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MOD-1 WIND TURBINE GENERATOR ANALYSIS AND DESIGN REPORT

General Electric Company
Space Division

May 1979

Prepared for
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Lewis Research Center
Under Contract NAS 3-20058

for
U.S. DEPARTMENT OF ENERGY
Office of Energy Technology
Division of Distributed Solar Technology

NOTICE

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Advanced Energy Systems
Philadelphia, Pennsylvania 19101

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Washington, D. C. 20545
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TABLE OF CONTENTS

APPENDIX A

CODE VERIFICATION

Abstract

This appendix includes a summary of the code verification task as it appeared in G. E. Space Division document No. 77SDS 4203, dated January 24, 1977, entitled "1500 kW Wind Turbine Generator Program Preliminary Design Report." The objective of the code verification task was to verify the computer codes used for rotor loads analysis by comparing predictions with experimental measurements from the Mod 0 WTG. The goal was to demonstrate the capability of the codes to predict load magnitudes within 20 percent, and to duplicate the harmonic content.

2.7 CODE VERIFICATION

A code verification task was performed to verify where practical all computer codes related to the rotor loads analysis and the coupled system dynamics. Data from the Mod-0 WTG during four operational conditions was used to compare experimental measurements of loads and deflections with the code predictions. The goal of the code verification was to demonstrate the capability of the codes to predict magnitudes within 20 percent and duplicate harmonic content (1P, 2P, etc.). This code verification was performed using the same analytical codes that are being used for Mod-1.

The flow of the Mod-0 verification analysis is shown in Figure 2.7-1 and follows the same methods being used for Mod-1 analysis. A model of the tower, bedplate and shaft was

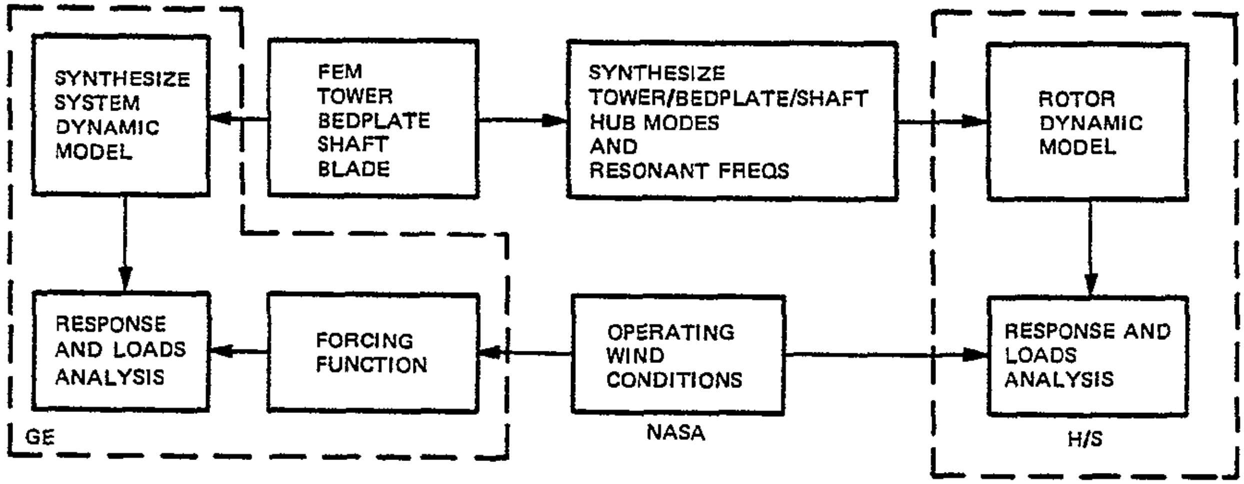

Figure 2.7-1. Mod "0" Code Verification

generated and provided to Hamilton Standard for their code verification using the F762 code. At GE, a model of the complete system including the rotor was developed and analyzed using the DYNAMO codes. The measured Mod-0 data was provided by NASA to define the wind conditions.

In this section, the Mod-0 WTG is described briefly, the selected Mod-0 operationing conditions are reviewed, the analytical model is described, comparison of measured model data and analytical predictions are given and predicted loads are compared with measured values.

2.7.1 DESCRIPTION OF THE MOD-0 WTG

The following excerpt from the NASA Report TMX-71601, "Early Operation Experience on the ERDA/NASA 100 kW Wind Turbine," provides a general overview of the Mod-0 WTG.

> "The 100 kW Experimental Wind Turbine is a part of the national wind energy program under the direction of the Energy Research and Development Administration (ERDA). The NASA Lewis Research Center has designed, built and erected this machine near Sandusky, Ohio, and is currently testing it to obtain engineering data on large horizontal axis wind turbines.
>
> The wind turbine has a 125-foot diameter, two-bladed rotor which drives a 100 kW capacity synchronous generator through a step-up gear box. The rotor is positioned

downwind of a 100-foot steel truss tower, as pictured in Figure 2.7-2. The rotor is designed to operate at a constant speed of 40 rpm, and it drives a 480 Volt, 60 Hz, three-phase generator at 1800 rpm. Constant rotor speed is maintained by controlling the blade pitch angle with an active feedback control system. The rotor, generator, transmission and associated equipment are mounted in a nacelle, Figure 2.7-3, which can be yawed to align the rotor with the wind. Power, instrumentation and control connections to the ground are made through slip rings.

The turbine was designed to begin generating power in winds of 10 mph (100 feet), and produce 100 kW at a wind velocity of 19 mph. In winds above 18 mph, the generator continues to operate at a 100 kW output by adjusting the pitch of the rotor to spill the excess wind energy. When the wind velocity exceeds 40 mph, the blades are feathered to bring the rotor to a stop in a horizontal position. A brake is then applied at the high-speed drive shaft to lock the rotor blades against rotation.

Final assembly of the machine was completed in September, 1975; it began operation in October. In December, 1975, the machine first achieved its design speed of 40 rpm and produced 100 kW of power. During the course of these initial operations, data was taken on the rotor, blades, the nacelle and the tower."

" The 100 kW wind turbine data system provides approximately 100 channels of real time continuous information. Some of this data is discussed below.

1. Wind velocity and azimuth from a meteorological tower

2. Wind velocity and nacelle yaw angle relative to the wind, from the wind turbine nacelle

3. Tower deflections x and y

4. Nacelle accelerations x, y, and z at the rotor shaft bearing support nearest to the rotor

5. Rotor blade pitch angle

6. Rotor blade bending moments, indicating beamwise (M_m) and chordwise (M_n) bending at two stations along the blade span

7. Rotor shaft torque M_z and bending moments Mx and M_y

8. Rotor speed and blade position

9. Alternator output.

Figures 2.7-4 and 2.7-5 give a schematic representation of these measurements, their location on the wind turbine, and the sign convention of each."

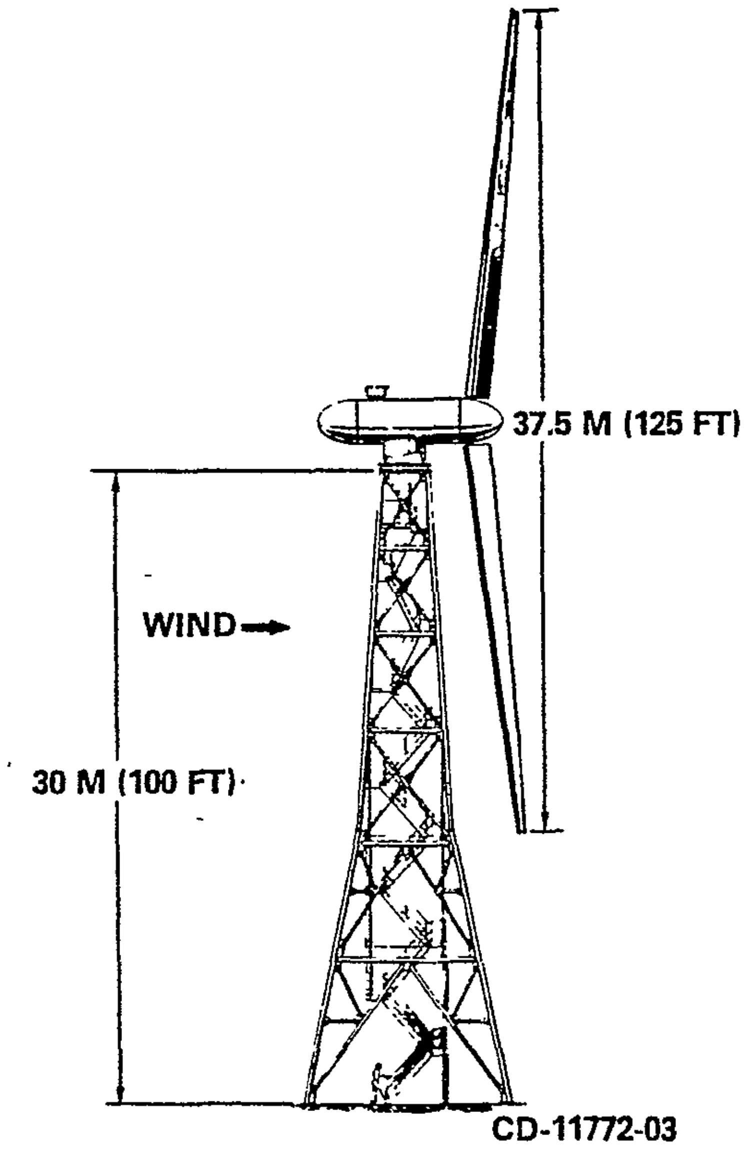

Figure 2.7-2. 100-Kilowatt Experimental Wind Turbine

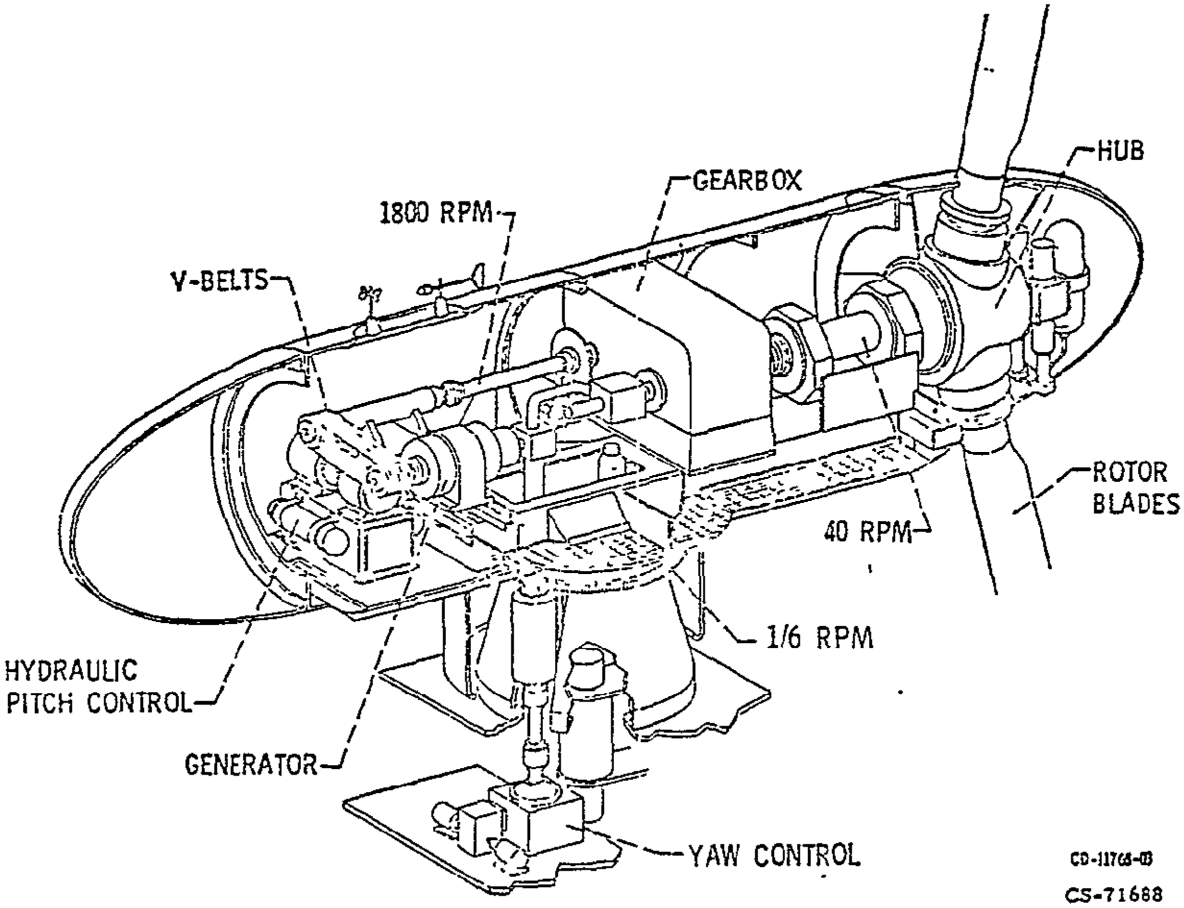

Figure 2.7-3. 100 kW Experimental WTG 100 kW Wind Turbine Drive Train Assembly

After initial operation of the Mod-0, two changes were made to the configuration. First, the stairs and external elevator, Figure 2.7-2, were removed to reduce the amount of velocity retardation behind the tower and, hence, reduce the dynamic loads on the blade. After additional operational data were obtained, the operating speed was reduced to 20 rpm by changing the pulley ratio in the drive train, Figure 2.7-3.

Included in the Mod-0 system are two structural non-linearities which may have a significant influence on the dynamic loads of the system. These are indicated in Figure 2.7-6. Within the low speed shaft of the generator drive train, a Faulk coupling is installed which varies considerably with the torque. At low torque, the coupling is relatively soft as indicated by the slope of the curve in Figure 2.7-6 near zero. As the torque is increased, the stiffness also increases as shown by the slope of the curve. The other non-linearity is in the yaw drive used to align the rotor with the wind. As indicated in Figure 2.7-6, there is a significant amount of free play permitting a yaw rotation of the nacelle. However, in one of the four cases selected for analysis, a "keeper" was installed in the yaw drive to eliminate the free play.

