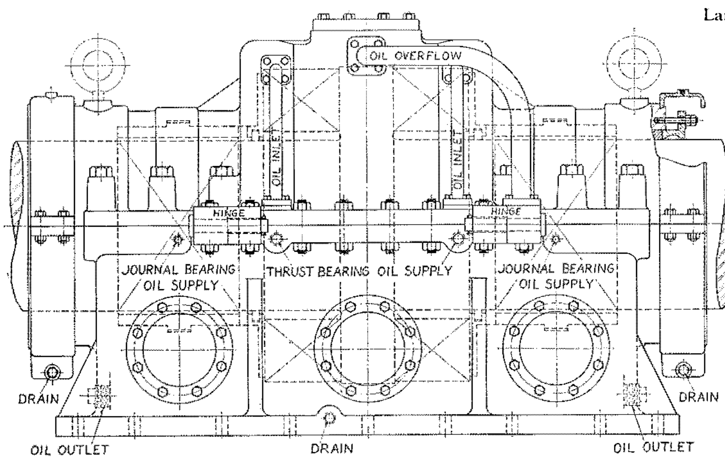


Figure 56
Kingsbury six-shoe propeller thrust bearing with two journal bearings. Collar is integral with shaft. Forced lubrication is used. See photo, Figure 57.

With this bearing arrangement, the thrust collar is always integral with the thrust shaft. To facilitate emergency repairs (as in Navy vessels), the shaft collar may have removable facing collars. See Figure 62 (e).

When external oil piping is not desired, automatic internal circulation can be furnished, with attached or built-in cooler. See page 46.

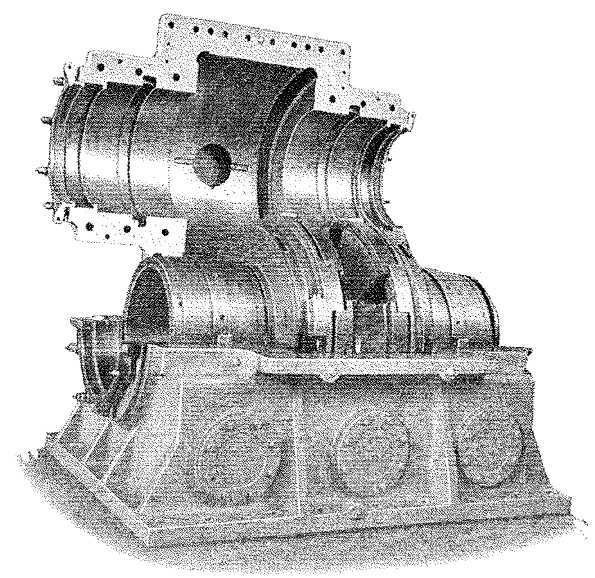


Figure 57
Large six-shoe propeller thrust bearing with two journal bearings. Collar and shaft not shown. See Figure 56.

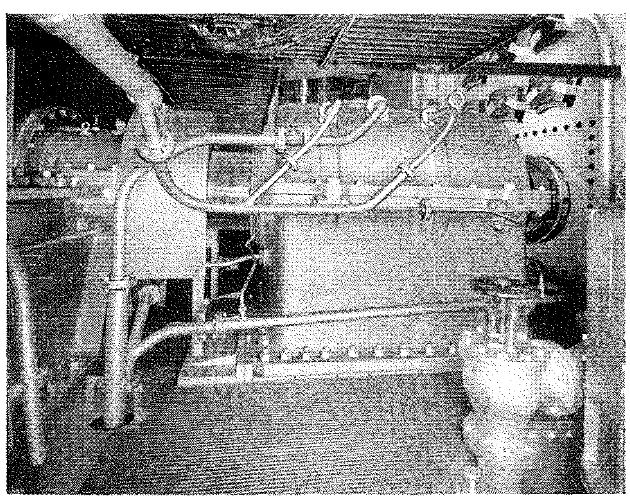


Figure 59
One of two six-shoe propeller thrust bearings installed on S.S. "America."

Dimensions and Rated Capacities

The nominal size of a Kingsbury Thrust Bearing is the outside diameter (in inches) of the shoe faces. The outer diameters of shoe bodies and the collar are slightly larger.

In Two-Shoe thrusts, the shaft diameter is roughly one-half of the nominal collar size. However, each style and size of bearing has several standard shaft diameters available. For a given collar size, the shoe area and the consequent thrust rating will depend somewhat on the shaft size, since the larger shafts reduce the area of the shoes.

Table I shows ratings of several Two-Shoe bearings at various speeds. The areas and capacities are mean values between the smallest and largest shaft sizes. For full particulars our separate catalog of dimensions and ratings for Two-Shoe bearings should be consulted.

Typical Ratings, Two-Shoe Adjustable Bearings — Table I

Bearing Size	Mean Area Sq. In.	Shaft Diameters	100 r. p. m.	150 r. p. m.	200 r. p. m.	300 r. p. m.	500 r. p. m.
9	16	3¾ to 4¼	......	3,200	3,400	3,600	3,900
12	27	4¾ to 5¾	5,300	5,700	6,000	6,500	7,000
17	59	6½ to 8	15,000	15,500	16,500	18,000	.....
23	105	9 to 10⅞	30,000	32,000	33,500	36,000	.....
29	165	11½ to 13⅝	51,000	54,000	56,500	.....	.....
37	267	15⅛ to 17⅝	90,000	96,000	100,000	.....	.....
45	394	18½ to 21¼	140,000	150,000	156,000	.....	.....

In Six-Shoe thrusts of *pedestal type*, with collar usually integral, there is an option of shaft sizes similar to that with Two-Shoe bearings; and the shoe area and consequent thrust rating are similarly affected by the shaft size.

In Six-Shoe thrusts for the *forward end* of the main gear or motor shaft, there is one standard collar bore for each bearing size, and one standard shoe bore. The shoe bore is one-half of the nominal bearing size. The shaft diameter at that point is therefore a little less, in order to clear the shoes (actually about equal to the smallest shaft size for the pedestal type). Aft of the shoes and base ring the shaft may be much larger, as in Figure 54, to suit the requirements of the gear or motor builder.

Table II shows specimen ratings of Six-Shoe bearings. It applies to bearings on the forward end, and is nearly correct also for the smaller shaft sizes of the pedestal type. For full particulars our separate catalog of dimensions and ratings for Six-Shoe bearings should be consulted.

Typical Ratings, Six-Shoe Equalizing Bearings — Table II

Bearing Size	Area,* Sq. In.	Shaft Diameters	60 r. p. m.	100 r. p. m.	150 r. p. m.	200 r. p. m.	300 r. p. m.	400 r. p. m.
13½	91.1	5⅝ to 7	......	23,500	25,000	26,500	28,000	29,500
17	144.5	7½ to 9	......	41,500	44,000	46,500	49,500	52,000
21	220	9 to 10⅞	......	69,500	74,000	79,000	84,000	.....
27	364	12⅛ to 14⅝	118,000	128,000	137,000	143,000	155,000	.....
31	480	13⅝ to 15⅞	165,000	179,000	192,000	200,000	......	.....
37	684	16¾ to 19⅜	253,000	275,000	295,000	310,000	......	.....
45	1012	20¼ to 23¼	405,000	440,000	470,000	......	......	.....

*Areas and ratings apply to standard bearings on the forward end, also to the pedestal type when the *smallest* shaft is used. For the largest shaft, the areas and therefore the ratings are materially less.

Final Choice Of Bearing Type And Size

The question often arises, "How can we tell whether to use a two-shoe bearing or a six-shoe?" The answer is to be found in the following paragraphs. The choice depends mainly on the bearing location, as discussed on pages 8 to 17. In general, the two-shoe type is used when the bearing is aft of the drive. However, when the thrust load would require a two-shoe collar unreasonably large as compared with the shaft diameter, the smaller six-shoe type should be chosen, with appropriate housing.

Every bearing must, of course, be large enough to carry the thrust load and also to accommodate the shaft passing through it. Thus thrust load and shaft diameter are basic factors. Another factor is the speed, which affects the rated capacity and is important as regards cooling.

The thrust load, if not determined by model tests or trials of comparable vessels, may be estimated approximately from the thrust factors in Table III, page 42, using shaft horse-power and vessel speed.

The required thrust shaft diameter, if not determined by accurate calculation, may be approximated from charts, page 42, based on power and revolutions. However, shaft sizes may have to be increased to avoid trouble from torsional vibration. Hence, the diameter must be definitely determined before the final bearing selection can be made.

Location Abaft the Drive

When the thrust bearing is aft and is separate from the propelling machinery, the normal choice is a pedestal type housing, which will include a journal element. Ordinarily the two-shoe type will be used, unless the thrust load would require a two-shoe collar unreasonably large as compared with the shaft diameter. In that case, the smaller six-shoe type is to be chosen.

The selection of bearing types and sizes will be more easily understood by study of the following examples, each of which makes use of the ratings on page 27, and shows for com-

Examples

(A) ***Given Conditions:*** 96,000 lbs. thrust load, 150 RPM, and 13¾ in. shaft dia. *required.*

Two-Shoe Selection: 37 in. size, 15⅛ in. shaft, 104,000 lbs. thrust rating.

Six-Shoe Selection: 27 in. size, 14⅜ in. shaft, 105,000 lbs. thrust rating.

Comments: The two-shoe bearing is heavy and bulky; the minimum shaft available is larger than required. The six-shoe is preferable for this case.

(B) ***Given Conditions:*** 48,000 lbs. thrust load, 200 RPM and 13 in. shaft dia. *required.*

Two-Shoe Selection: 29 in. size, 13⅝ in. shaft, 52000 lbs. thrust rating.

Six-Shoe Selection: 27 in. size, 13⅝ in. shaft, 120,000 lbs. thrust rating.

Comments: The two-shoe is not much larger than the six-shoe, and would be less expensive. It is therefore the better choice.

(C) ***Given Conditions:*** 27,000 lbs. thrust load, 300 RPM, and 10¾ in. shaft dia. *required.*

Two-Shoe Selection: 23 in. size, 10⅞ in. shaft, 32,000 lbs. thrust rating.

Six-Shoe Selection: 21 in. size, 10⅞ in. shaft, 65,000 lbs. thrust rating.

Comments: The two-shoe bearing is decidedly the better choice. The 10¾ in. shaft requirement governs the bearing size in this case. Smaller bearings of either type would carry the thrust load.

parison the two-shoe and the six-shoe (pedestal type) choice. However, for any actual application, the marine engineer should always consult the full tables of dimensions and ratings published in our specialized catalogs of two-shoe and six-shoe bearings.

If the two-shoe *type* is found satisfactory, the particular *style* to use depends mainly on the journal bearing construction required, as explained on pages 20-21. We can always make a suitable selection if we have the particulars listed under "Data Needed With Inquiries," on page 47.

Sometimes in large multi-screw turbine driven vessels, the thrust bearings must be located well aft instead of in the engine room, in order to avoid longitudinal shaft vibration at critical speeds. The choice here is six shoes, with each bearing in a self-contained housing mounted in the shaft tunnel: in most cases each has its own lubricating system.

If conditions require a six-shoe pedestal bearing, despite its somewhat higher cost, the matters of lubrication and cooling require attention. We should be consulted in all such cases. Lubrication and cooling will usually be by forced feed from a general lubricating system, if that is available; otherwise each thrust bearing may have its own independent system. Complete self-contained bearing units, including coolers and internal circulating devices, are available when required.