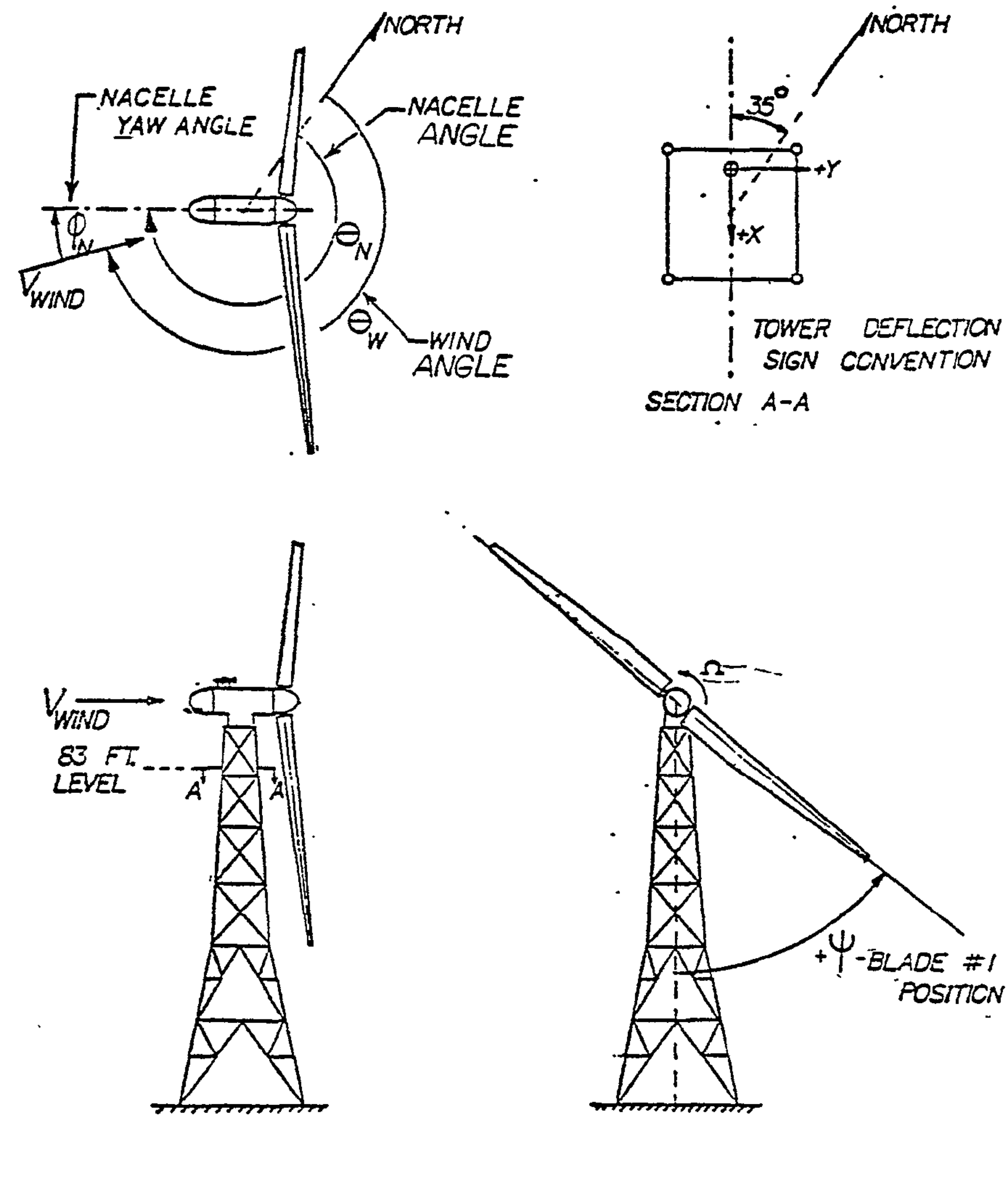

Figure 2.7-4. Mod-0 Sign Conventions and Tower Deflections

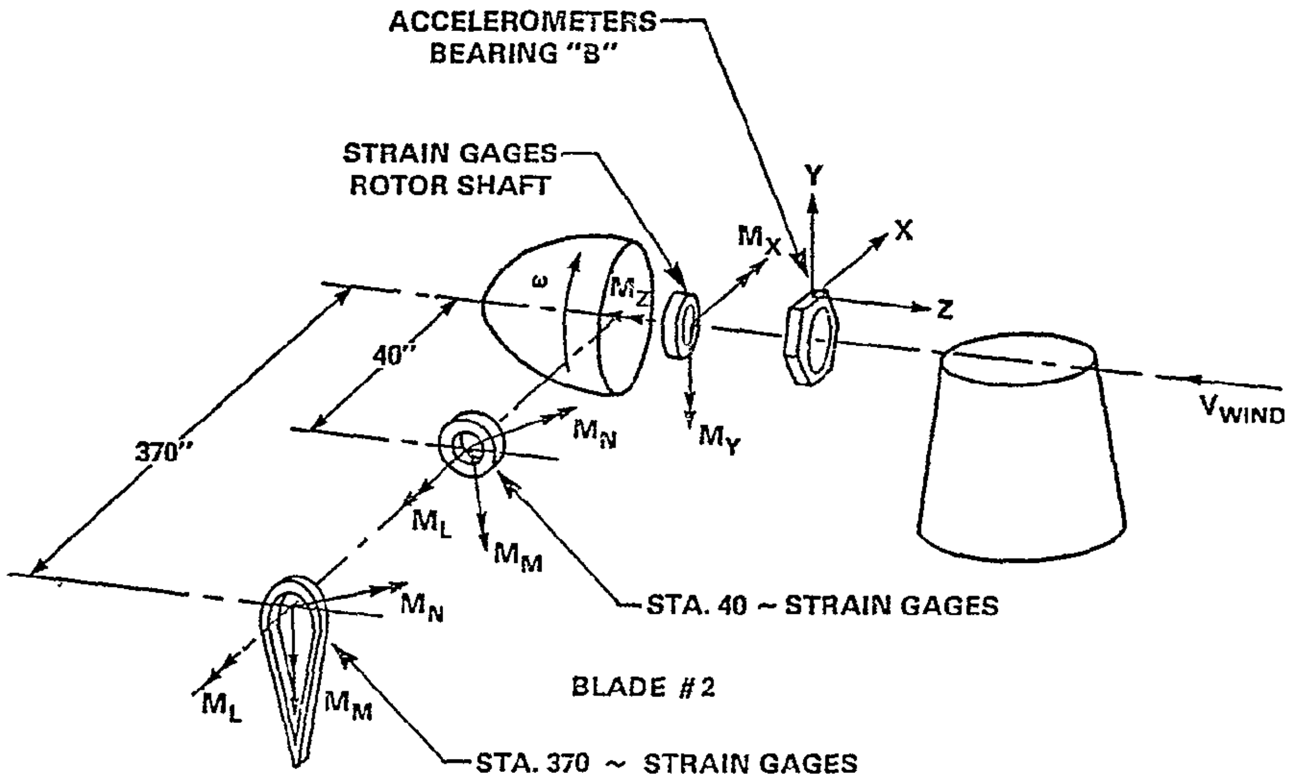

Figure 2.7-5. 100 kW WTG Experimental Measurements

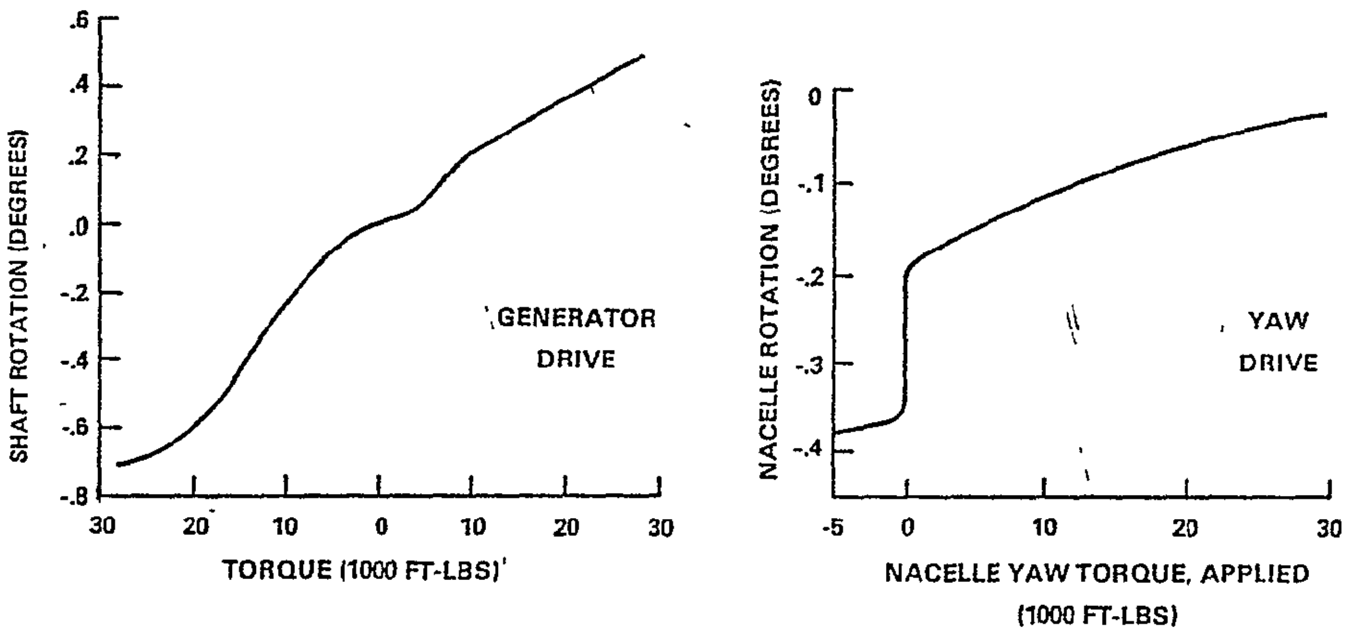

Figure 2.7-6. Mod "0" Nonlinear Characteristics

2.7.2 DESCRIPTION OF MOD-0 LOAD CASES

The four Mod-0 load cases selected for comparison of measured and analytically predicted loads are shown schematically in Figure 2.7-7. The significant characteristics of each case are described below:

Case 1. This case was obtained shortly after initial operation of the WTG and includes a large "tower shadow" effect. The nominal operating condition was at 40 rpm with a wind velocity of 28 mph, producing an output power of 100 kW.

Case 2. This is a gust condition in which the nominal velocity increased from 20 to 35 mph. The stairs were removed, reducing the tower blockage. Nominal operating conditions were 20 rpm with 37 kW of output power.

Case 3. During this condition, large torsional oscillations of the drive train were experienced. The "torque bloom" appears to be a limit cycle oscillation resulting from coincidence of the drive train torsional resonant frequency with a multiple of the rotor speed. Nominal conditions were 41 rpm, 28 kW of output power with the tower stairs removed.

Case 4. In this case, the "yaw keeper" was installed, eliminating the free play in the yaw drive. The stairs were removed providing minimum tower blockage. Nominal operating conditions were 40 rpm, and 98 kW of output power with a nominal wind speed of 25 mph.

During the initial stages of the verification analysis, it became apparent that some of the measurements were conflicting. As a result, "soft" measurements were considered to be the output power, the wind velocity and the wind direction. The rotor power derived from the shaft torque and operating speed was considered to be the best measure of the WTG power. The wind direction was specified by NASA based on their review of the data. The wind velocity measured on the nacelle was considered to be the best velocity measurement. However, the wind velocity and blade pitch angle were considered to be soft and could be varied to achieve the rotor power.

Wind tunnel data on a scale model of the Mod-0 tower both with and without stairs were available to estimate the velocity retardation behind the tower. The data show a considerable scatter depending on the wind direction, Figures 2.7-8 and 9. The average velocity was determined by NASA, using the actual width of the shadow as the averaging distance. Typically, the shadow width is between 1.4 and 1.5 times the geometric width of the tower. On this basis, the velocity profiles shown in Figure 2.7-10 were selected as representative profiles. The profile for the stair configuration is based on measured profile data using a value of 0.3 for the maximum retardation and providing an average velocity ratio over the tower shadow width of approximately 0.65. For the "bare" tower, a simplified notch having a height of 0.8 was selected in that no prevalent profile was evident from the data.* All

*Based on a review of the results from the first Case 4 analysis, it was felt that the tower shadow was slightly less severe than indicated by the wind tunnel measurements. Conse- a second analysis was made using a similar shape with a velocity ratio of 0.76 in the retarded flow area.

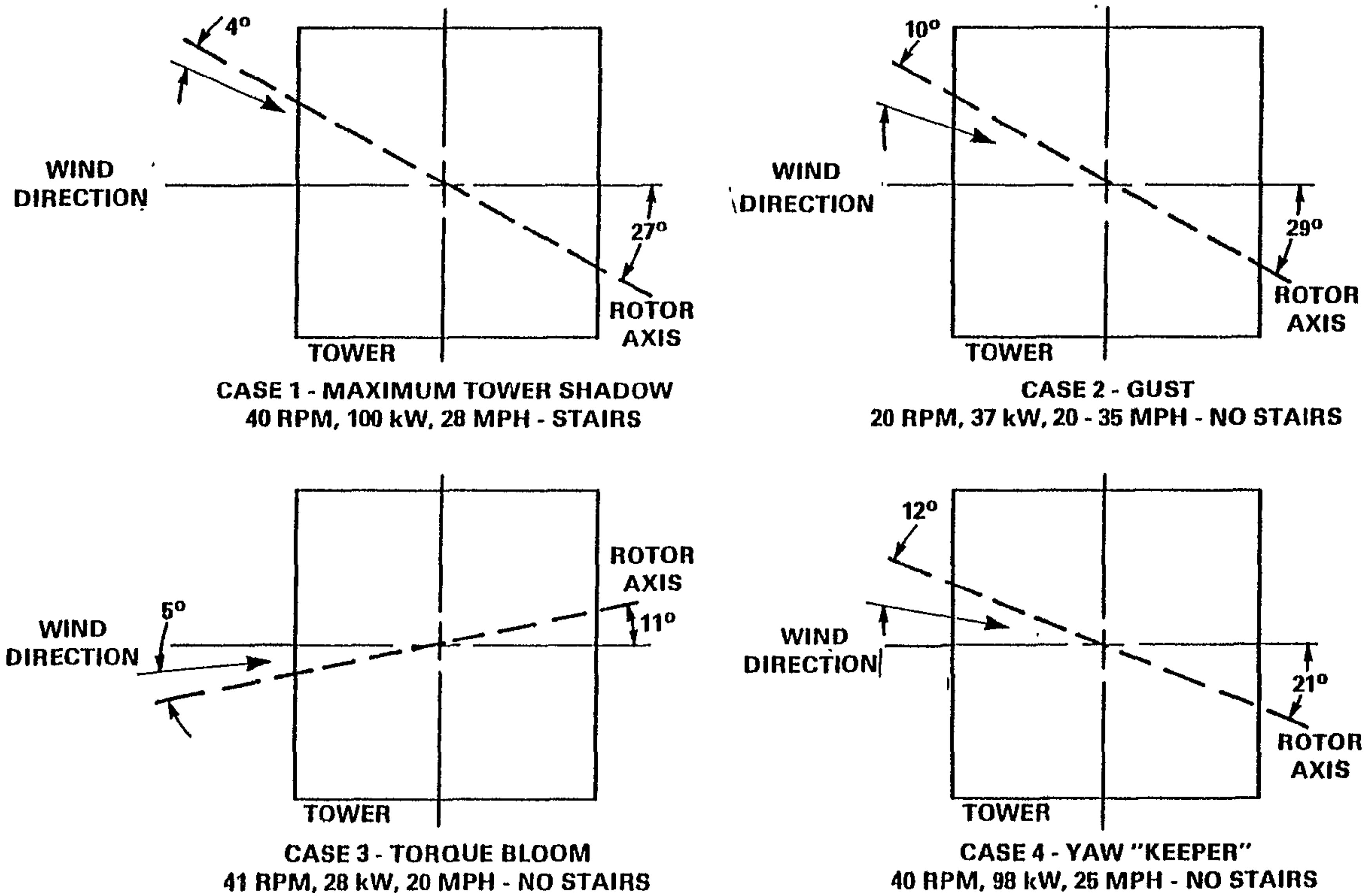

Figure 2.7-7. Schematic of Four Mod 0 Load Cases

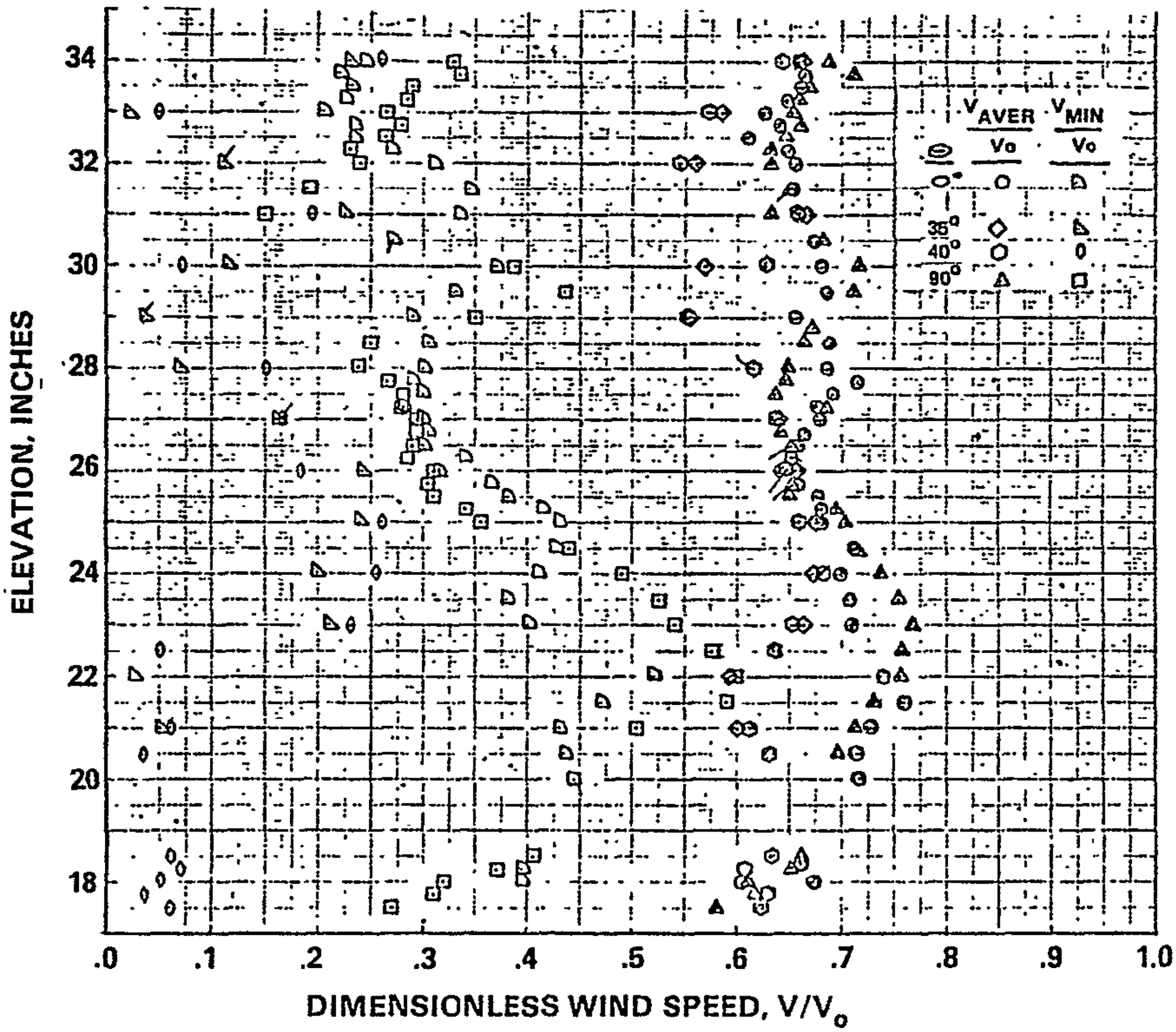

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Figure 2.7-8. Vertical Distribution of the Average and Minimum Wind Speeds in the Wake of the Mod-0 Tower Model 1/25 Scale with Stairs and Elevator Cars

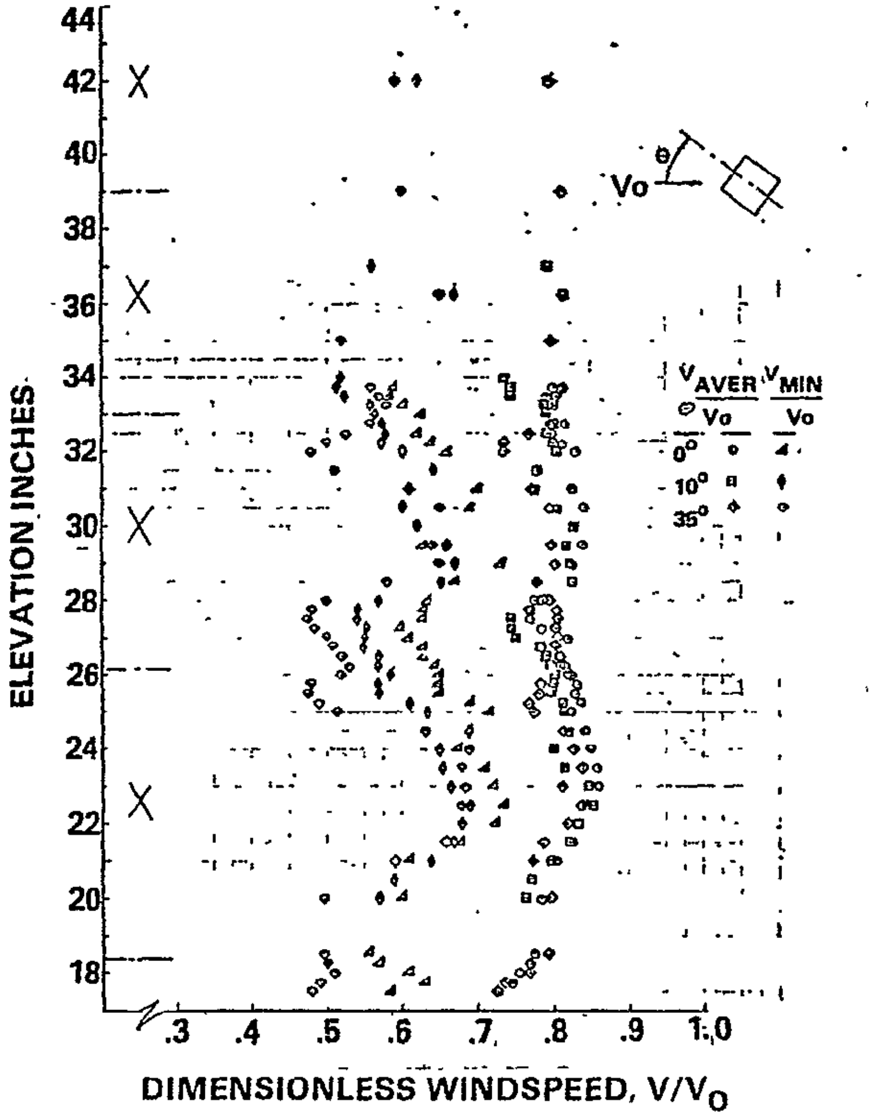


Figure 2.7-9. Vertical Distribution of the Average and Minimum Wind Speeds in the Wake of the Bare Mod-0 Tower Model 1/25 Scale Model - Bare (without stairs and rails)

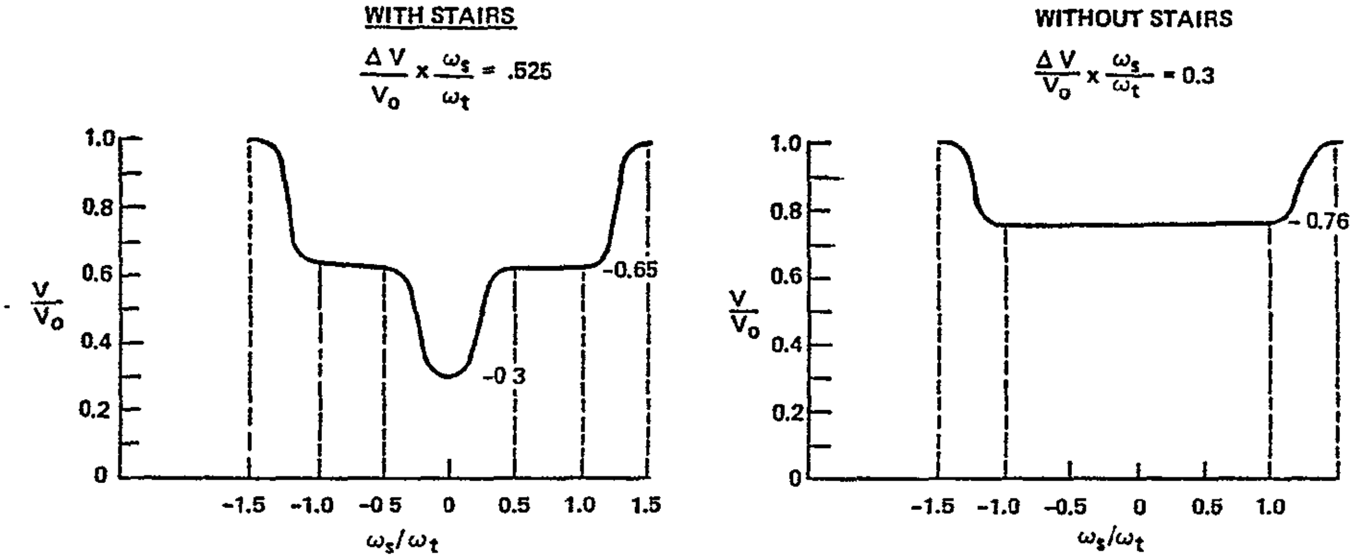


Figure 2.7-10. Mod-0 Tower Shadow Models

profiles consider the shadow width to be 50 percent wider than the tower. As indicated previously, the shadow follows the tower outline providing a tapered shadow.

2.7.3 MOD-0 DYNAMIC MODEL DEVELOPMENT

2.7.3.1 Approach

The development of the Mod 0 dynamic model followed the same methods used in the Mod 1 analysis. This system also divided naturally into substructures for separate modal analysis at bearing attachments located at each rotor blade, at the hub, the low and high speed shaft supports and the bedplate/tower interface. A natural division at the yaw drive was also made. This produced five substructures for the system: tower, bedplate, hub/shaft, and two rotor blades. Each of the substructures was analyzed separately by formulating finite element models to obtain substructure modes and frequencies. Assembly into a complete system model was accomplished by modal synthesis using stiffness coupling through flexible links representing the bearing attachments. The completed model was then used to obtain the coupled eigenvalues and eigenvectors for use in the forced response and loads analysis (see Figure 2.7-11).

Since modal test data existed for this system, the structure was modeled to duplicate the test configuration. This required the nacelle along the east-west axis, blades in a horizontal position and feathered and low speed shaft locked. Comparison to modal test data determined

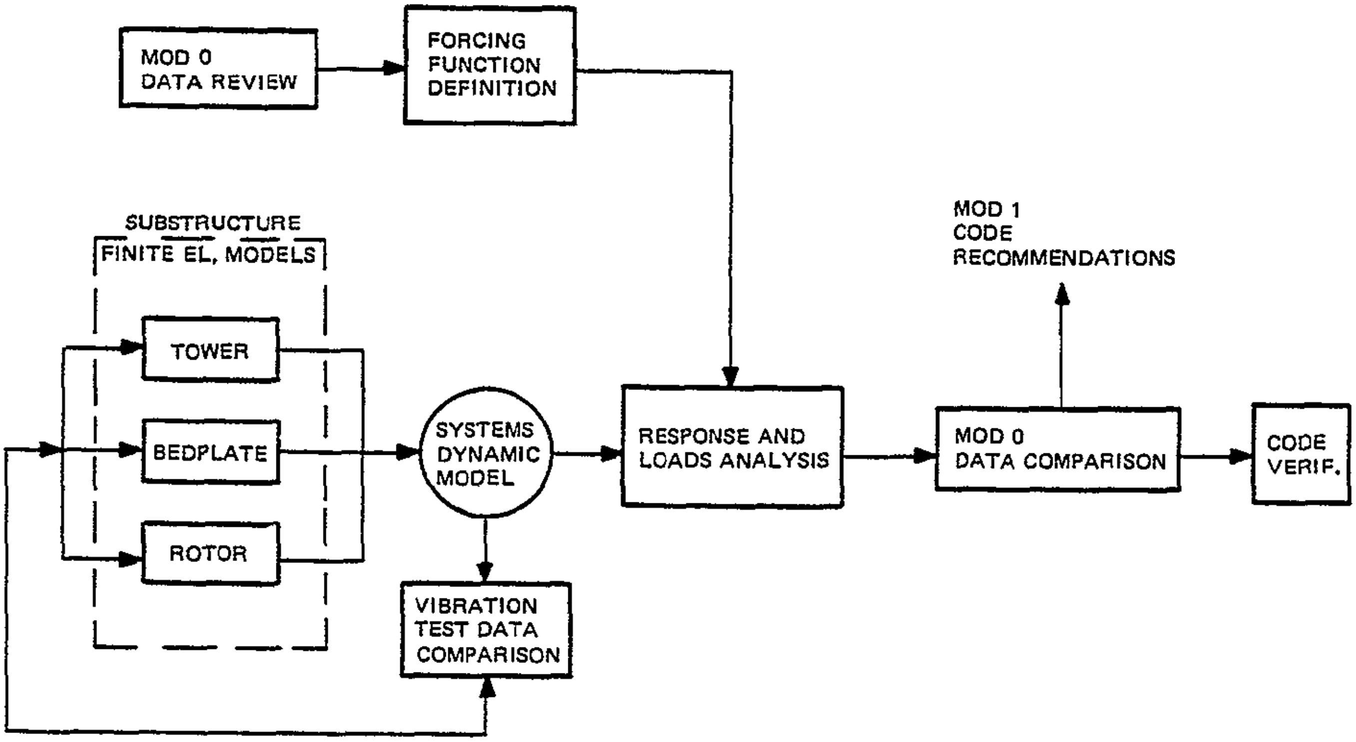


Figure 2.7-11. Mod-0 Dynamic Model Analysis Flow

if any required modifications to the model parameters were required. Development of subsequent models to match the load case conditions required only a change in blade pitch angle, release of the shaft lock and the appropriate nacelle orientation.

2.7.3.2 Analytical Substructures

The test model was developed using data obtained from NASA in the form of drawings, reports and tabulated mass and stiffness data. These data were used directly as input to the substructure models and as checkpoints to verify the model substructures prior to final assembly:

Tower. The tower model was developed using drawings obtained from NASA (CF 758447 through CF 758450). It consisted of approximately 584 bars, rods, and plate elements representing each member of the tower truss and stairway, according to the drawing. These elements were connected at 264 nodes (1512 degrees of freedom). A computer plot of this substructure is shown in Figure 2.7-12. Through modal compression methods the number of nodes were reduced for inertia loading to 29, most with 3 degrees of freedom. Weight distribution was provided by the computer program with the gross weight adjusted to match the weight data given by NASA TMX-71979 with differences of less than 1 percent. Eigenvalue solutions with the tower cantilevered at the base were made as input for the system synthesis. This model was used for all load cases.

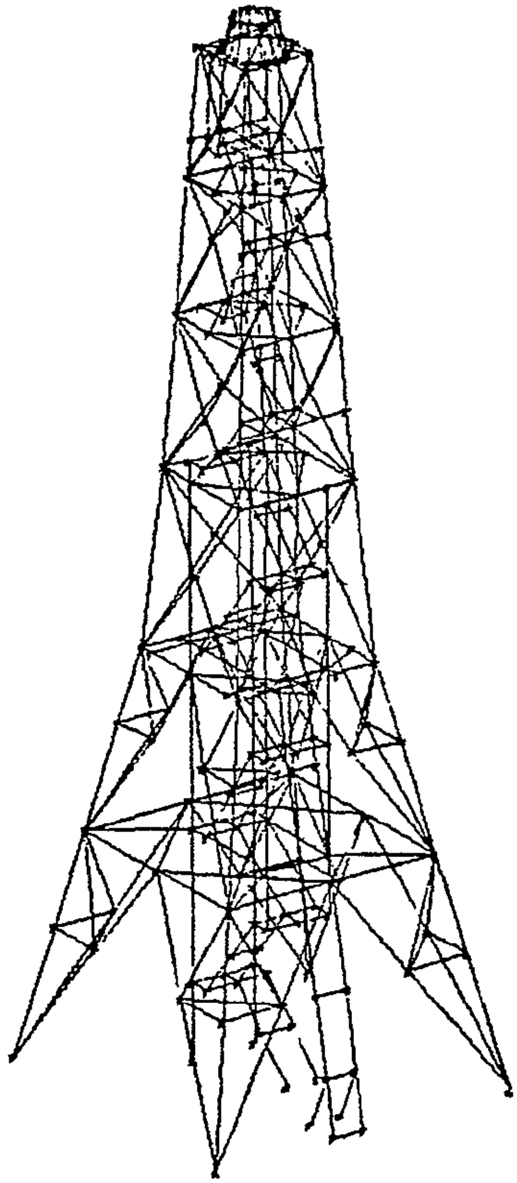


Figure 2.7-12. Mod-0 Tower Computer Model

Bedplate. The bedplate model was developed using NASA supplied drawings CR 757667 and CR 757669. There were 394 nodes (222 degrees of freedom) connected by 571 bar and plate elements representing this structure. For the eigenvalue problem, the nodes were reduced to 37 for inertia loading, most with 3 degrees of freedom. Figure 2.7-13 shows a computer plot of the complete bedplate analytical model. Weight distribution was based on the computer analysis of the structure. Component weights and adjustments were made according to NASA memo to agree with the total machine weight of 38,347 pounds within less than 1 percent. A cg check was also made which agreed within 3 percent of the memo notation of 4.8 feet toward the hub. Eigenvalues and eigenvectors of the bedplate were calculated with free attachment coordinates at the yaw bearing interface as the coupled solution input data.