Bearing Attached to Aft End of Engine or Gearing

Here there are several designs to consider, some of them modifications of standard styles, others special. The choice between two shoes and six is made as explained above.

The standard housings can be altered if necessary to provide struts between the lower half and the base of an engine (Figure 46). Preferably these struts are separate members, bolted in place. Another modification sometimes useful is to raise the supporting flange to a point near the shaft centerline, to permit bolting down on the engine bedplate (Figures 15 and 45) or on a special foundation.

The compact GC type, with its circular flange, is available for bolting to a vertical facing on the engine or flywheel casing, in locations corresponding to Figure 21 and Figure 43.

A bearing at the after end of the gear housing usually has a thrust collar integral with the shaft. Sometimes split facing collars are provided, bolted to a flange formed on the shaft. See Figure 62(e). In special cases the collar may be made removable by being combined with the coupling.

Bearing at Forward End of Reduction Gears

These are normally of six-shoe type. The housings are a part of the gear case, leaving only the internal thrust elements to be furnished by us. The thrust elements are forward of the journal bearing. The shaft can be reduced in size, and the collar need be only large enough to carry the thrust on six shoes.

These six-shoe bearings are used in the great majority of geared turbine drives, excepting only those cases where the thrust bearing must be placed elsewhere to minimize critical longitudinal vibration, or because of interference by the condenser.

Direct Electric Motor Drive

Electric propulsion motors are sometimes equipped with thrust bearings on the forward end. If the frame design permits, a standard two-shoe bearing may be used, with its own journal bearing arranged to support the rotor, and with the usual automatic lubrication. If needed, removable top and bottom journal bearing shells can be furnished. Since the center of thrust of the two-shoe type is below the shaft center, the integral collar usual with two-shoe bearings is desirable, though a collar forged on a sleeve might be used.

Special Requirements

Special and unusual requirements may make it advisable to depart from standard types of bearings. Several examples of this are shown on pages 43 to 46. We should always be consulted, both as to the possibility of adapting a standard design to special needs, and as to the best choice of design if something different must be used.

End Closures

In most cases, the end closure has no oil pressure to resist. It is merely a wiper. Sometimes it must also act to exclude dirt or water.

At the usual shaft speeds, in the bearings here considered, the stuffing box shown in Figure 60(i) is quite satisfactory. It is standard for styles GH and GF. The gland is split, with halves bolted together. It should never be set up tightly: that would score the shaft.

When endwise compactness is desired, a single felt ring with a plain follower, Figure 60 (n), is substantially equivalent to a stuffing box. It is used in Style GK.

The Crown Ring, Figures 60 (m) and 61, has several advantages. It consists of a split bronze ring, serrated on its inner face, and held in place by a bronze "adapter" of internal "comb" form. The serrations act to push back oil creeping along the shaft. The "comb" acts as a barrier against entrance of water and dirt. This closure is usually employed with Style GC and may be used with GF to meet special needs.

For various special conditions other closures may be used, especially to resist passage of oil, water or dirt at various speeds.

Since in no case does the journal bearing shell extend as far as the end of the bearing housing, there is little tendency for oil to creep along the shaft. The higher the speed, the more rapidly oil is thrown off at the end of the shell. The lower the speed, the more readily it is stopped by the closure.

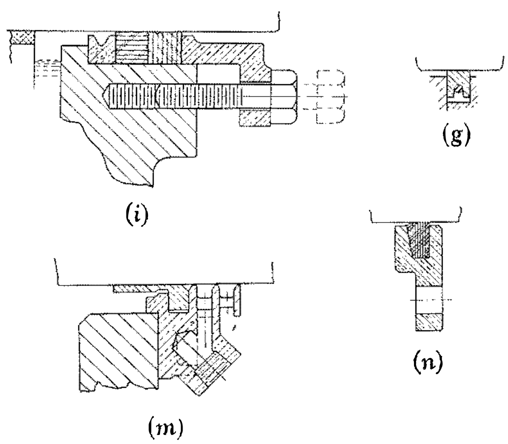


Figure 60
End closures commonly used. (i) Stuffing box used with Style GH bearing and usually with GF. (g) Simple oil seal ring. (m) Crown Ring all-metal closure. Standard with GC bearing; may be used with GF. (n) Felt ring compressed into bronze ring. Standard with GK bearing; sometimes used with GF.

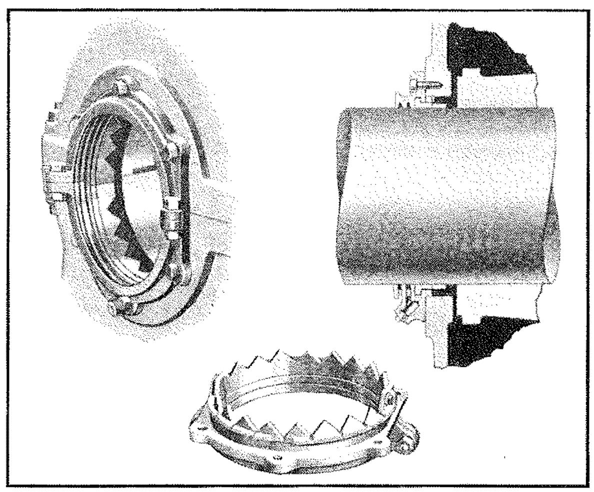
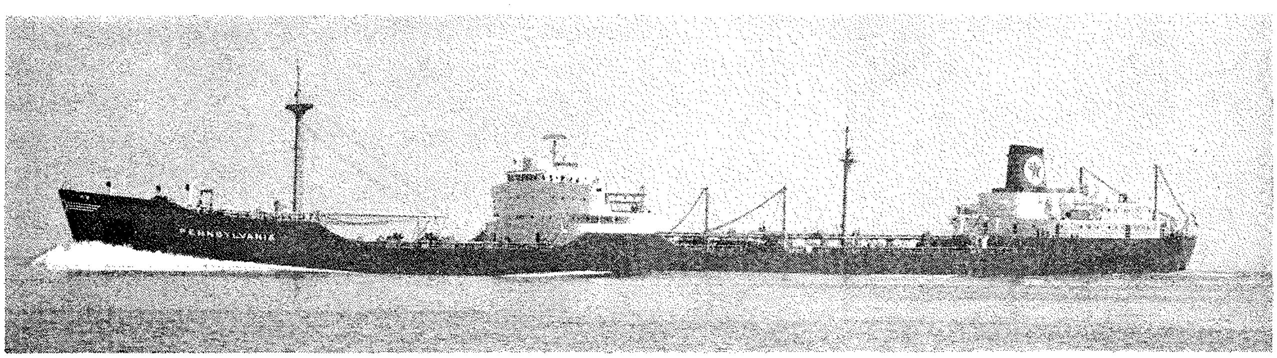


Figure 61
Crown Ring Bearing: consisting of split bronze crown ring and split adapter ring.



28,000-ton tanker "Pennsylvania." One of several equipped with the reduction gearing and Kingsbury thrust bearing shown in Figures 25 and 63.
Courtesy Bethlehem Steel Co., Shipbuilding Div.

Thrust Collar Details

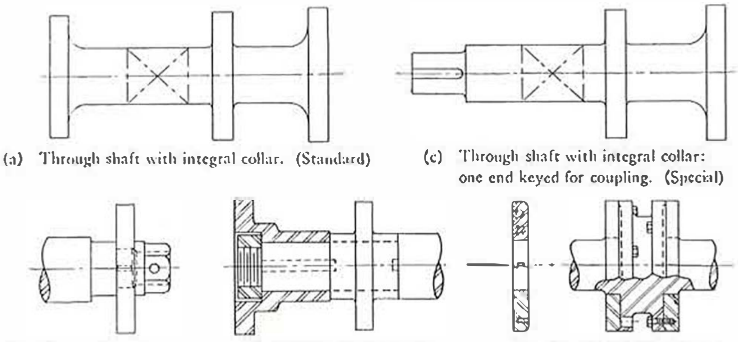
Standard Collars. The commonest form of thrust shaft is short, with the thrust collar and coupling flanges forged on it. See Figure 62 (a). Near the collar, forward or aft, the shaft is usually finished for a journal bearing. Sometimes provision is made for a removable coupling on one end; Figure 62 (c). Small shafts can be supplied, completely finished, by us; larger ones by the customer, finished to our instructions.

When located at the end of a shaft, the thrust collar is made removable, as shown in Figure 62 (b), and is furnished by us. This construction is used with six-shoe bearings. For two-shoe bearings a collar with sleeve-type hub, as in Figure 62 (d), is preferable because of the eccentric loading: the flange shown may be included or omitted.

Special Collars. When the shaft shoulder is radially narrow, a sleeve-type hub should be used: see Figure 62 (d). The hub length should be 1 to 1½ times the bore, and the shaft shoulder should be wide enough for durability.

Special applications may require renewable facing collars when the thrust bearing is placed aft. A typical design is shown in Figure 62 (e). These facing collars are split, and the halves are tongued and grooved together, and bolted securely to the shaft collar.

Collar Dimensions. The collars shown in Figures 62 (a) and 62 (b) follow the standards given in our dimension catalogs, furnished on request. Those in Figures 62 (c), (d) and (e) are variable according to conditions. We should always be consulted.



(a) Through shaft with integral collar. (Standard)
(c) Through shaft with integral collar: one end keyed for coupling. (Special)
(b) Removable collar on end of shaft. (Standard)
(d) Collar and end flange on sleeves. (Special)
(e) Removable facing collar for quick repair. (Special)

Figure 62
Thrust Collar Details.

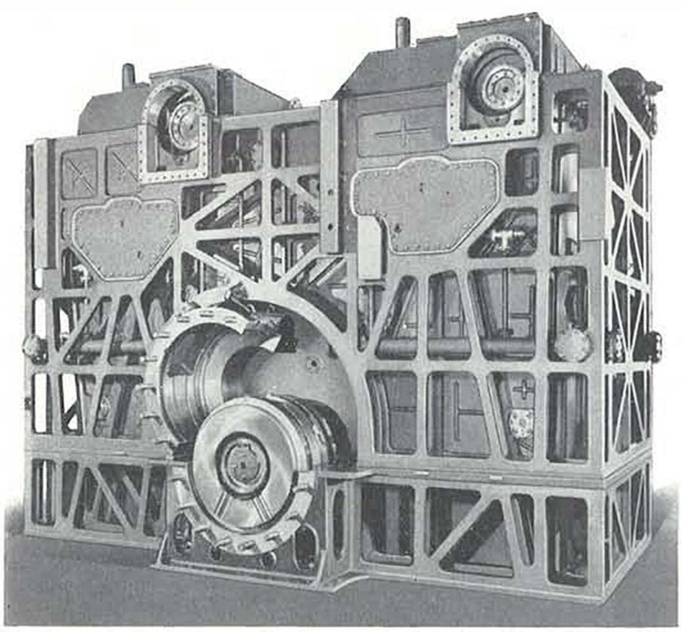


Figure 63
Reduction gear housing, showing six-shoe Kingsbury Bearing. Used on oil tanker "Pennsylvania" (page 30), and sister ships. See also Figure 25, page 14.
Courtesy The Falk Corp.