Hub/Shaft. This substructure was modeled using straight beam elements both for the hub and the high and low speed shafts. This substructure was modeled using shaft and hub drawings supplied by NASA. Nodes were selected to correspond to the bearing attachment locations, coupling, gearbox, bearing supports and other mass loading points. Gearbox flexibility was included in the form of beam elements which considered gear ratio effects. Twenty-one nodes were used to represent this complete assembly from the hub to the generator. Weight distribution was made according to the NASA memo referenced previously and the stiffness was checked by calculating eigenvalues using a rigid mass and inertia representation of the rotor blades. Input for the coupled solution was obtained by calculating eigens without rotor blade mass and with free attachment coordinates. A free shaft model was used reflecting a soft coupling through the generator.

Rotor. Rotor blade mass, stiffness and airfoil properties were provided by NASA and were used as input data for the blade computer program, STRAP. Mode shapes and frequencies were calculated with free attachment coordinates. They were also calculated with the blade cantilevered from the lower bearing. Since test data were available for the rotor blades mounted to a simulated hub, this provided a convenient checkpoint for blade natural frequencies. Table 2.7-1 shows the frequencies comparison for in-plane and flapping modes differ by 6.4 to 7.5 percent. This first torsion mode differed only by 4.9 percent. As a result of this comparison, the free blade modes were used as input to the coupled system solution. Thirteen nodes with 6 degrees of freedom each were used to represent a single blade.

2.7.3.3 Stiffness Links

The stiffness links required to assemble the complete system were developed based on data supplied by NASA and from the bearing manufacturers. These data were used to form stiffness matrices which were used as the flexibility links in the modal synthesis which related each substructure at its attachment coordinates. Bearings were identified by the drawings which aided in obtaining the stiffness properties direct from the manufacturer. In addition to bearing stiffness, the blade to hub stiffness link also included the pitch actuator stiffness since the actuator mechanism was not modeled explicitly. This was accomplished by using the actuator stiffness as the rotational restraint about the blade axial direction. In a similar fashion, yaw drive system stiffness (obtained from test data) was included in

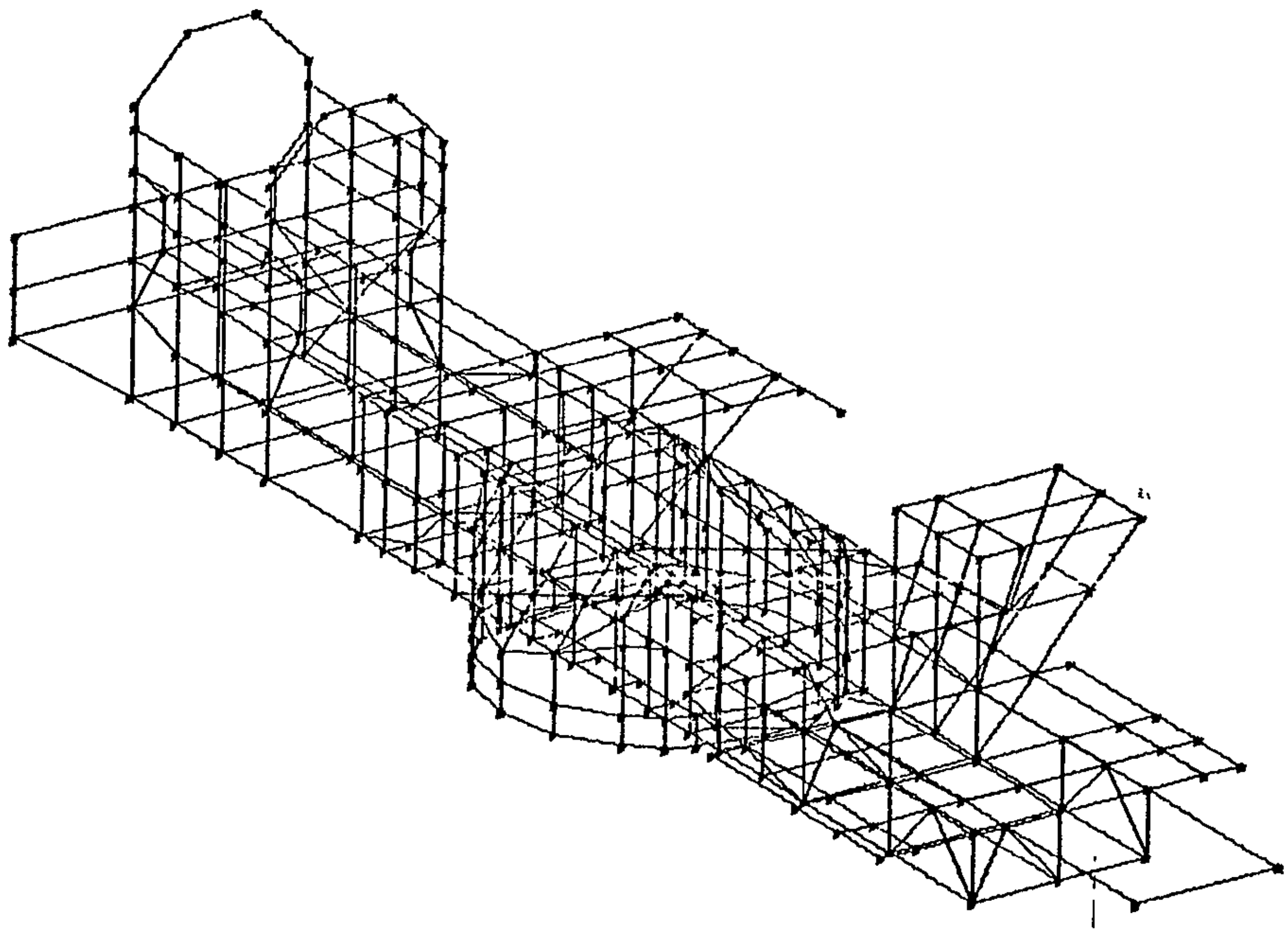


Figure 2.7-13. Mod-0 WTG Bedplate Computer Model

Table 2.7-1. Cantilevered Mod "0" Blade Frequencies

Mode No.	Analysis*	Test**	% Difference	Description
1	1.60	1.73	7.5	1st Flapping
2	2.49	2.66	6.4	1st In Plane
3	4.62	4.99	7.4	2nd Flapping
4	9.06	9.80	7.5	2nd In Plane
5	9.68	10.38	6.7	3rd Flapping
6	34.4	32.8	4.9	1st Torsion

*Cantilevered at lower bearing.
**Mounted to simulated hub.

the yaw bearing stiffness matrix as rotational restraint about the yaw axis. Separate stiffness links were developed for the blade to hub, hub/shaft to bedplate, and bedplate to tower interfaces.

2.7.3.4 Coupled Dynamic Model

The Mod 0 structure was assembled with the same procedures and methods described for Mod 1. The stiffness coupling methods in the SCAMP computer program were used considering the stiffness links between two substructures at a time:

1. Tower to bedplate
2. Bedplate to hub/shaft
3. Hub/shaft to blade No. 1
4. Hub/shaft to blade No. 2

The orientation of each substructure was such that the total assembly configuration matched that of the modal test. This was done to enable a direct comparison with modal test data and thus, validate the substructure models and the assembly technique.

A plot of the complete dynamic model for the modal test configuration is given in Figure 2.7-14. A total of 468 degrees of freedom for all substructures was used to describe the dynamic behavior of the system. For the coupled system solution, the final size of the eigenvalue solution was reduced by the dynamic transformation in the SCAMP program, as shown in the summary of Table 2.7-2. The column heading "Eignevalue Size" size" gives the substructure eigens going into SCAMP. Of the 468 available modes; a total of 333 modes were used to synthesize the coupled model. The final size of the eigenvalue problem ([illegible]1 DOF) was determined from the total of modes "kept". The distribution of kept and reduced modes is also shown in the table.

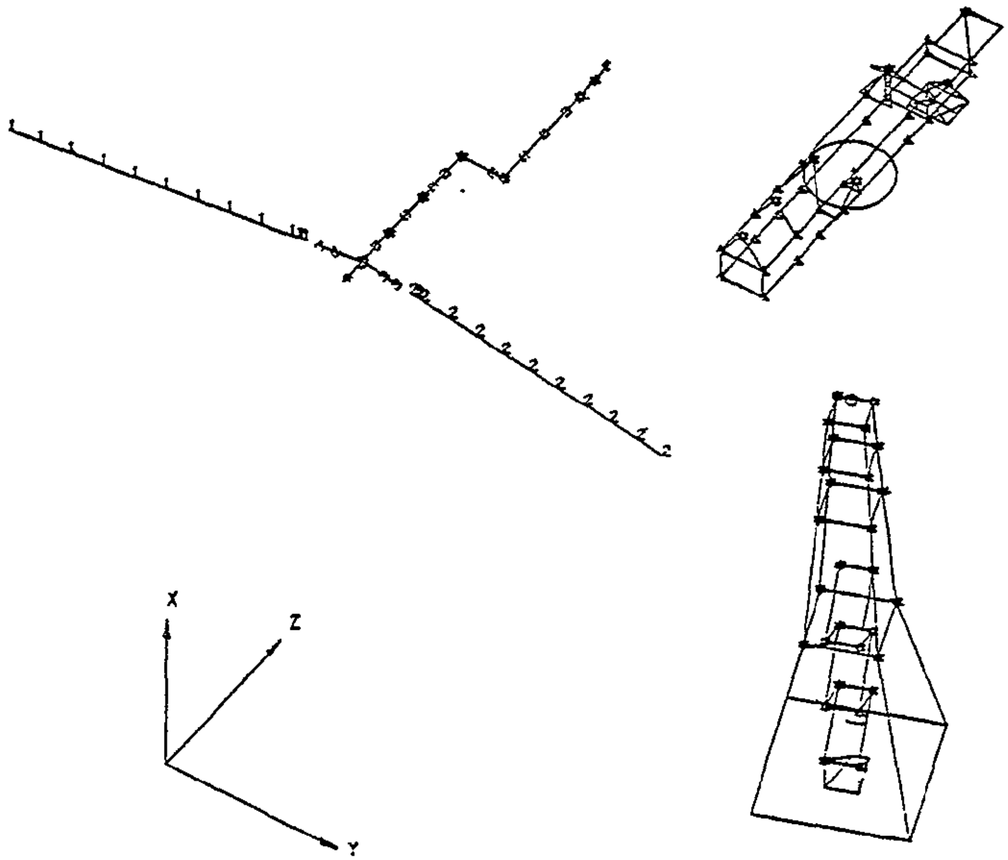


Figure 2.7-14. Mod-0 Model Test Configuration

Table 2.7-2. Substructure DOF Table

Substructure	Joints	DOF	Eigenvalue Size	Modes Kept	Modes Reduced	Total Modes Used
Tower	264	1512	90	18	36	54
Bedplate	394	2222	114	14	28	42
Hub/Shaft	21	108	108	27	54	81
Blade No. 1	13	78	78	26	52	78
Blade No. 2	13	78	78	26	52	78
Totals	705	3998	468	111	222	333

The final solution resulted in 20 modes under 10 Hz. Plots of the eigenvalues and eigenvectors are given in Figures 2.7-15 through 2.7-19 for the fundamental structural modes. These modes were used to compare directly with the results reported in NASA TMX-71879, 3426, and the final report by the University of Cincinnati on the modal testing.

2.7.3.5 Modal Test Comparison

A direct comparison of analytical and test results was possible for four tower modes and three blade modes. Although other frequencies were measured on test, modal identification was lacking and correlation on a frequency basis alone was not attempted. Excellent agreement with test, however, was obtained for those comparable modes as shown by the listing in Table 2.7-3. All of the first 22 modes noted in the tabulation were clearly identified for the structure as high as second bending in the tower and as high as third order blade modes. Four stairway modes appeared in this set of 22 modes which were distinct but not coupled significantly to any blade response.

This comparison showed agreement in modal frequency within less than 1 percent for the first tower bending (N-S), first tower torsion and second tower bending (N-S). The first rotor flatwise and edgewise modes differed by only 3 to 4 percent and the others by not greater than 9 percent. A check of modal displacements also showed good agreement for the tower bending modes (Modes 4 and 5). The upper bay motion plotted in Figure 2.7-20 shows similar diagonal motion as that measured on the test which was obtained along with similar amount of displacement. This diagonal motion was not observed on identical

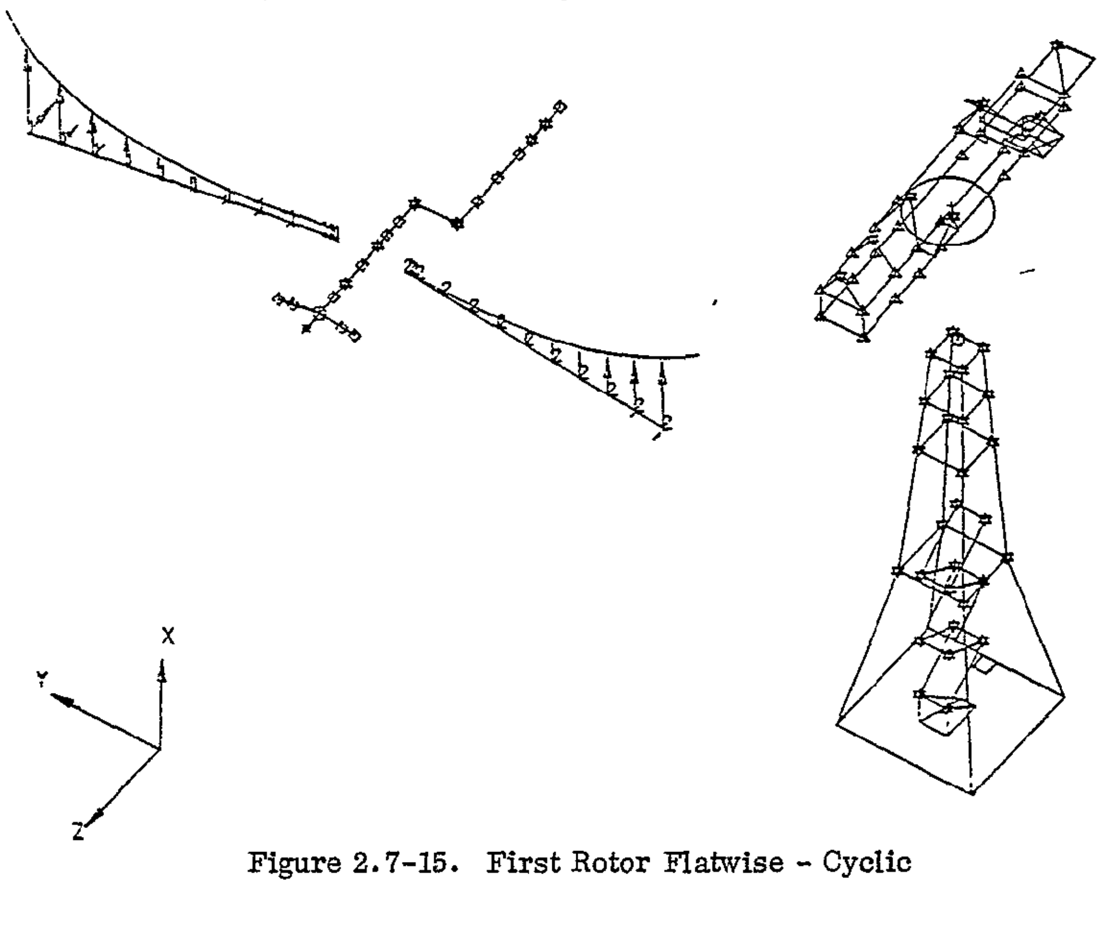


Figure 2.7-15. First Rotor Flatwise - Cyclic

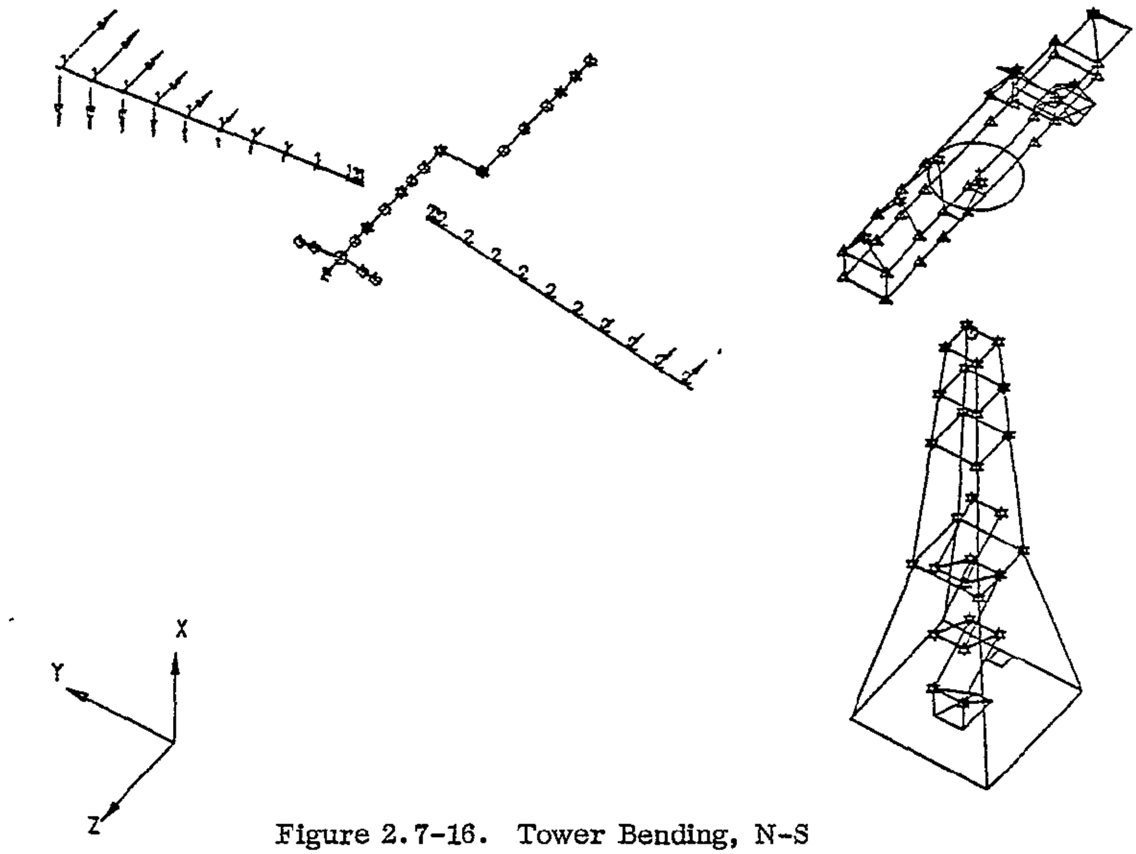


Figure 2.7-16. Tower Bending, N-S

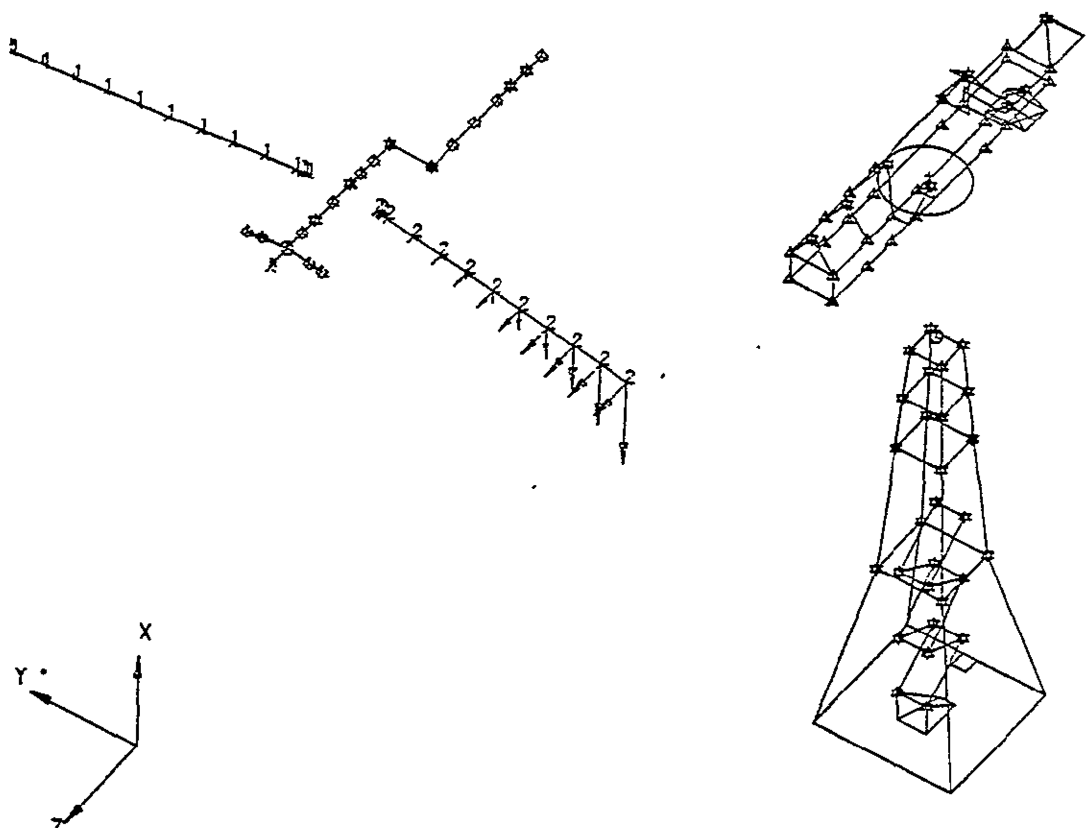


Figure 2.7-17. Tower Bending, E-W

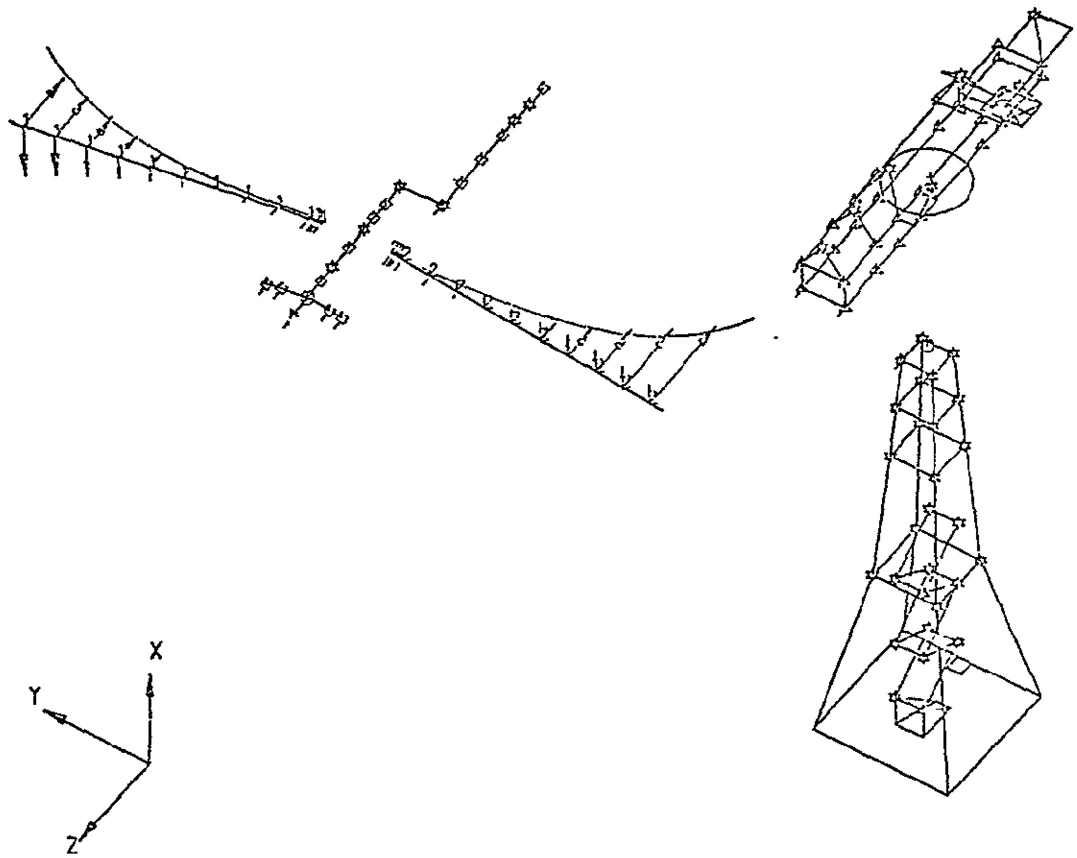


Figure 2.7-18. First Rotor Edgewise - Collective

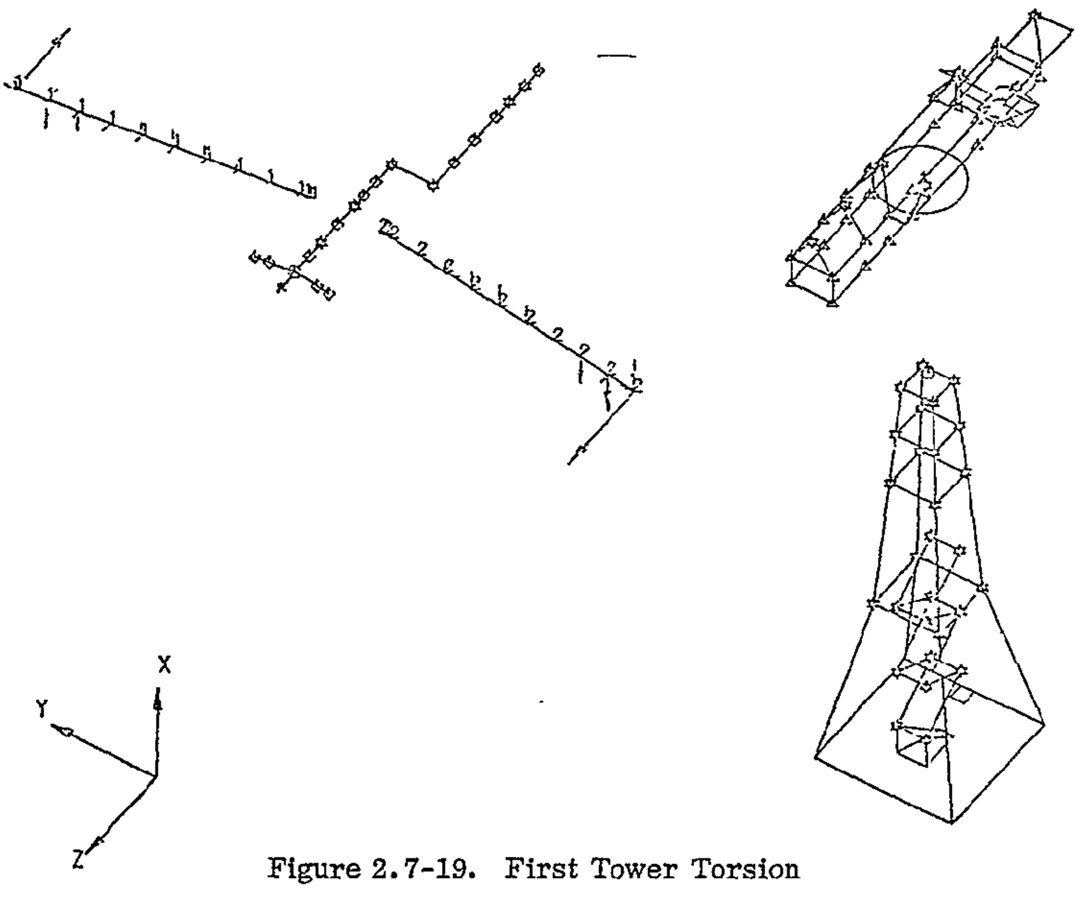


Figure 2.7-19. First Tower Torsion

Table 2.7-3. MOD-0 Modal Test Comparison

Mode No.	Descriptive	GE Analysis (Freq = Hz)	Modal Test (Freq = Hz)
1	Bedplate Rotation	1.07	
2	Shaft Torsion	1.53	
3	First Rotor Flatwise - Cyclic	1.78	1.73
4	Tower Bending, N-S	2.09	2.1
5	Tower Bending, E-W	2.40	2.2
6	1st Rotor Edgewise, Collective	2.63	3.0
7	2nd Rotor Flatwise, Collective	3.54	
8	Tower Stairs, Lat., N-S	3.78	
9	2nd Rotor Flatwise, Cyclic	3.84	
10	2nd Rotor Edgewise, Cyclic	4.49	
11	2nd Rotor Edgewise, Cyclic	4.66	
12	2nd Tower Stairs, N-S	5.04	
13	Lower Tower Stairs Twist	6.47	
14	Upper Tower Stairs Lat	6.67	
15	3rd Rotor Flatwise, Cyclic	6.84	
16	3rd Rotor Edgewise, Cyclic	8.37	
17	3rd Rotor Flatwise, Collective	8.90	
18	Rotor Torsion	8.99	
19	3rd Rotor Edgewise, Collective	9.72	
20	1st Tower Torsion	9.76	9.8
21	2nd Tower Bending (with torsion N-S)	10.38	10.4
22	2nd Tower Bending, E-W	10.46	

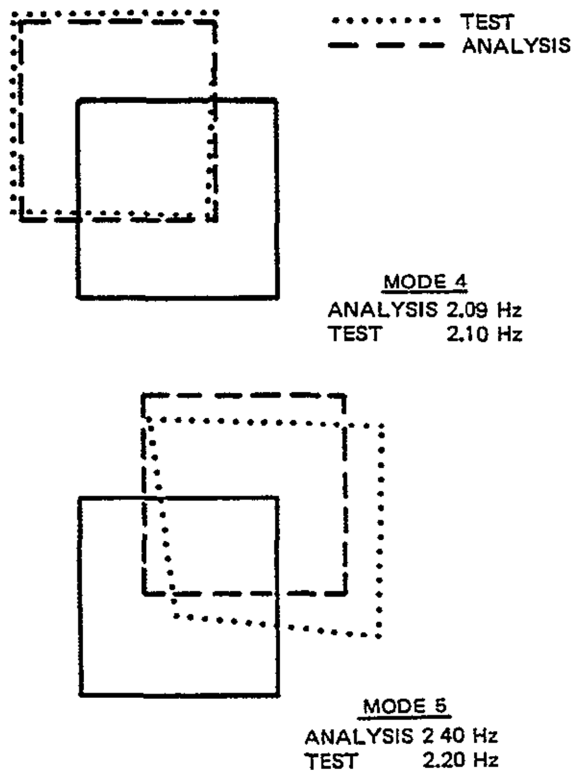


Figure 2.7-20. Tower Bending Deflection Comparison

modal solutions where the shaft was unrestrained. It also did not appear on the tower alone modal plots which points to a blade/tower coupling rather than the suspected influence of the stairway structure.

A review of several checkpoints in the model buildup and synthesis provided a substantial basis to confirm the model adequacy:

Weight	—	Within 1 percent on major substructures. CG of bedplate within 3%
Frequency	—	Cantilevered blade within 7.5% Three coupled system modes within 1% Two coupled system modes within 3-4% Two coupled system modes within 6-9%
Modes	—	Modes clearly identifiable Diagonal tower motion reproduced Tower fundamental mode displacements similar

Based on the confirmation of the model by the test data, the configuration was then considered ready to be modified for the test conditions under study.

2.7.4 DYNAMIC MODEL CONFIGURATIONS

The Mod 0 analytical model for the modal test served as a baseline model for the test cases under study. With this configuration established, the shaft was released and the blade and bedplate were reoriented for each test condition.

A summary of the test conditions proposed for study is given in Table 2.7-4, along with the modal test. (These conditions are discussed further in Section 2.7.5). Operating condition, blade and nacelle configurations were supplied as a part of the strip chart, digital printout and tabulated data supplied by NASA. Flex coupling and yaw bearing stiffness data were also supplied by NASA but not as a part of the test measurement package. It was necessary to use these data analysis to "tune" the shaft to the desired 4P shaft torsion frequency for the Case 3 configuration and 6P for the other cases. This was done by adjusting the flex coupling stiffness until the resonant shaft condition which occurred during the "torque bloom" condition was achieved. The yaw drive stiffness value was obtained from the nacelle load deflection data provided by NASA. The basis of selection was the range of tower deflection shown by the strip charts. The effective generator inertia was increased to reflect the pulley ratio change for Case 2.