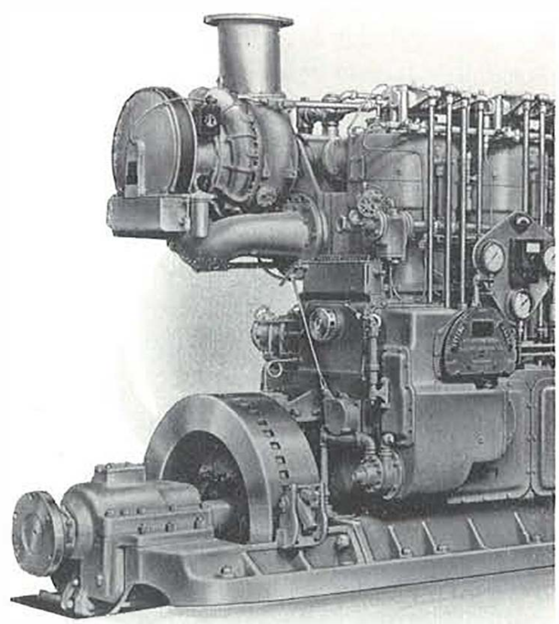


Figure 64
Six-shoe Kingsbury Bearing, in special housing, on Diesel engine bedplate.
Courtesy Atlas Diesel Div., National Supply Co.

Circulation and Cooling

A Kingsbury Thrust Bearing carries a highly concentrated load—usually from 250 to 500 lbs. per square inch of shoe area. Hence some method of circulating the oil, and removing the heat due to oil film shear, is an essential part of every Kingsbury Bearing.

The simplest arrangement is that employed in Two-Shoe bearings with automatic lubrication and air cooling. The thrust collar dips into a bath of oil in the lower housing: oil adhering to the collar is scraped off at the top and streams down to the collar faces and the journal bearing, and returns to the bath. The heat in the oil is carried away by the surrounding air and through the foundation. A reasonable amount of ventilation is required for air cooling. Wood foundations are inferior to steel for dissipating heat.

The power loss depends both on the load and on the speed, but mainly on the speed. If the speed of the Two-Shoe bearing demands additional cooling, a water coil is used in the bath. Whether air-cooled or water-cooled, the Two-Shoe bearing normally operates at a temperature of 120° to 150° F., and therefore uses a rather heavy oil. Specific oil recommendations are made for each installation, and are included in the instruction booklet.

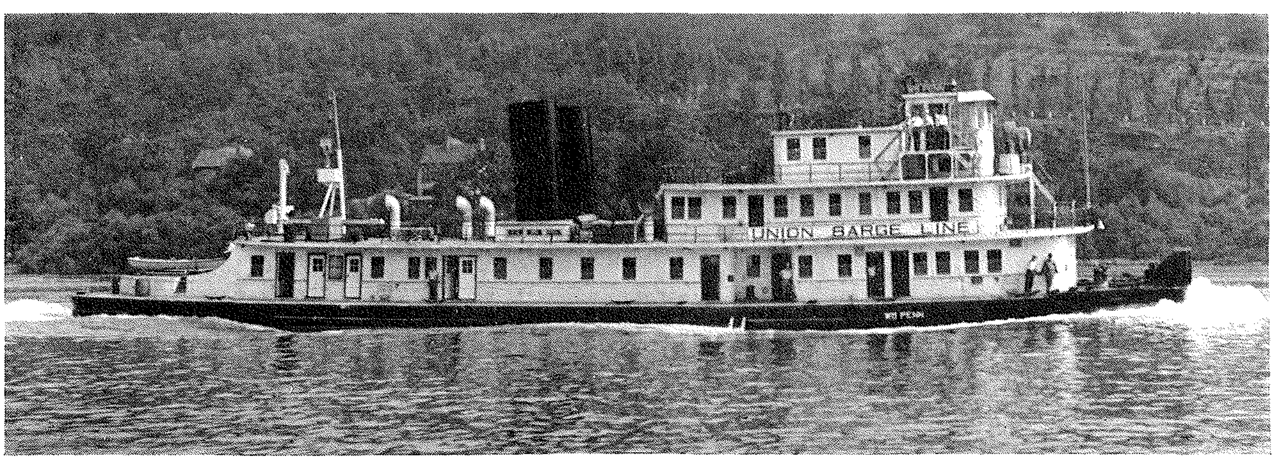
If preferred, a Two-Shoe bearing may be supplied with circulating oil from an outside reservoir and cooler, with return pipe.

Six-Shoe bearings, built into the reduction gear housing, run in a restricted cavity. In the circulation cycle, the oil is filtered and cooled, then enters the bearing through suitable passages in the base rings, and flows out at the top, above the collar. The rate of flow is such as to absorb the heat due to oil film shear, with a temperature rise of usually not over 15 to 20 degrees F. between inlet and outlet.

Sometimes it is necessary to mount a Six-Shoe bearing well aft of the drive. In that case, it is not feasible to use the main lubricating system. Instead, an independent system may be employed, including an oil pump either chain or motor driven. Alternatively, a self-contained system is available, involving a development of the two-shoe oil scraper, adapted to supply oil to six shoes from a bath in the lower housing. That makes it unnecessary to fill the entire housing with oil.

The Line Shaft Bearing has its own reservoir and means of circulating the oil, as noted in the description of that bearing, page 37.

The cooling requirements for any given combination of load and speed, with either Two-Shoe or Six-Shoe bearings, can be closely estimated, and the required flow of oil or water specified accordingly.



Towboat "William Penn," equipped with two Kingsbury Bearings on propulsion shafts.

Foundation and Alignment

Although the thrust bearing is a small unit, the force it receives must be transmitted to the entire hull. To accomplish this without harmful concentration of stresses, the bearing foundation should extend far enough to distribute the thrust.

The thrust shaft must also maintain alignment with the line shaft and drive shaft despite weaving, hogging and sagging of the hull, and the overturning leverage of the thrust.

No problem is presented when the thrust bearing is built into a turbine reduction gearset,

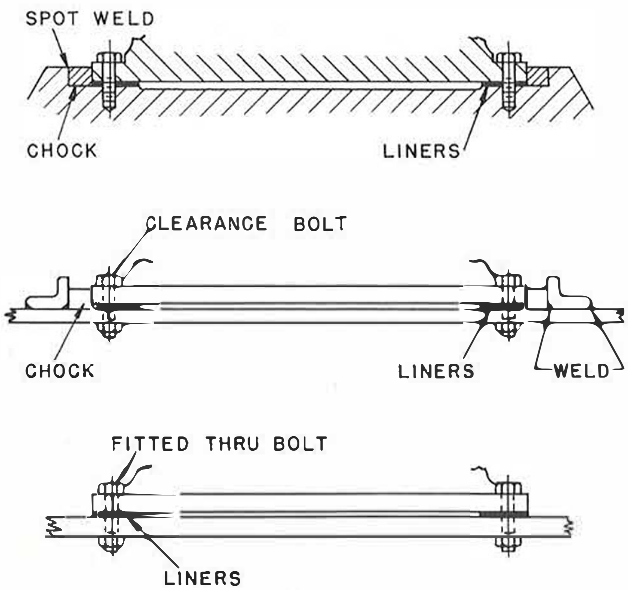


Figure 65
Various methods of securing alignment of housings.

or when it is just aft the drive on an extension of the engine foundation. In either case the foundation can readily be made adequate not only to support the drive but also to transmit the thrust to the hull.

If, however, the thrust bearing is to be located farther aft, special study is needed, and cooperation between naval architect, machinery builder and owner. If possible, the thrust bearing foundation should be extended forward and tied directly to the foundation of the engine or the reduction gears. In a wooden hull, approved practice favors a composite foundation of steel plates or angles on a substantial wood base.

The housing should always rest on metal. To secure it in exact alignment, the best method is to fit liners underneath, in connection with fitted chocks fore and aft. If the foundation is built up of structural shapes, a riveted or welded angle is used with each of the chocks.

An alternative method is to use fitted through-bolts. These are suitable for the smaller mountings or lighter thrust loads.

When liners or shims are used, they must not be thick unless the bolts or dowels are designed to resist bending. A group of thin shims is worse in this respect than one thick one. Solid liners having a thickness of one bolt diameter or more may be used, provided the bolt fits them tightly and also fits tightly in reamed holes above and below the liner.

If a short special base is used, the foundation must be extra stiff to withstand the overturning leverage of the thrust. If the thrust housing be tied directly to the engine frame, the base flange need not take any thrust.

The use of self-aligning bearing units never excuses careless alignment; its purpose is mainly to compensate for unavoidable flexure and weaving under stress.

Since the hull (whether steel or wood) is unavoidably flexible, and the foundation must "go with" it, it follows that needless stiffness of the shaft will increase the liability to bearing troubles. To limit its diameter, the shaft is sometimes made of alloy steel. It follows that the exact combination of thrust bearing location, shaft design and foundation design, must be worked out for each particular ship.

Besides the tendency of the foundation to deflect under the overturning action of the propeller thrust, there is the problem of hull weaving in a seaway. If the thrust bearing must be located midway between the drive and the tail shaft, it is especially necessary to provide a rigid foundation to keep the thrust bearing in line.

It is usual, and preferable, to make the final line-up after the hull is water-borne.

Installation and Operation

***Note:** A special Instruction Booklet is furnished with each bearing.*

Two-Shoe Propeller Thrusts

Although installation is a matter for the shipyard or the engine builder, a few notes are given here. See also "Foundation and Alignment," page 33.

The bottom of the housing is machined flat, and is intended to be supported on the foundation direct or by blocks of carefully selected thickness to bring the thrust shaft into line with the connecting shafting. The housing should be so placed fore and aft as to center the thrust collar in the space provided for axial adjustment when the shaft couplings have been made up. The end closures must not bind.

The thrust forces may be transmitted to the foundation by steel angles and fitted chocks or wedges, or directly through bolting. However, it is desirable that at least two bolts on each side be fitted by reaming, in order to determine the exact location of the housing before the chocks or wedges are finally fitted into place. If chocks or wedges cannot be used, all the holding-down bolts should be fitted.

Before assembling, make sure that all bearing parts, as well as the interior of the housing and oil piping, are clean. The slushing compound should be washed off, preferably with kerosene, and all traces of dirt removed. This is particularly necessary for the bearing surfaces of shoes and collar. *A bearing surface is not clean until a white cloth wiped over it shows no soil.*

To adjust the jack screws which support the thrust shoes, the following procedure should be followed:

A total end play of about .001″ per inch of collar diameter is satisfactory. After the housing has been bolted down, run all the jack screws in until the shoes bear evenly on the collar. Use a hand wrench, and take light fractional turns on alternate screws till the oil films stop yielding. Lock the jack screws on the ahead (loaded) side. Then back the jack screws on the go-astern side by exactly equal amounts, using a "feeler" with thickness equal to end play, back of the supporting pivot of each shoe on the unloaded side. See Figure 67. Lock the jack screws and remove the "feelers." Now re-oil the collar.

Before placing the housing upper half, remove the top cover plate and the bronze oil scraper. Replace these after the upper housing is bolted down.

After assembling, and before starting to run, pour oil into the housing up to the "HIGH" mark on the oil level gauge.

Operation is only a matter of maintaining the proper oil level, and checking occasionally at the drain plug to note any contamination of the oil by water or sludge. If a cooling coil is used, it should be checked for possible leakage.

Six-Shoe Propeller Thrusts (on end of shaft)

The following refers to a bearing in the forward end of a reduction gear case.

One-piece or "solid" base rings must be assembled in sequence with the thrust collar over the end of the shaft. Such items as seal rings and filler pieces, if solid, must be assembled likewise in their proper order.

The usual procedure is to start with the main gear shaft on its journal bearings, and with the end and top covers of the thrust bearing housing removed.