In order to calculate responses and loads it was necessary to develop a seaprate system dynamic model for each case because no two nacelle positions and corresponding blade angles, drive and shaft configurations matched another case. In addition, within a particular case configuration, a separate system model with rotor positions at 0° (12 o'clock), 45°, 90°, and 135° were also required for the piecewise linear response analysis, thus requiring a total of 16 coupled system models for the four cases.

A summary of mode descriptions and frequencies for Case I is given in Table 2.7-5. This summary is for the rotor blade in the horizontal position and is given as a comparison point for the test model. Modal plots for these modes are given in Figures 2.7-21 through 2.7-25. From this comparison, some similarities with the modal test model were evident, especially in the tower modes. Although mode position shifted on the blade modes, frequencies remained essentially unchanged for tower first and second bending, tower torsion and the four stair modes. Bedplate rotation mode changed by only about 10 percent but blade modes shifted considerably in mode position and character. This was expected since the blade was not feathered and centrifugal stiffening is present due to blade rotation at 40 rpm.

The other rotor positions were calculated and plotted but are not given here. It was possible to track the change in some modal frequencies with azimuth change such as bedplate rotation and tower bending (both directions). The fundamental and second flapwise mode, however, was noted to change very little with azimuth. Changes in the edgewise modal frequency with azimuth were not as easy to trace as the rotor azimuth changed. This was due to the coupling of this mode with some of the tower modes. Stair modes maintained their modal frequencies.

The modal data for the four rotor azimuth positions for each case were computed in a similar fashion. The correlation of the feather condition results with the analytical model for the modal test gave the required confidence that the modes and resonant frequencies could be used as input to the forced response problem.

Table 2.7-4. MOD-0 Load Cases

	Load Cases				
	Test	Case 1	Case 2	Case 3	Case 4
Operating Conditions					
RPM	0	40	20	41	40
KW	0	100	37	27.8	98
Wind Velocity Vw	0	28	2-35 mph	20.5	25
Inflow Angle On	0	4°	10°	-5°	12°
Gust	0	—	See V_W	—	—
Blade					
Airfoil Pitch	-90°	-8°	-20°	-8.5°	-5.5°
Powered/Feathered	Feathered	Powered	Powered	Powered	Powered
Model No.	M0B05L				
Shaft					
Flex Coupler K (in-lb/rad)	$1.1X10^8$	$1.1X10^8$	$1.1X10^8$	$0.6X10^7$	$1.1X10^8$
/rev	6p	6p	6p	4p	6p
Freq (Hz)	3.86	3.86	3.86	2.95	3.86
Model No.	SFT002	SFT002	SFT002	SFT003	SFT002
Nacelle					
Angle On	235°	262°	264°	224°	256.2°
Tower/Bedplate	0°	-27°	-29°	+11°	-2.12°
Yaw Bearing					
Stiffness K (in-lb/rad)	1.65E8	1.65E8	1.65E8	1.65E8	1.65E9
Spring No.	WTG003	WTG003	1.65E8	WTG003	WTG002
Stairs		V	—	—	—

Table 2.7-5. MOD-0 Modes and Frequencies - Case 1
(90° Position)

MODE NO.	DESCRIPTION	FREQUENCY (Hz)	FREQUENCY (1/REV)
1	ROTOR ROTATION	0.	0.
2	BEDPLATE ROTATION	1.19	1.79
3	1ST ROTOR FLATWISE - COLLECTIVE	1.77	2.65
4	TOWER BENDING, N-S	2.07	3.10
5	1ST ROTOR EDGEWISE - CYCLIC	2.39	3.59
6	TOWER BENDING, E-W	2.52	3.78
7	ROTOR BLADE TORSION	3.25	4.88
8	1ST ROTOR FLATWISE - CYCLIC	3.41	5.12
9	TOWER STAIRS, LATERAL, N-S	3.78	5.67
10	2ND ROTOR FLATWISE - COLLECTIVE	4.86	7.29
11	TOWER STAIRS TWIST	5.03	7.54
12	2ND ROTOR FLATWISE - CYCLIC	5.13	7.70
13	TOWER STAIRS - DIAGONAL	6.46	9.69
14	2ND ROTOR EDGEWISE - CYCLIC	6.50	9.75
15	TOWER STAIRS - UPPER	6.69	10.35
16	2ND ROTOR EDGEWISE - COLLECTIVE	6.86	10.29
17	3RD ROTOR FLATWISE - CYCLIC	8.95	13.43
18	3RD ROTOR FLATWISE - COLLECTIVE	9.53	14.30
19	2ND ROTOR TORSION	9.66	14.49
20	1ST TOWER TORSION	9.76	14.64
21	2ND TOWER BENDING, N-S	10.14	15.21
22	2ND TOWER BENDING, E-W	10.58	15.87
23	TOWER STAIR TWIST	12.91	19.37
24	4TH ROTOR FLATWISE - CYCLIC	14.59	21.89
25	4TH ROTOR FLATWISE - COLLECTIVE	15.48	23.22

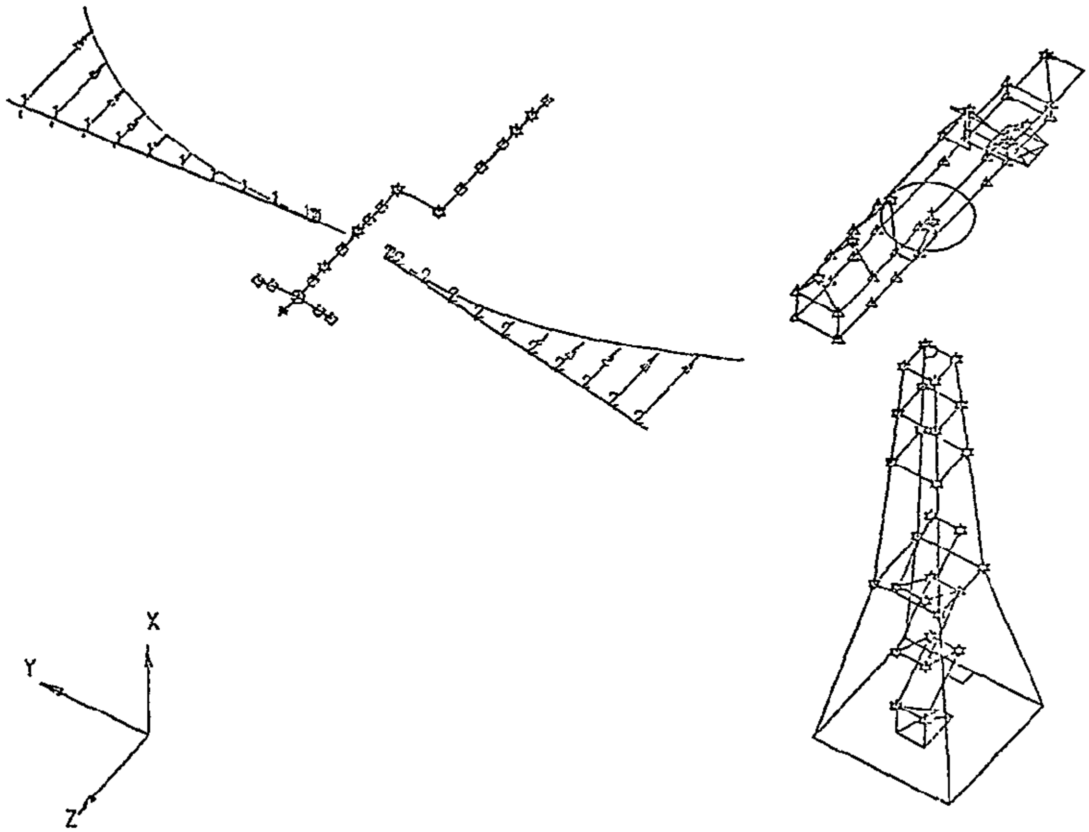


Figure 2.7-21. First Rotor Flatwise - Collective

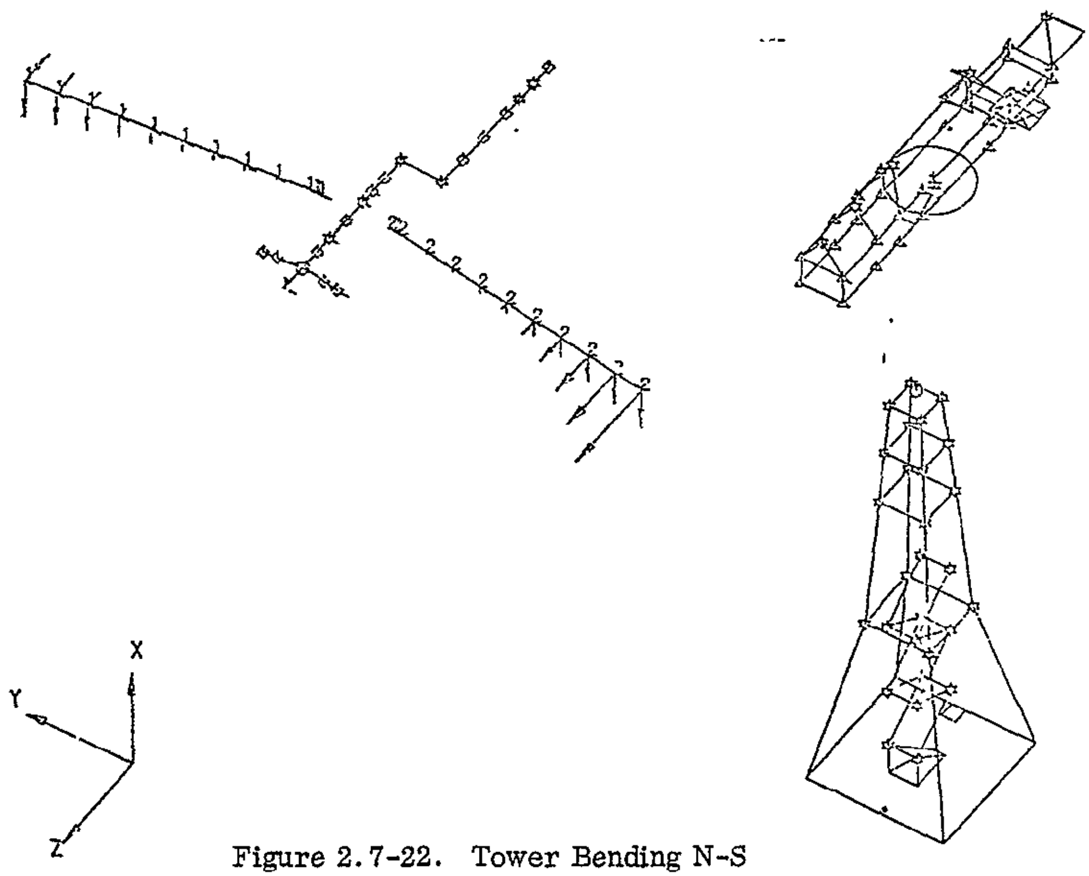


Figure 2.7-22. Tower Bending N-S

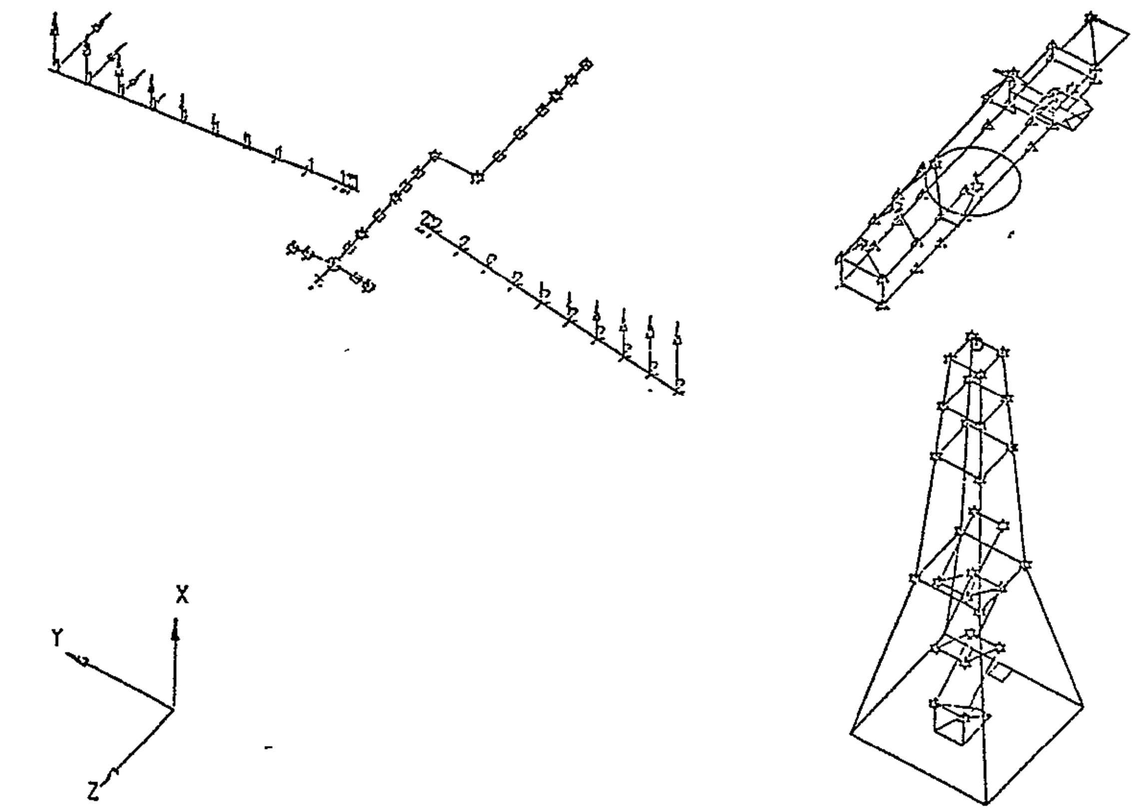


Figure 2.7-23. First Edgewise - Cyclic

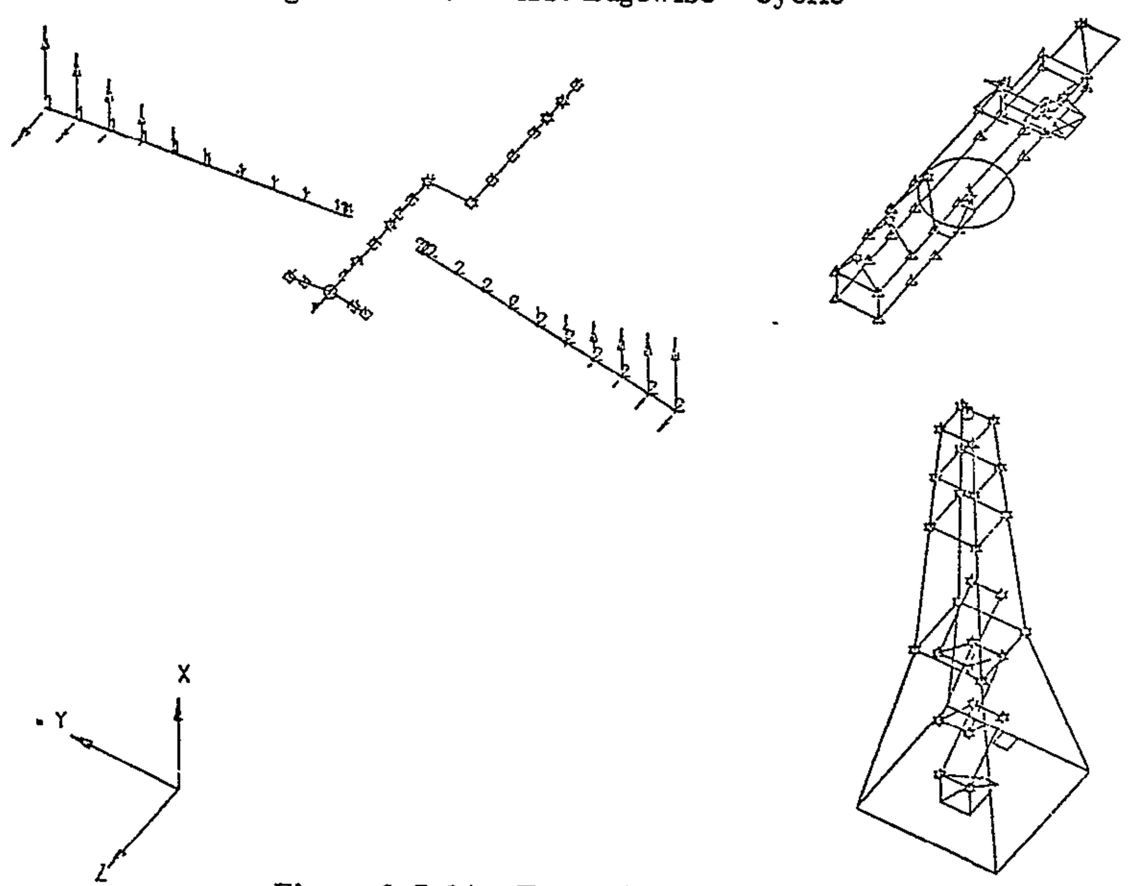


Figure 2.7-24. Tower Bending, E-W

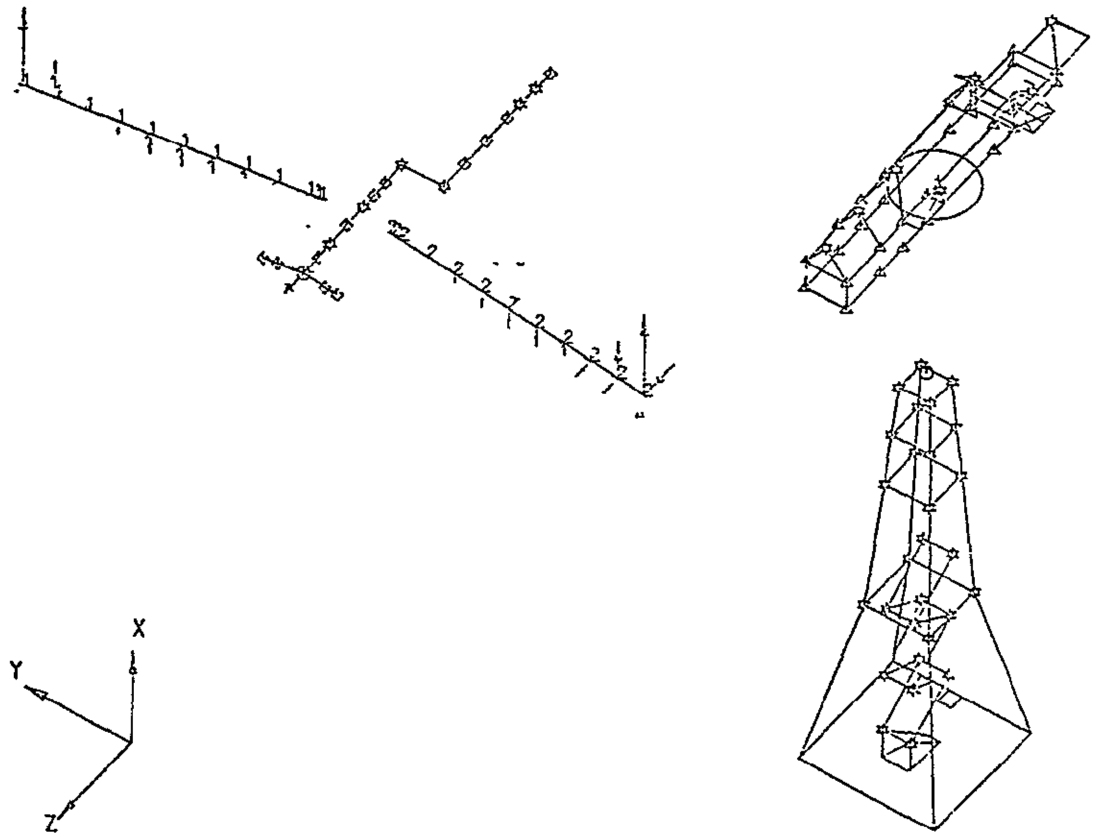


Figure 2.7-25. Tower Torsion

2.7.5 CODE VERIFICATION RESULTS

The code verification task has been completed for Case 1 and 4. Selected results are included to indicate the correlation with measured results. One run was made for Case 1 and two runs have been made for Case 4 although the changes in the Case 4 run are insignificant. Excellent correlations were obtained with regard to peak and cyclic loading, bedplate accelerations and tower deflections including comparable harmonic content.

The operating conditions for each case were obtained by adjusting the wind velocity to obtain the desired rotor power. The blade position was set based on the measured blade angle. The wind direction was also set at the value specified by NASA. Using the measured shaft torque and rotor speed to establish the rotor power, the wind velocity was varied until the desired rotor power was obtained. This established the aerodynamic and inertial forces to excite the Mod-0 system.

The first 20 elastic modes of the system were used at each of eight rotor positions with aerodynamic damping coefficients obtained from the fully coupled mode shapes. The resonant frequencies and modal damping coefficients for the first ten modes at each rotor position are given in Table 2.7-6A and 6B. Examination of the table indicates significant differences in the system dynamic characteristics for the models of the two case. Case 1 includes a flexible yaw drive system which has a major contribution to the first three modes and, although the damping is relatively high, reduces the modal damping relative to Case 4.

Table 2.7-6A & 6B. Summary of Resonant Frequencies and Modal Dumping Coefficients for the First Ten Modes

CASE 1								
	0°		45°		90°		135°	
MODE NO.	f(1/REV)	C/Cc	f(1/REV)	C/Cc	f(1/REV)	C/Cc	f(1/REV)	C/Cc
1	2.30	.2217	1.96	.3271	1.78	.3953	1.91	.3599
2	2.45	.2529	2.64	.3233	2.66	.3632	2.66	.3758
3	2.67	.3818	2.77	.1885	3.10	.1284	3.03	.1376
4	3.63	.0774	3.69	.0806	3.59	.0825	3.51	.0739
5	4.11	.0539	3.93	.0554	3.78	.0622	3.96	.0633
6	4.86	.0518	4.90	.0566	4.87	.0700	4.82	.0783
7	5.08	.0508	5.21	.1127	5.12	.1273	5.04	.0808
8	5.66	.0500	5.66	.0500	5.66	.0500	5.66	.0500
9	6.42	.1873	6.95	.1378	7.29	.1927	7.10	.1569
10	7.29	.1866	7.30	.1862	7.54	.0562	7.30	.1734

CASE 4								
	0°		45°		90°		135°	
MODE NO.	f(1/REV)	C/Cc	f(1/REV)	C/Cc	f(1/REV)	C/Cc	f(1/REV)	C/Cc
1	2.36	.3930	2.31	.4046	2.24	.4177	2.28	.4073
2	2.66	.4008	2.65	.3862	2.65	.3789	2.66	.3870
3	3.02	.0565	3.04	.0587	3.24	.0730	3.20	.0685
4	3.67	.0754	3.74	.0759	3.58	.0715	3.55	.0708
5	4.11	.0534	3.93	.0535	3.86	.0602	4.02	.0562
6	4.93	.0514	4.92	.0508	4.92	.0519	4.92	.0525
7	5.66	.0500	5.66	.0503	5.66	.0510	5.66	.0504
8	6.40	.1914	6.16	.1815	5.97	.1749	6.08	.1695
9	7.32	.1864	7.30	.1874	7.30	.1920	7.30	.1917
10	7.38	.0528	7.54	.0513	7.55	.0507	7.54	.0517

The damping of the higher modes is due largely to structural damping although significant aerodynamic damping is evident in some of the modes, e.g., Mode 9. It will be noted that a structural damping coefficient of 0.05 was added to the aerodynamic damping coefficients. This is typical of the damping of spacecraft structures excited at relatively large amplitudes and was considered representative for Case 1 in view of the yaw drive motions that occur. The structural damping coefficient of 0.05 was also used for Case 4 although it may be excessive because the yaw drive free play is eliminated in this case.

Initial comparisons of the time histories of the blade root bending moments (Station 40) are shown in Figures 2.7-26 and 2.7-27 for Case 1 and 4 respectively. In general, the major peaks of the load time history are preserved in the analytical predictions with the proper phasing. The harmonic analysis of the blade root moments is shown in Figure 2.7-28 and 29, and also compare favorably. The Case 1 comparisons are not as good as Case 4 but, in view of the linearized modeling of the yaw drive, this would be expected. The analysis predictions tend to be conservative for flapwise bending but are in closer agreement with measured loads for chordwise bending.

The comparison of the analytical and measured loads is summarized in Table 2.7-7. The peak value is compared with the range of peak values over approximately three cycles and a similar comparison is made of the cyclic components (half the difference between the maximum and minimum loads). A harmonic content and waveform rating is also shown based on a criteria of matching or exceeding the major harmonics components: Excellent (80 to 100%), Good (50 to 80%), Fair (20 to 50%), Poor (less than 20%). The table indicates that over 80 percent of the calculated loads were within 20 percent of the measured values and that the harmonic content was good to excellent for most comparisons. For two loads, Lockheed REXOR calculated values are also shown. For the main drive shaft "pitch" moment (M_{YY}), the measured values appear to be in error while the calculated values compare favorably with REXOR calculations. The main drive shaft M_{XX} values appear high as indicated by the REXOR calculations. In general, the waveforms compared favorably although there appears to be a significant amount of 1P loading which may be due to blade inertial or aerodynamic loading or variations in the wind conditions.

The tower accelerations and deflections are compared in Table 2.7-8. The calculated values are generally much larger than the measured values although the waveform comparisons indicated comparable response frequencies as high as 10P (bearing "B" vertical acceleration). A portion of the difference is due to a large 2P component of response in the calculated transverse accelerations which may result from the linearization of the yaw drive system for Case 1. Although the calculated responses tend to be conservative, the code provides an excellent tool for the early identification of undesirable dynamic couplings throughout the system.

To aid in understanding the dynamic loading of the system, the mode providing the largest contribution to the load at each interface is tabulated. The modal load table for Case 1 and 4 are given in Tables 2.7-9 and 2.7-10. At the blade root (Interface 4) for Case 1, the table shows that the maximum negative flapwise moment (MY) occurs at a rotor position of 216 degrees with the largest contribution from Mode 3 (Model positioned at 45 degrees). This