All parts must have been carefully cleaned as described above for two-shoe bearings.

The leveling plates are first reassembled in the base rings, where they are loosely held, either by set screws or by wire retaining rings, depending on the design.

Before assembling the outer base ring in place, the shaft nut should be driven very tight on the collar, and then locked. Oil the thrust collar faces.

Do not insert the shoes till thrust collar and base rings are in place. Then insert them one at a time by rotating the base rings. See that the base rings do not hit or touch the collar faces. After the shoes are inserted, turn the base rings to bring keys to top center.

A filler piece is used to establish the proper end play: sometimes two are used in order to fix the fore-and-aft position of the thrust collar, and hence the position of the main gear. Usually the filler piece, or pieces, will have been ground to the proper thickness by the gear manufacturer, and should require no attention from the ship's engineer. However, the procedure for checking the end play and for correcting it, if that is made necessary by abnormal wear, should be understood by the Chief Engineer. Such checking must always be done with the housing cap bolted down. The exact procedure will depend on the housing construction and the feasibility of jacking the main shaft. See "Checking End Play," page 36.

End play allowances for the individual bearing are given in the instruction booklet.

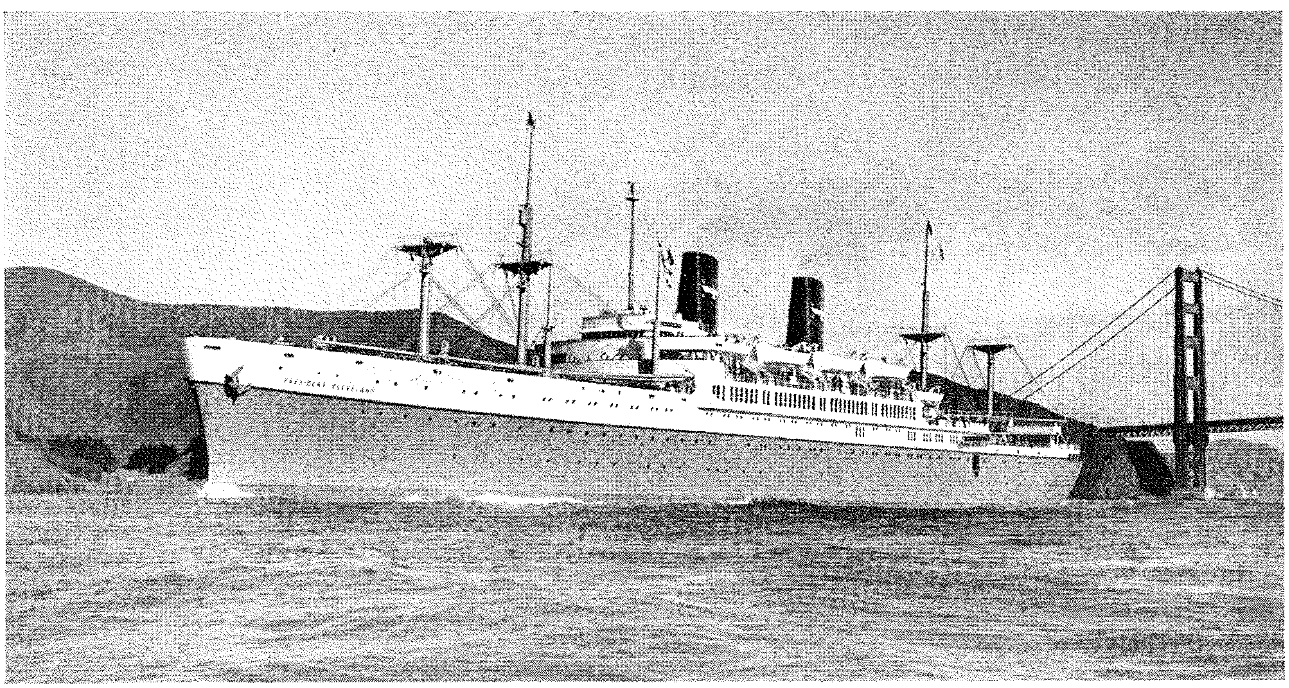
After the end filler piece and end cover are in place and the end play properly checked, pour oil into the thrust cavity for starting lubrication.

Operation is only a matter of supplying the required flow of clean cool oil, and checking the bearing temperature from time to time.

Six-Shoe Propeller Thrusts (aft of drive)

Six-Shoe pedestal-type thrust bearings in the shafting have integral thrust collars, and are arranged for radial assembling of all other internal parts. Cleaning, and lining up the lower housing, are done as with two-shoe bearings. The filler plate (or plates) are ground to correct thickness in the Kingsbury shops.

With the housing in its approximate position, and the journal bearing lower shell in place, the shaft is lowered to the bearing, lined up, and the flanges bolted. After final adjustment of the housing position, holes are drilled and reamed for the holding-down bolts. The lower halves of the base rings, with the leveling plates assembled in them, are rotated into place; the shoes are inserted, and the upper halves and bearing shell assembled. The housing cover is then bolted down, and the housing is filled with oil.



S.S. "President Cleveland." This and her sister ship, S.S. "President Wilson," are each equipped with two Kingsbury Thrust Bearings on the propulsion shafts. Bethlehem Steel Co., Shipbuilding Div., builders.

Checking End Play

Propeller shaft end play should be checked periodically. The method will depend partly on the accessibility of some convenient part of the shaft for checking, and also (in case the shaft movement cannot readily be checked) on the type of thrust bearing.

Below are indicated briefly the principal methods of checking. For fuller information, see instruction book furnished with the bearing.

Checking End Play While Running

The simplest method (when it can be used) of checking end play is to use a micrometer measuring instrument on some accessible part of the shaft. This should be done, if possible, when the propeller is turning slowly, as when the ship is maneuvering to approach her pier, before the machinery and shaft are cold.

Sometimes the measuring device can be mounted at the forward end of the propeller shaft. This may be feasible with a two-shoe bearing at the forward end of an electric propulsion motor, also with a six-shoe bearing at the forward end of reduction gearing.

If there is no provision for such measurements at the forward end of the propeller shaft, it is often possible to mount a dial indicator to bear on one of the shaft flanges. (If the thrust bearing is mounted far aft, with a flexible coupling between it and the line shaft, the only place for a measuring device is at a flange of the thrust shaft.)

If the machinery is cold, it will be necessary to jack the shaft fore and aft. This should be done cautiously, while rocking the shaft slightly to right and left by the turning gear to relieve the static friction.

Six-Shoe Bearing: Measuring at Thrust End Cover

With the power off, the end play of a six-shoe bearing built into the forward end of reduction gearing can be checked by inserting a thick dummy filler, accurately machined, replacing the regular filler in the housing bore. The *top* cover must be firmly bolted down. First tighten the *end* cover flange bolts evenly with the dummy filler in place (avoid springing cover), then measure the air gap between the cover flange and housing. Dummy filler thickness less regular filler thickness, less air gap, equals end play. Details are described in instruction books. If necessary, grind or shim the regular filler, or replace with one of proper thickness.

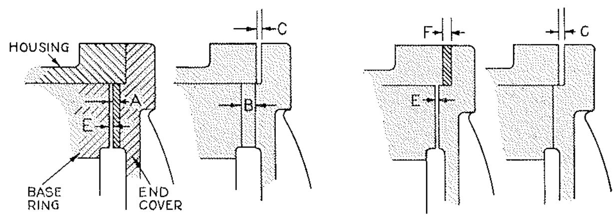


Figure 66
Measuring end play in six-shoe bearing. A, regular filler plate. B, dummy filler. C, air gap. E, end play. F, regular filler ring. Left: B − A − C = E. Right: F − C = E.

If the filler is a ring under the end cover flange, instead of in the housing bore, the filler is simply removed and the flange bolts tightened. Filler thickness, less air gap, equals end play.

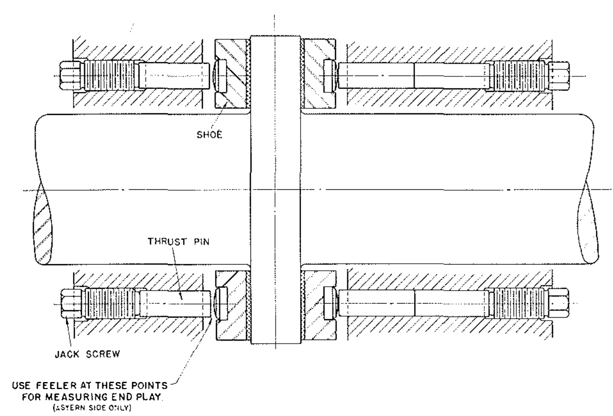


Figure 67
Measuring end play in two-shoe bearing.

Two-Shoe Bearing: Measuring Inside Housing

The procedure is the same as described under "Installation of Two-Shoe Thrusts," page 34. The engine crankshaft position should first be checked, to make sure the endwise bearing clearances are correct, and the thrust bearing jack screws should be adjusted forward or aft if necessary.

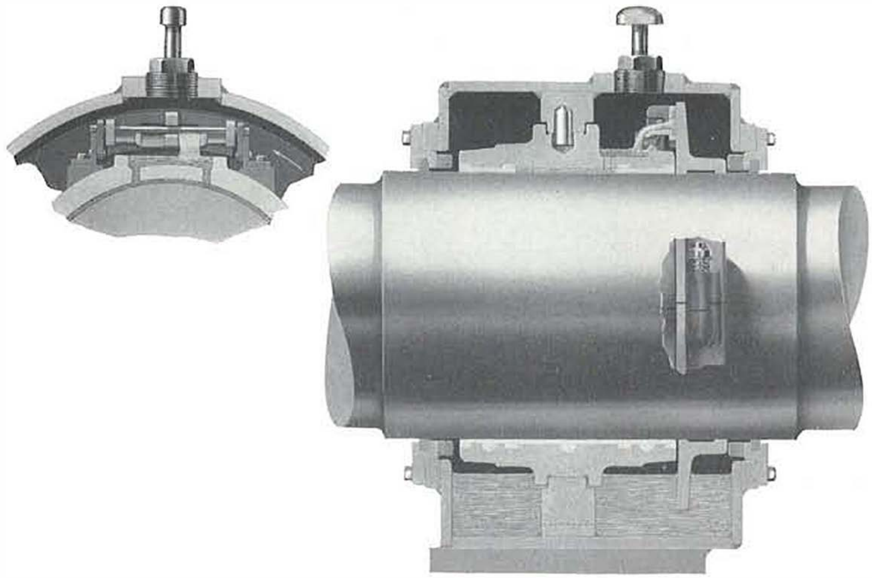


Figure 68
Kingsbury Line Shaft Bearing.

Line Shaft Bearings

One important requirement of line shaft bearings is that they should be able to run for long periods without attention.

Another is that they should be as near wearless as possible, even under the difficult conditions imposed by hull weaving and flexure of the shaft.

Kingsbury Line Shaft Bearings are designed to meet these requirements. They run under flooded lubrication, which requires no moving parts other than a "pumping disc" secured to the shaft. This device produces a stream of oil which, entering a large pocket in the upper half of the journal bearing shell, cools and lubricates the bearing.

Hull weaving and shaft flexure are taken care of by making the bearing shell self-aligning. Since the load normally carried consists only of the weight of the shaft itself, a long shell is unnecessary. In our experience, a long shell is distinctly undesirable even if self-aligning, and much more so if not, since shaft flexure tends to wear a long shell bell-mouthed, thus reducing the effective bearing area and hastening the destruction of the bearing. With the short self-aligning shell used in the Kingsbury Line Shaft Bearing there is no undue concentration of load at any point, and the oil film is practically uniform from end to end. Although the shell is self-aligning, reasonable care must be taken to line up the housing with the shaft at original installation. If that is done, the continuous oil film will give indefinitely long bearing life and remarkably high load capacity.

As shown in Figure 68, the pumping disc is spring-clamped on the shaft. It carries oil from the bath in the base to a bronze wiper at the top. The wiper has a shallow groove, meeting the face of the disc and ending in a dam, and oil adhering to the disc builds up a slight pressure in the groove. Under that pressure the oil is expelled into a spout leading to the pocket in the top of the shell. The heat due to film shear is carried by the oil stream to the shaft and housing walls, and thus dispersed. With lubrication and cooling thus completely automatic, the bearing can run for long periods with no more attention than occasionally to check the oil gauge and to sample the oil at the drain plug for possible water or sludge.

The usual end closures are felt rings, similar to those of the Style GK two-shoe bearing, Figure 60 (n), page 30.

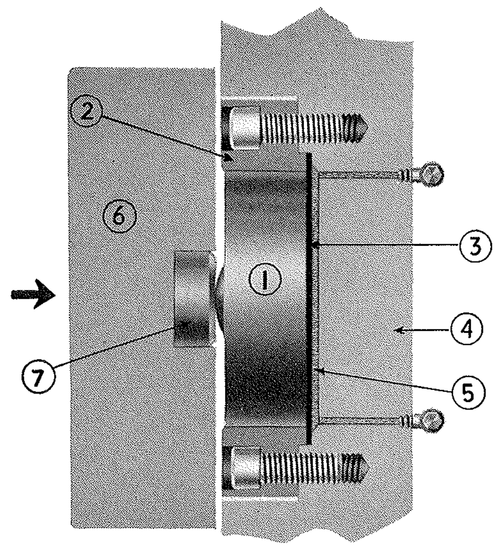


Figure 69
Kingsbury Pressure Cell (principal elements).

Kingsbury Thrust Meters

Kingsbury Thrust Meters are used to make direct measurements of ship propeller thrust, thereby making possible full-scale performance tests of various propeller designs. The meters differ in detail, but all are similar in principle. In each case the meter is attached to the thrust bearing housing, temporarily or permanently.

In effect, the Kingsbury Thrust Meter is a hydraulic weighing system, interposed somewhere between the thrust shoes and the bearing housing.

The essential feature of the Thrust Meter is the Kingsbury Pressure Cell: Figure 69 illustrates the important parts. The Pressure Cell consists of a *piston* (1) sliding in a *cell ring* (2), and bearing against a flexible *diaphragm* (3) clamped at its rim between the cell ring (2) and the *cell plate* (4). Behind the diaphragm is a shallow space (5), of about the same diameter as the piston, filled with liquid. The thrust load from each shoe (6) acts through the hardened insert (7) against the piston, and is balanced by liquid pressure, measured by an accurate Bourdon tube gauge. Thus the load is transferred from a mechanical to a leak-proof hydraulic support.

The diaphragm is made of Neoprene, and the liquid is lubricating oil (rubber and castor oil were formerly used). There may be one or several cells, depending partly on the amount of thrust to be measured. If there is more than one, all the cells are connected to one gauge, and all are supplied with oil by one hand pump.

To measure the thrust, oil is pumped in behind the diaphragms until the load is completely carried by the oil, instead of by the regular supports. This involves a small piston movement, which is limited by stops in both directions to prevent damage to the diaphragms. An arrangement of "feeler rings" shows definitely when the thrust is borne by the oil instead of by the stops. A dial indicator usually supplements the feelers. The thrust load is then equal to the combined area of the pistons multiplied by the observed oil pressure. During the test, little or no further pumping is required to keep the pistons clear of the stops, even under widely fluctuating loads. Due to the simplicity of the Pressure Cell, the measurements are exceptionally accurate.

When not testing, the oil pressure is released, and the thrust is transmitted through the stops to the housing.

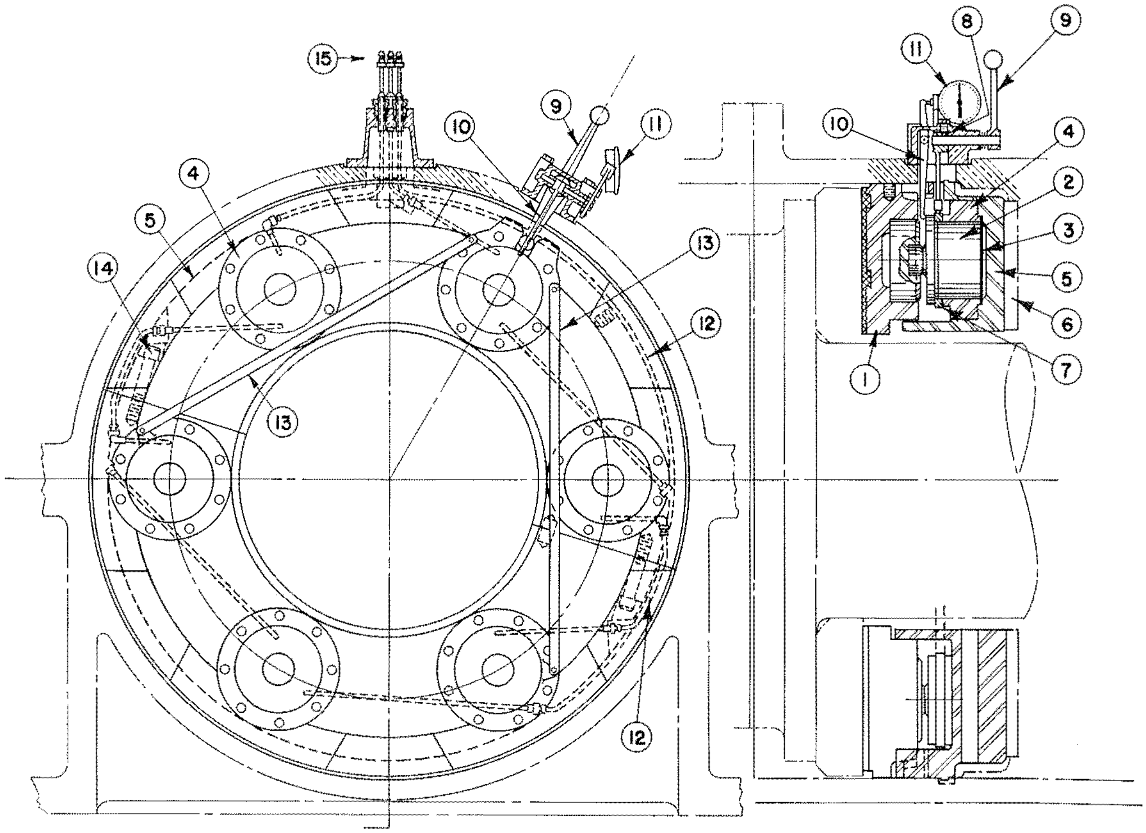


Figure 70
Thrust Meter, Base-Ring Cell Plate Type. 1, Shoe. 2, Piston. 3, Diaphragm. 4, Cell Ring. 5, Cell Plate. 6, Filler Piece. 7, Feeler Ring. 8, 9, Feeler Ring Lever and Handle. 10, 11, Indicator Lever and Indicator. 12, High Pressure Oil Tubing. 13, Links connecting Feeler Rings. 14, Screws joining halves of Cell Plate. 15, High Pressure Oil Connections.

Three Principal Types

The three principal types of Kingsbury Thrust Meters are the *Base-Ring Cell Plate Type*, the *Cage Type*, and the *Two-Shoe Type*. These are shown respectively in Figures 70, 71, and 73.

Base-Ring Cell Plate Type

In this form of Thrust Meter (Figure 70), the leveling plates and base ring are temporarily replaced by a cell plate of similar dimensions, and the thrust from each shoe is applied directly to one of the meter pistons. There are as many cells as there are shoes (usually six), all connected to one hand pump and to one gauge. The oil passages are shown dotted in the transverse section.

Each piston has a flange, and a *feeler ring* (7) is located between the flange and the *cell ring* (4). When the oil pressure balances the thrust the feeler ring is free, and that fact may be checked by moving the *feeler handle* (9). The arm (10) bearing against the piston and connected to the *dial indicator* (11) shows the exact movement of the piston. During a test, the oil pump and release valve are operated when necessary to keep the dial indicator reading constant.

This type of Meter is adaptable to any mounting for equalizing bearings. Another set of cells may be mounted suitably to measure either the astern thrust or the film-reaction from the astern shoes when running ahead.

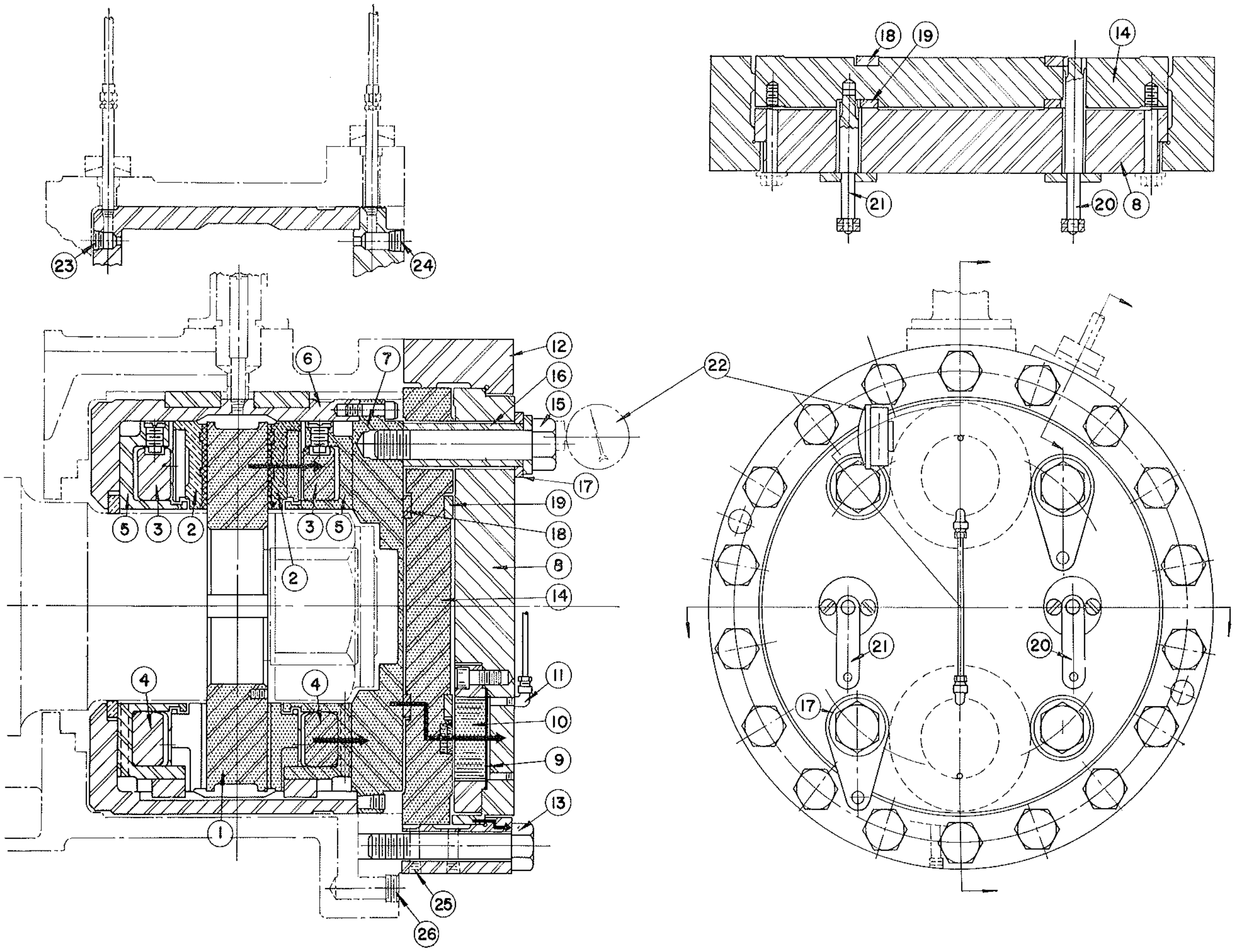


Figure 71

Thrust Meter, Cage Type, showing transmission of AHEAD thrust. 1, Collar. 2, Shoe. 3, 4, Leveling Plates. 5, Base Ring. 6, Cage. 7, Cage Cover. 8, Cell Plate. 9, Diaphragm. 10, Piston. 11, High Pressure Oil Connection. 12, Thrust Ring. 13, Bolt. 14, Thrust Plate. 15, Spacer Bolt. 16, Spacer. 17, Feeler Washer. 18, 19, Feeler Rings. 20, 21, Feeler Handles. 22, Dial Indicator. 23, 24, Lubricating Oil Inlet Connections. 25, 26, Oil Drains.

Cage Type

In the *Cage Type*, (Figure 71), the standard thrust bearing elements are temporarily replaced by reduced-size elements enclosed in a steel *cage* (6) which can slide freely in the housing bore. The meter parts are attached to the forward end of the cage.

Under *ahead* thrust, the forward *base ring* (5) acts as a unit against *cage cover* (7), from which the thrust is transmitted through the *thrust plate* (14) to the pressure cell *pistons* (10). The transmission of thrust from base ring to pressure cell is marked by the heavy arrows. From *cell plate* (8) the force is transferred through the stationary *thrust ring* (12) and bolts (13) to the housing.

The cell plate is slightly movable; ahead for ahead thrust, aft for astern thrust. This fact is utilized to employ the same pressure cells for measuring also *astern* thrust (Figure 72). The force is then transmitted from the cage through the *spacer bolts* (15) to the cell plate (8); thence through the pistons (10) and the edge of the thrust plate (14) to the housing. (Parts transmitting thrust to the Pressure Cell are grayed.)

The movement of the cage and cell plate is sufficient to ensure that the *feeler rings* (18), (19) and *feeler washer* (17) shall be either definitely free or definitely tight, according to the direction of thrust and the amount of oil pumped into the pressure cells. For more refined checking, the *dial indicator* (22) is mounted to show the exact movement of the pistons which results from pumping or releasing oil.

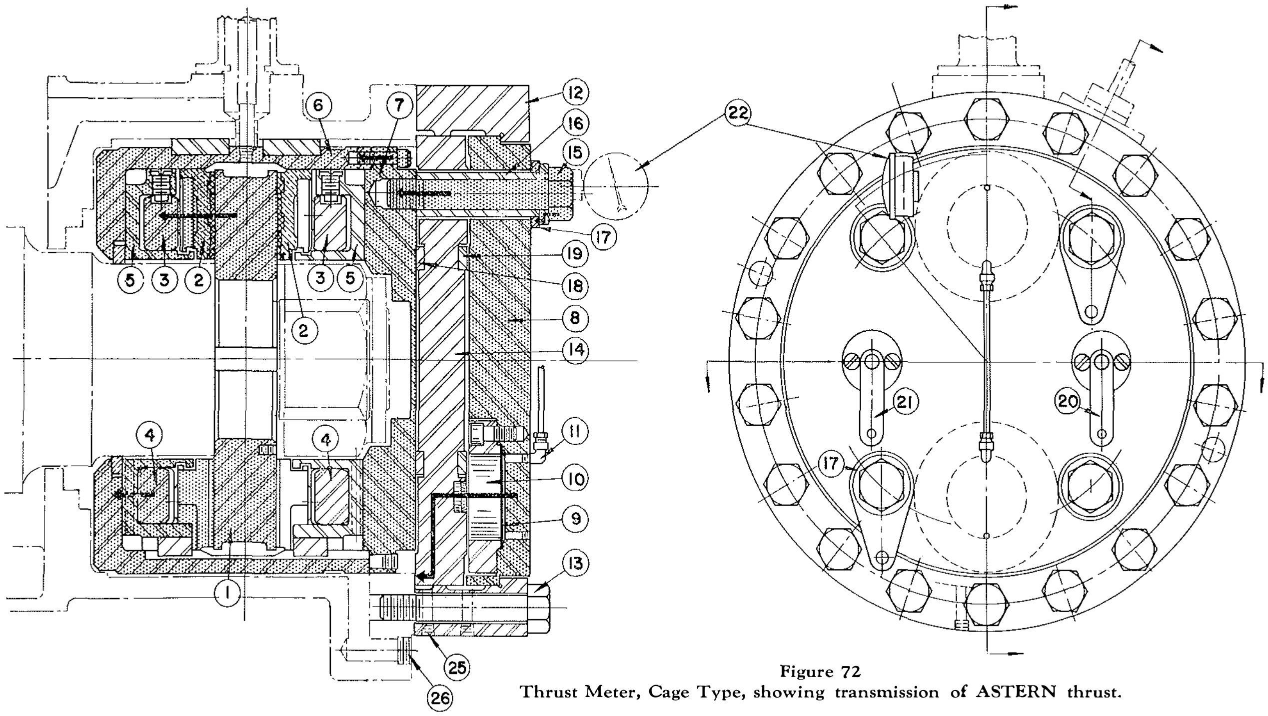
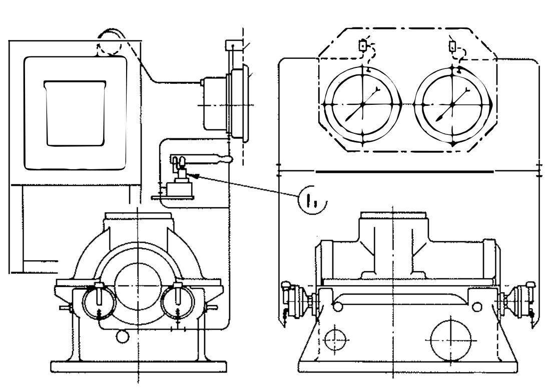


Figure 72
Thrust Meter, Cage Type, showing transmission of ASTERN thrust.



Two-Shoe Type

In Figure 73 is shown a simple adaptation of the pressure cell principle to two-shoe bearings.

The usual jack screws are removed, and in the place of each is substituted a hollow *thrust screw* (1) whose outer end is expanded to contain the *pressure cell* with its *piston* (6) and the *diaphragm* (7). A movable *piston pin* (2) inside the hollow thrust screw transmits the thrust from the thrust pin to the piston (6). A toothed *feeler ring* (4) and a *dial indicator* (12) are used to test the axial freedom of the pistons.

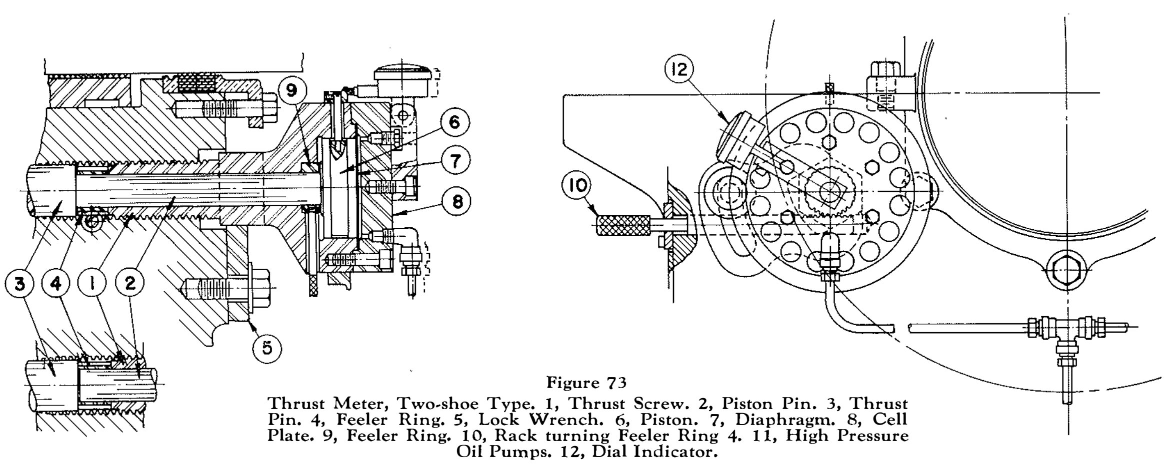


Figure 73
Thrust Meter, Two-shoe Type. 1, Thrust Screw. 2, Piston Pin. 3, Thrust Pin. 4, Feeler Ring. 5, Lock Wrench. 6, Piston. 7, Diaphragm. 8, Cell Plate. 9, Feeler Ring. 10, Rack turning Feeler Ring 4. 11, High Pressure Oil Pumps. 12, Dial Indicator.

Corrections to Meter Readings

With the Base-Ring Cell Plate Type, a small pressure is carried in the cells on the non-working side, in order to steady the meter readings and to avoid need of oil-volume adjustment during crash-reverse tests and other sudden maneuvers. The readings computed for the non-working side must therefore be subtracted from the simultaneous readings computed for the working side.

Where a Cage Type meter is used, the gauge reading indicates net thrust, allowing automatically for the oil-film reaction on the non-loaded side of the collar.

For all types, the rake of the shaft requires that the axial component of the rotating weight be added to the observed net thrust ahead, or subtracted from the net thrust astern.

Estimating Thrust Shaft Diameter for Given H.P. and R.P.M.

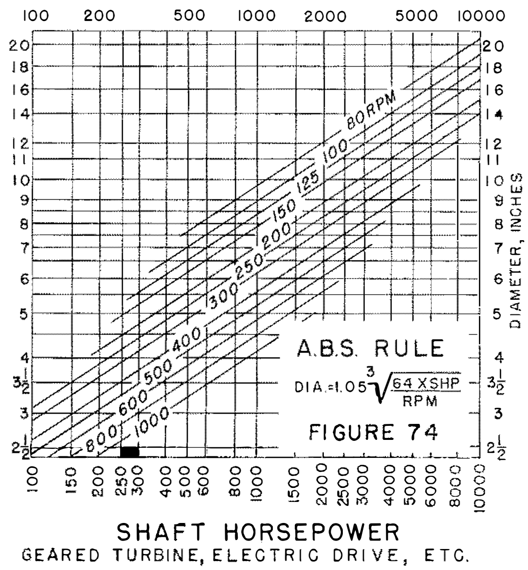
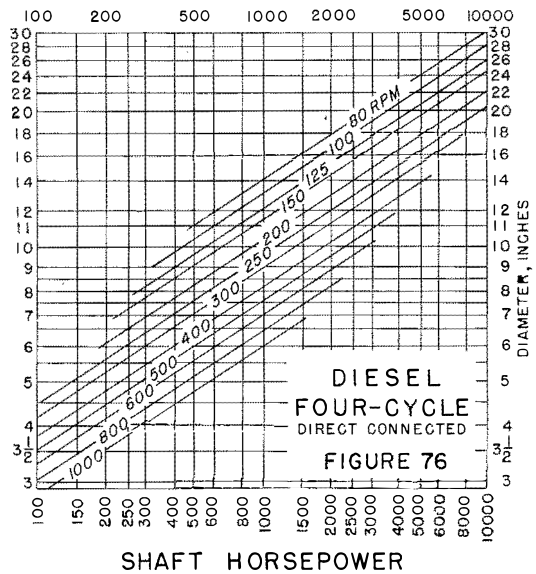


Figure 74

Figure 74 is based on American Bureau of Shipping rules for shafting under substantially uniform driving torque. It also covers Diesel drives using hydraulic or electric couplings.

Figures 75, 76

Figures 75 and 76 are based on actual practice for a large number of installations.



CAUTION

Torsional vibration calculations for the complete rotating assembly in a given vessel may call for shaft diameters considerably larger than shown by the charts.

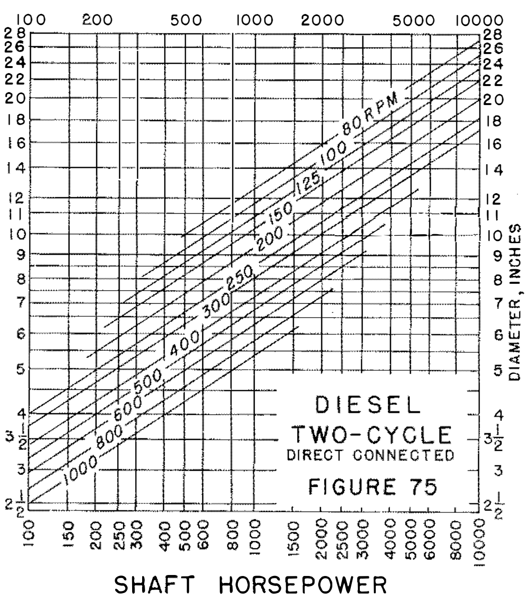


Table III

Thrust Factor is based on propeller efficiency and wake fraction.

Deeper drafts use higher factors.

Higher revolutions use lower factors.

*Use maximum power and speed running free. Factors allow for towing conditions.

Estimating Propeller Thrust

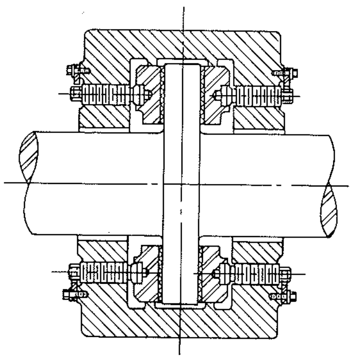
TABLE III

Type of Vessel	Thrust Factor	
	One Screw	Two Screws
Passenger (Ocean)	265-290	205-230
Passenger (Coastwise)	265-300	210-235
Passenger (River)	230-260	190-220
Freighter (Ocean)	250-300	210-240
Freighter (Coastwise)	260-300	225-250
Freighter (Great Lakes)	280-320	
Tanker (Ocean)	260-290	215-250
Tanker (Coast)	270-310	2[illegible]5-250
Ferry (Single or Double Ended)	200-250	180-230
Yacht	200-260	170-220
Towboat *(Ocean)	240-290	200-250
Towboat *(Harbor)	250-300	200-240
Barge (River)	260-310	200-240

Naval vessels are purposely omitted.

Formula: $\text{Thrust (lbs.)} = \frac{\text{Thrust Factor x S.H.P.}}{\text{Vessel speed in knots}}$

Special Thrust Bearings



On a previous page it was mentioned that, in situations where standard Kingsbury Thrust Bearings cannot be applied, some special design can usually be found to meet the need.

The following illustrations show bearings which were actually developed to satisfy particular requirements, usually as to space or lubrication. Many others could be shown; and we are always ready to discuss ways of meeting unusual conditions.

Thrust Bearing Without Journal Bearing

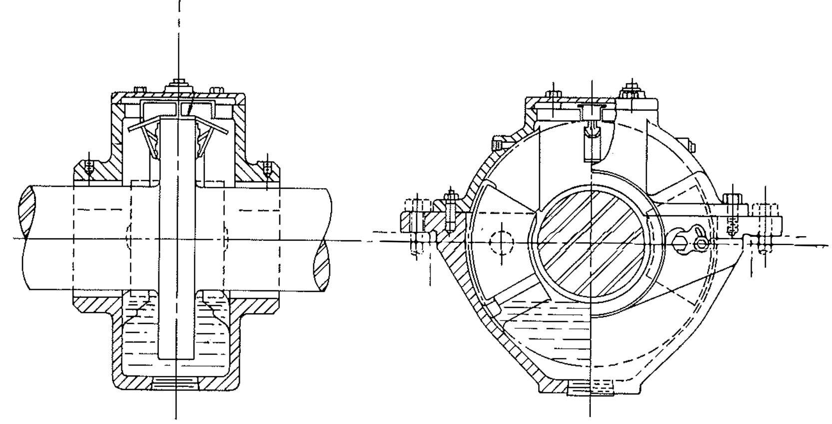


Figure 77
Showing how a 2-shoe Kingsbury bearing can be mounted without journal bearing. Compare Figure 20, page 13.

Sometimes it may be desired to locate the thrust bearing where an adjacent journal bearing is already provided. Figure 77 shows how this can be done with a two-shoe bearing.

The jack screws bear directly on the shoes. Lubrication is by bath and scraper, as in the standard type.

This arrangement is sometimes used in the crankcases of Diesel engines, as illustrated in Figure 20, page 13. Since crankcase oil is used, no closures are required around the shaft.

Special "Short" Designs

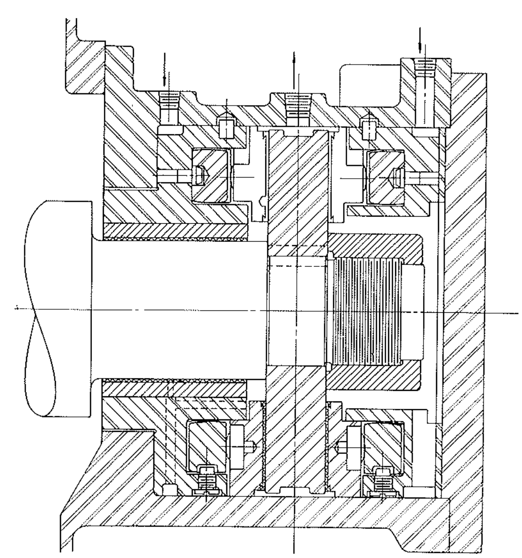


Figure 78
Compact design for thrust and journal bearing at forward end of gear or motor housing. Special base ring functions also as a journal bearing shell.

For locations requiring unusually short overall length of thrust and journal elements there are several possibilities.

One is the compact six- or eight-shoe arrangement in Figure 78. It is there shown as applied to the forward end of a reduction gear housing, but it could be adapted to other locations with special gear arrangements. The forward base ring is of standard form: the after base ring is a fit in the housing and is shaped to contain and support the journal bearing.

If desired, the journal diameter could be increased, and smaller shoes of larger bore substituted on the after side of the collar. The

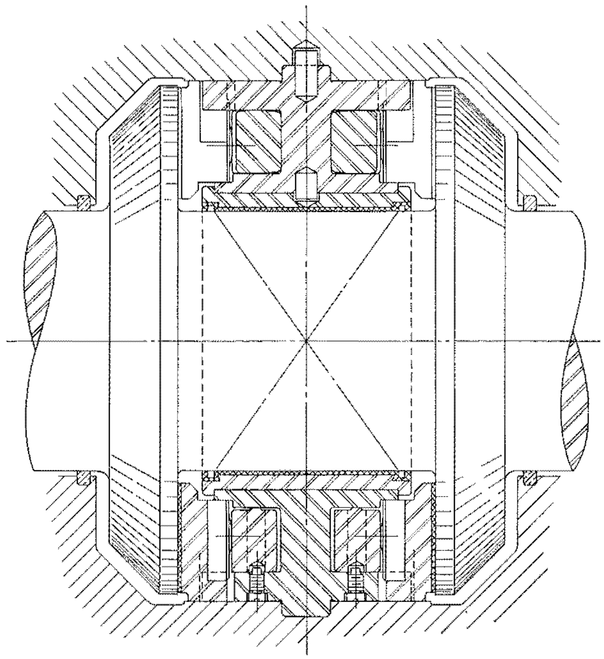


Figure 79
Compact equalizing thrust bearing built around a split journal bearing. The two collars are integral with the shaft.

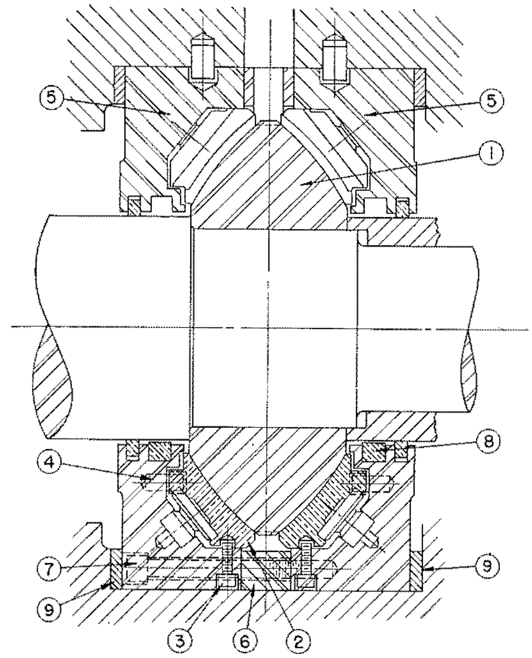


Figure 80
Double spherical bearing. It combines thrust and radial functions. 1, Collar. 2, Shoes. 3, Screws retaining shoes. 4, Shoe Stop Pins. 5, Base Rings. 6, Spacer Ring. 7, Bolts holding base rings together. 8, Oil baffles. 9, Filler Pieces.

reduced shoe area would still be ample for the astern thrust.

For special locations, usually outside of the gear case, the double-collar-plus-journal-bearing arrangement (sometimes called "straddle type") shown in Figure 79 is a space saver. The split journal bearing shell is expanded to serve also as a double base ring carrying leveling plates and two sets of shoes. It is held fore and aft by a massive central flange fitting into a groove bored into the housing.

For extreme shortening, the thrust and radial functions may be combined in a double spherical bearing, such as is shown in Figure 80. Such bearings are used also in steam and gas turbines for utmost compactness. Their friction is about equal to the combined friction of a flat thrust collar and the conventional journal bearing. The oil flow is similar to that with six-shoe "flat" bearings.

As a matter of interest, large "single" spherical bearings on vertical shafts are sometimes used to carry the much greater rotating loads of large hydroelectric generators.

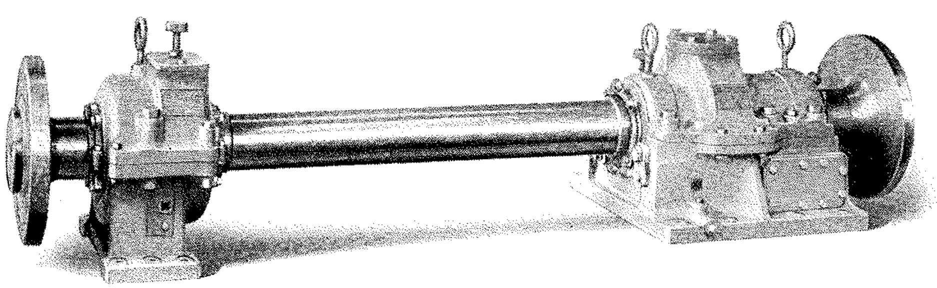


Figure 81
Line Shaft Bearing and Two-Shoe Thrust and Journal Bearing.

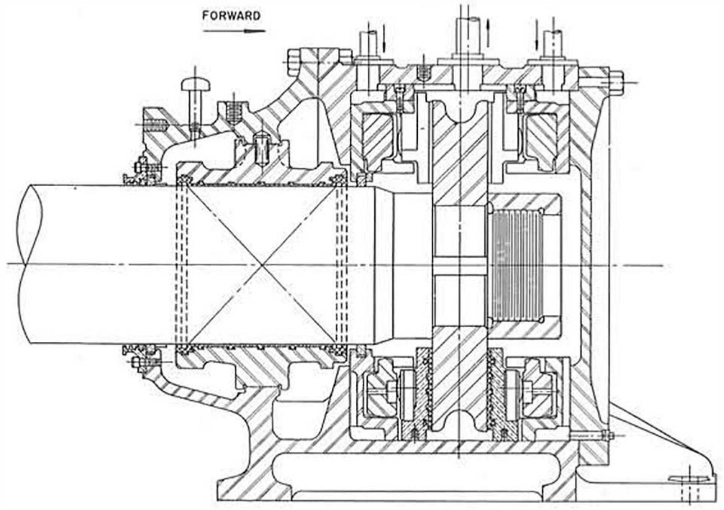


Figure 82
An unusual housing arrangement for the journal bearing and thrust bearing at the forward end of an electric propulsion motor.

Separate Mounting with Electric Motor

An adaptation to special requirements with electric motor drive is shown in Figure 82. Heavy duty required the six-shoe thrust, also a self-aligning journal bearing supporting the forward end of the rotor shaft. Available clearances forward and aft dictated the unusual shape of the housing. The end cover is designed to be removed and replaced by a thrust meter if desired.

Thrust Bearing at After End of Gear Case

Sometimes it is best to locate the thrust bearing at the after end of the gear housing. An example of this arrangement is shown in Figure 83. In this design, facing collars are used; see Figure 62: but they might be omitted and the bearing shortened.

The journal bearings also are by Kingsbury. By reason of lubrication features not shown, the short shells are ample to carry their loads.

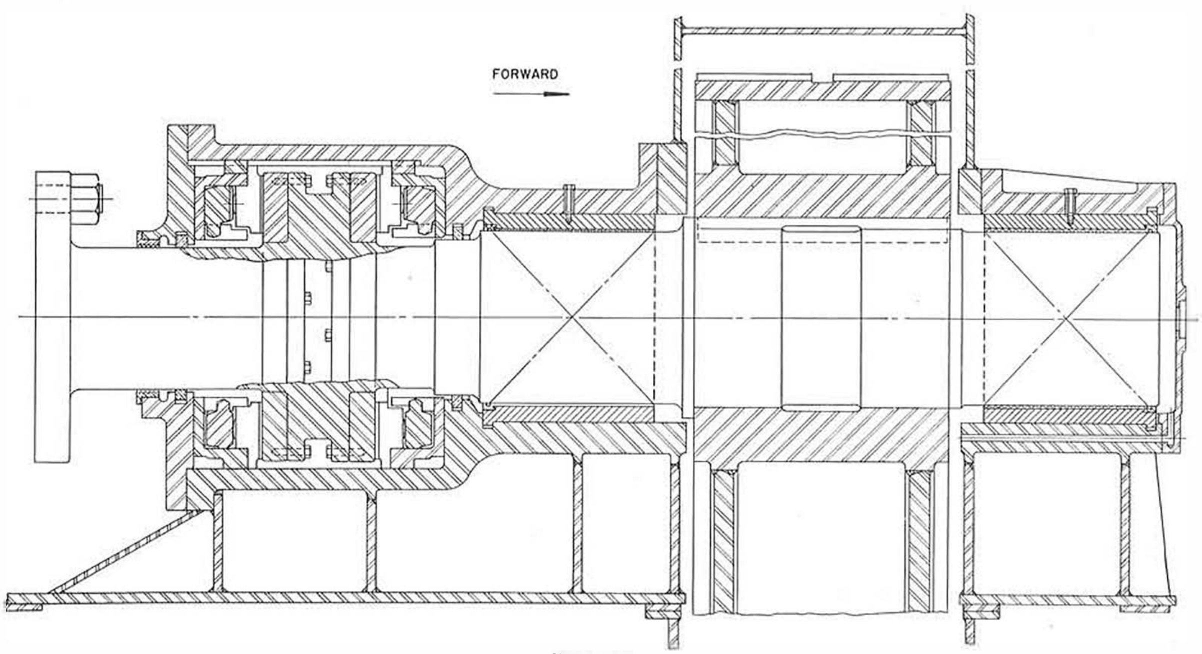
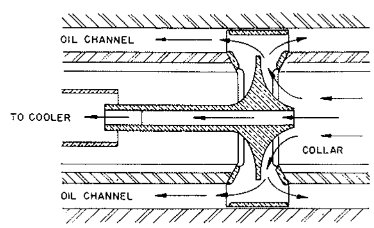


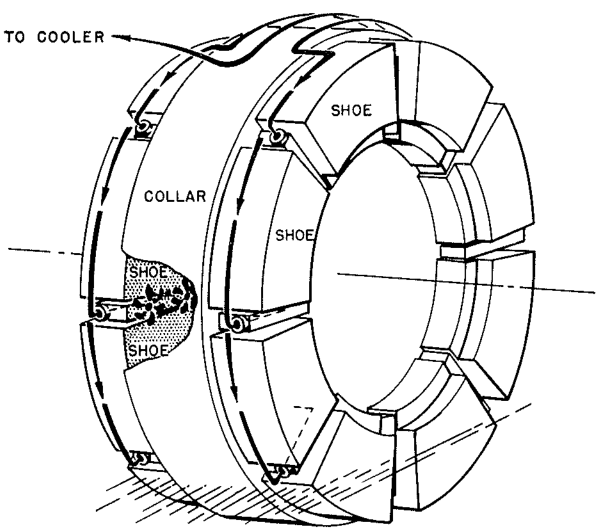
Figure 83
Six-shoe, equalizing thrust bearing at after end of gear case.

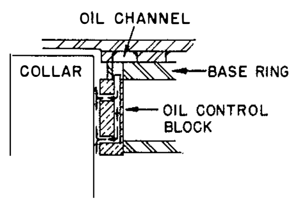
Large Pedestal-Type Thrust Bearing with Automatic Lubrication

Below is illustrated a recent improvement over the lubricating arrangement shown on page 26, for six- or eight-shoe thrust bearings on large ships. The bearings on that page were designed to run full of oil, which is piped under pressure from an external source, discharged at the top and returned. That involves depending on a pump and extended piping. It also requires caring for some leakage of oil at the ends of the bearing housing.

These features are avoided with the novel system of automatic circulation shown in Figure 84. By this system, oil is picked up by the collar rim from the bath level below the shaft, as in standard two-shoe bearings. At the top it is taken off by a special three-way scraper, and is delivered partly to the cooler, partly to channels so arranged that all of the eight shoes on each side are flooded.







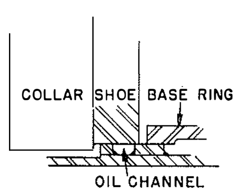


Figure 84
Principle of thrust bearing lubrication, "Atlantic Seaman" and class. At top, plan section of scraper showing 3-way distribution of oil. Left, diagram of distribution to oil control blocks. Lower right, sections showing oil channel surrounding shoes and base ring, and flow of oil into one control block.

Figure 84 shows the principal details. Oil on the middle portion of the width of the collar rim enters a shallow groove in the underside of the scraper. At the end of the groove it meets a dam, and is pushed up by the oil behind it into a spout leading to a cooler mounted at the side of the housing base. From the cooler it returns to the bath.

Oil on both sides of the middle strip is taken off by the scraper and deflected, left and right, into channels between rings welded into the housing. These rings surround the shoes and base rings, which rest loosely in them. Each pair of rings, on each side of the collar, forms a channel, which is continually filled with oil flowing down as long as the shaft turns in either direction.

These streams of oil enter holes drilled lengthwise in brass "oil control blocks," which are secured to the base rings and loosely fill the spaces between the shoes. Entering the control blocks, the oil issues from drilled holes in streams impinging on the collar faces. The holes are of various diameters to suit the "head" of oil above them; hence each shoe receives an equal share of oil for lubrication and cooling. Since the top and bottom spaces receive oil anyway, no control blocks are needed in them.

This system was first applied to the "Atlantic Seaman" class supertankers, each having a single screw and a pedestal-type, 43-inch, eight-shoe Kingsbury thrust. On those ships the thrust housing contained no journal bearing: instead, two separate Kingsbury line shaft bearings were located just forward and aft of the thrust; and a third Kingsbury line shaft bearing was placed farther aft.

Spare Parts

A Kingsbury Bearing correctly chosen, properly aligned and supplied with clean oil, is practically indestructible for the life of the ship. However, spare parts are customarily carried as a matter of insurance. Usually these are governed by the regulations of the American Bureau of Shipping. They always include the shoes for ahead thrust, sometimes a lower half journal bearing shell and a cooling coil where fitted, and occasionally the thrust collar where it is separate from the shaft.

Damage that may occur is usually limited to the ahead shoes, or perhaps a cooling coil. Other parts practically never need replacement. This may be contrasted with the fact that, with ball or roller bearings, a failure calls for replacement of the entire bearing.

Currently, for propeller thrust bearings, A. B. S. Spares consist of all the shoes for the ahead side of one bearing only (same for single screw or multiple).

Data Needed With Inquiries

To make specific recommendations for propeller thrust applications, we should have the fullest possible information on conditions to be met, as follows: —

(A) Thrust load at full power (per shaft)

(B) Journal bearing load (if unusually heavy)

(C) Revolutions per minute at full power

(D) Shaft diameter through bearing

(E) Solid or split base rings (6-shoe only)

(F) Is a removable lower half shell required in journal bearing?

(G) Will thrust collar be integral with shaft?

(H) Is cooling water available?

(J) Is external lubricating system available?

(K) Which machinery arrangement, pages 8 and 9, applies? (Give space limitations, where applicable.)

If the thrust load is not known, it may be estimated closely enough from the formula and factors on page 42. For more accurate calculations, particularly for towboat applications, we should have the following additional data:

Single, twin or multiple screws?

Full power (s. h. p. per shaft)

Corresponding vessel speed (running free, for towboat)

Propeller diameter

Propeller pitch

Number of blades

Shaft diameters, if not known, may be approximated from data on page 42.

Standard Guarantee

Any bearing or part furnished by us, which shall prove defective in design, material or workmanship, within one year after installation and test, will be replaced without charge f.o.b. Philadelphia, if returned to our factory. No allowance will be made for labor or other expense in connection therewith unless authorized in writing by an officer of the Company.

For oil coolers and cooling coils, in accordance with usual trade practice, there is no specific guarantee period.

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