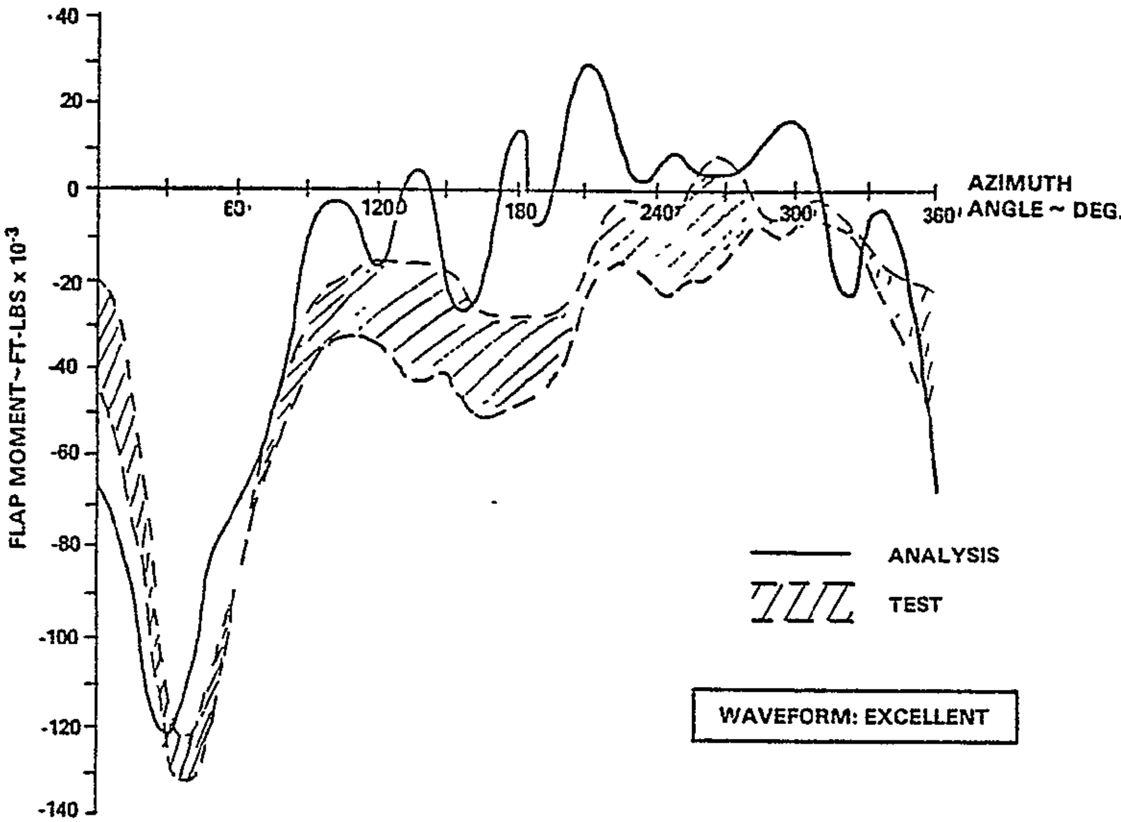


Figure 2.7-26(a). Comparison of Case 1 Blade Root Flapwise Moment

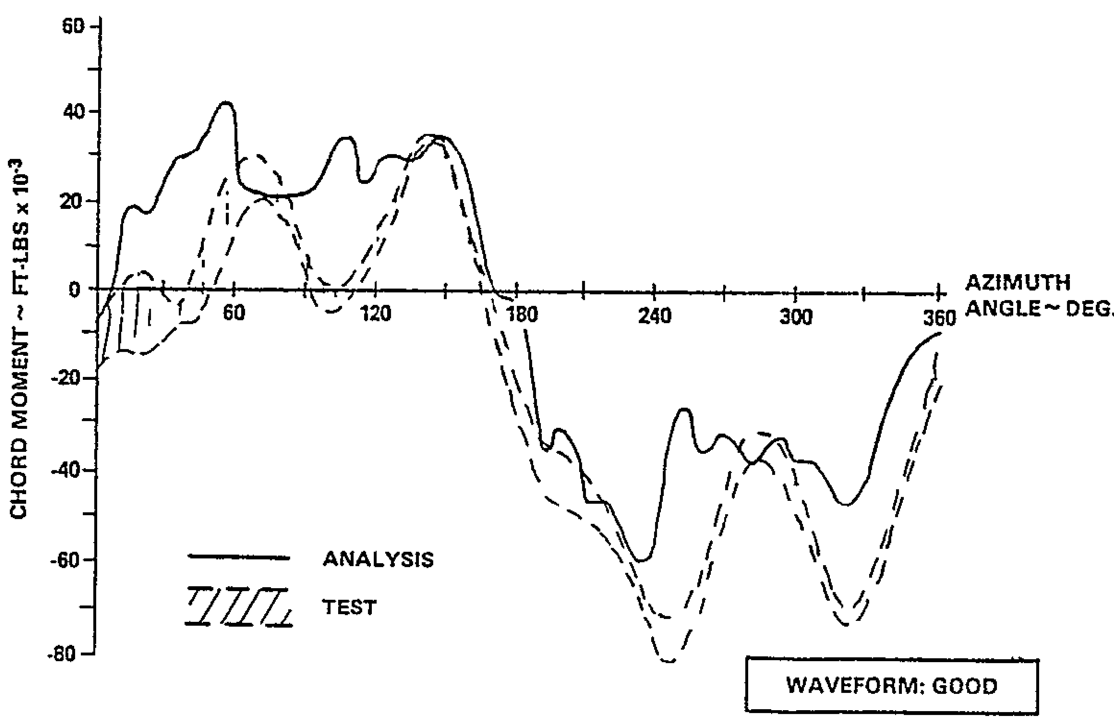


Figure 2.7-26(b). Comparison of Case 1 Blade Edgewise Moment

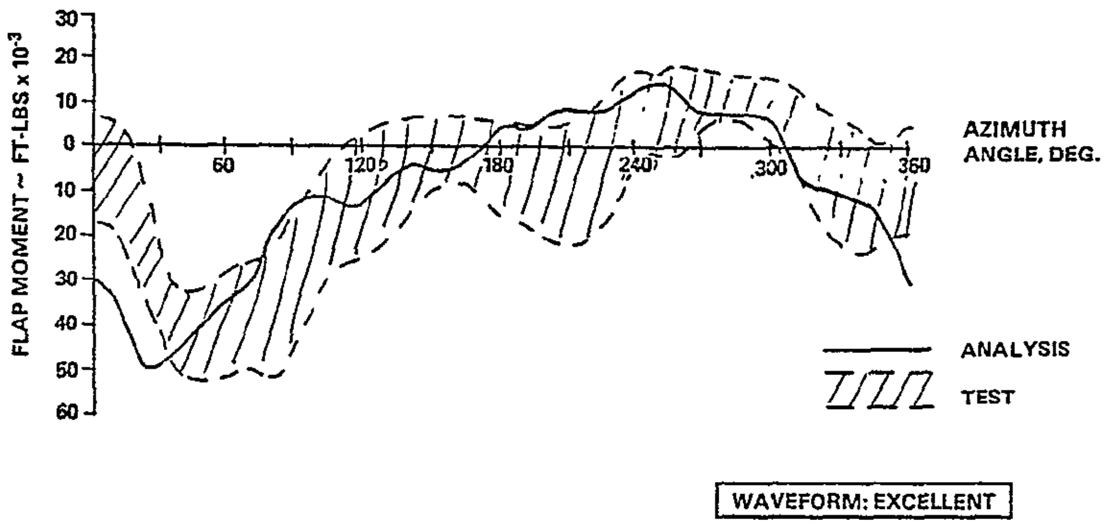


Figure 2.7-27(a). Comparison of Case 4 Blade Root Flapwise Moment

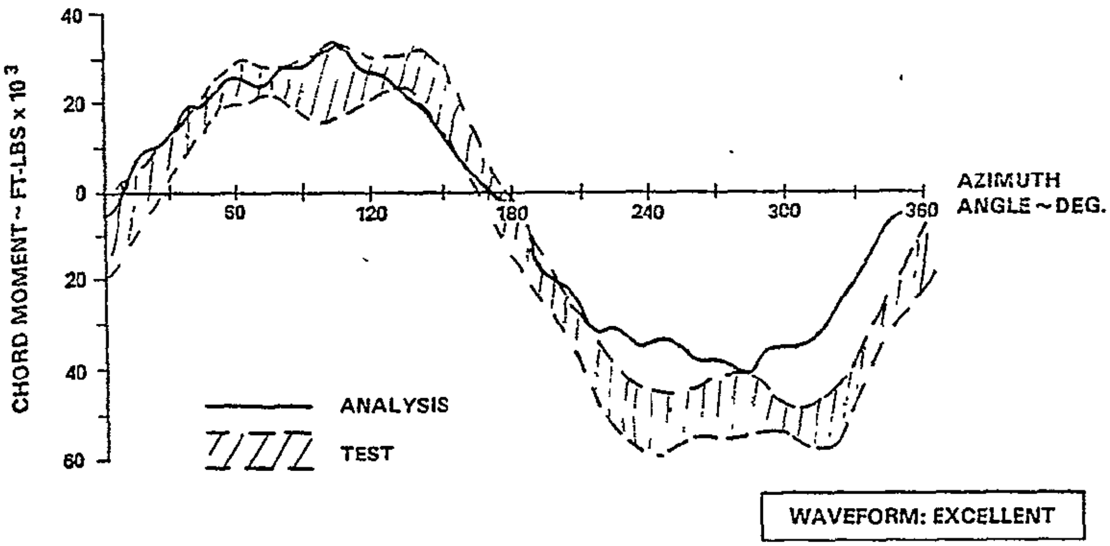


Figure 2.7-27(b). Comparison of Case 4 Blade Root Edgewise Moment

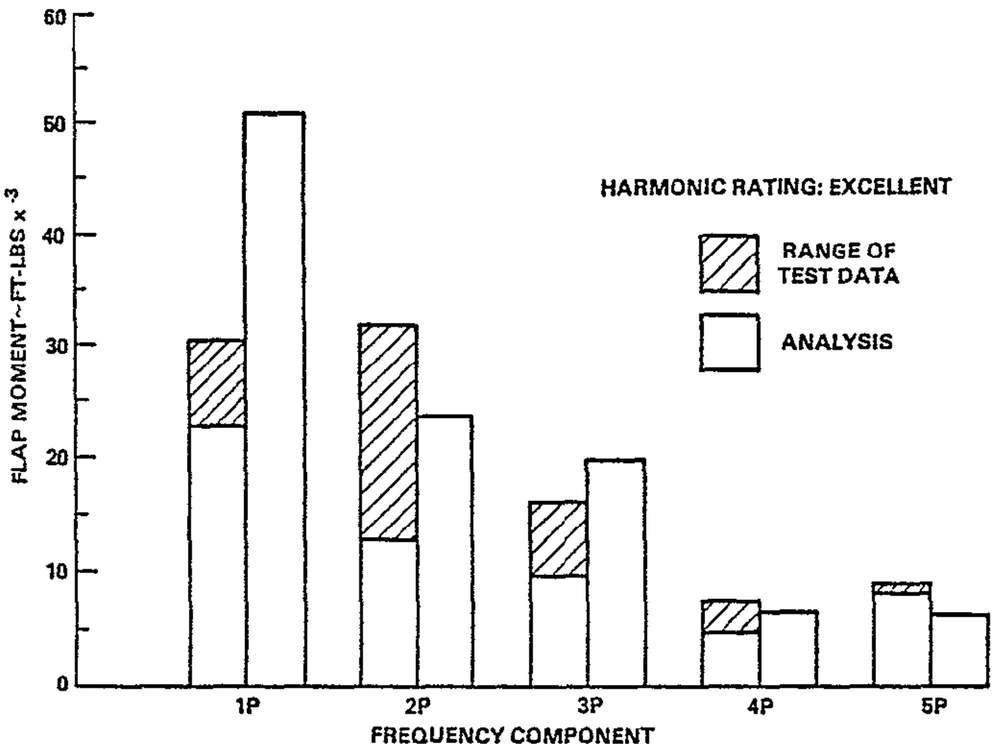


Figure 2.7-28(a). Harmonic Components, Case 1 Blade Root Edgewise Moment

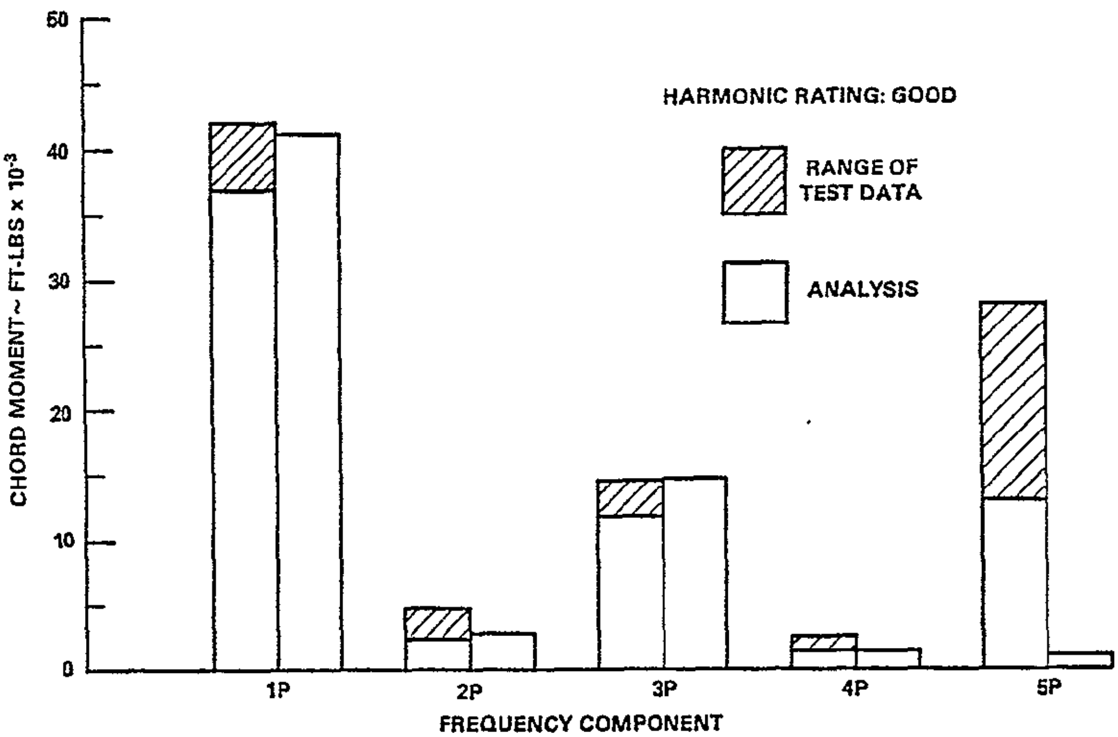


Figure 2.7-28(b). Harmonic Components, Case 1 Blade Root Edgewise Moment

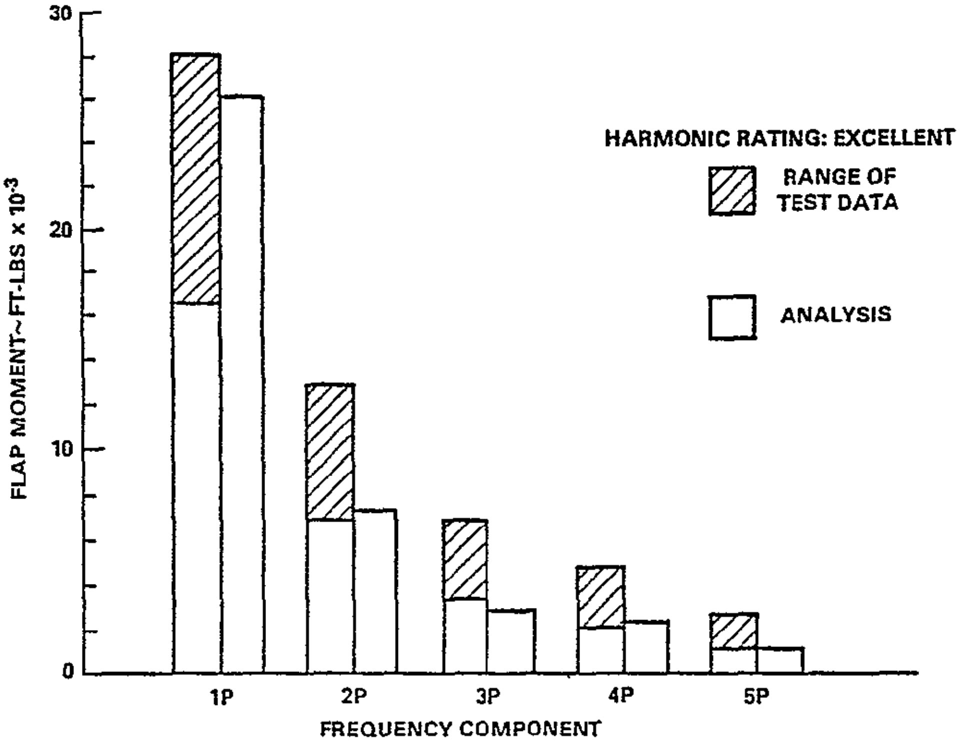


Figure 2.7-29(a). Harmonic Components, Case 4 Blade Root Flapwise Moment

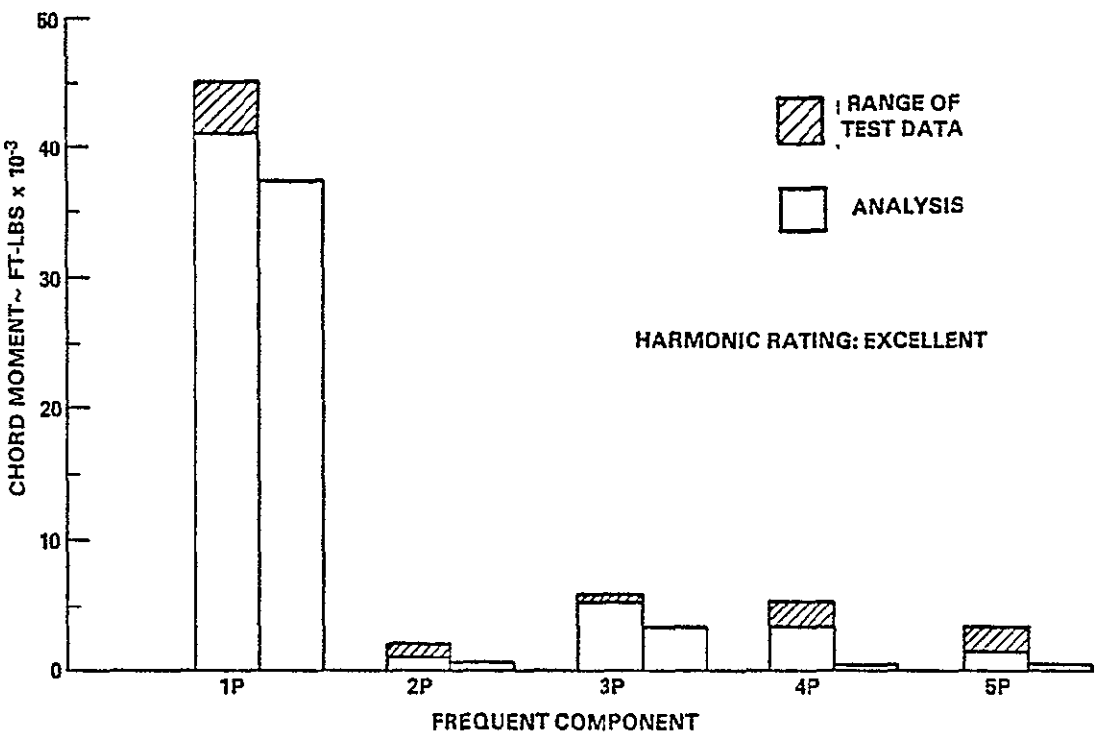


Figure 2.7-29(b). Harmonic Components, Case 4 Blade Root Edgewise Moment

Table 2.7-7. GETSS Load Comparison Summary (Ft - Lbs x 10^{-3})

MOMENT	CASE	PEAK MEASURED	PEAK ANAL.	PEAK %	CYCLIC MEASURED	CYCLIC ANAL.	CYCLIC %	RATING HARMONICS	RATING WAVEFORM
FLAP AT STATION 40	1	123-132	122	-1	63-70	76	+9	E	E
	4	33-53	50.3	0	24-36.5	32.2	0	E	E
FLAP AT STATION 370	1	28-32	29.5	0	16.8-21.3	22	+3.3	-	G
	4	5.5-13	10.2	0	6 5-12.5	8.8	0	E	E
CHORD AT STATION 40	1	72-82	58	-19	53.5-58.5	51	-5	G	G
	4	52-59	42	-19	40.5-45.5	39	-4	E	E
CHORD AT STATION 370	1	17-22	21	0	14-19.8	17	0	-	G
	4	7.5-8	12.5	+56	6.5-8	8.2	+2.5	G	G
DRIVE SHAFT, M_{XX}	1	55-56 (21)	38	-31	47.5-48 (21)	39	-18	-	E
	4	32-34 (?)	17	-47	31-33 (?)	17.5	-44	P	G
DRIVE SHAFT, M_{YY}	1	141-145 (122)	128	(+5)	70 5-72.5 (122)	128	(+5)	-	G
	4	66-83	66	0	61-75.5	65	0	E	E
DRIVE SHAFT, M_{ZZ}	1	4.6-6.8	7.0	+3	5.0-7.0	11.5	+65	-	F
	4	2.3-4.2	4.2	0	1.65-3.95	2.9	0	E	-

( ) REXOR PREDICTIONS

- 82% OF PREDICTED VALUES ARE WITHIN 20% OF MEASURED VALUES
- HARMONIC CONTENT: GOOD TO EXCELLENT
- WAVEFORM REPRODUCED

Table 2.7-8. Acceleration and Deflection Summary

BEARING "B" DIRECTION	CASE	MEASURED ACCEL. (±g)	ANAL.	%	WAVEFORM RATING	HARMONIC RATING
VERTICAL	1	.25 - .4	.87	+120	E	–
	4	.018 - 07	.10	+ 43	E	G
TRANSVERSE	1	.14 - .2	.28	+ 40	G	–
	4	.044 - .058	.087	+ 50	G	E
THRUST	1	.07 - .11	.18	+ 64	G	–
	4	.014 - .055	.031	0	G	E

TOWER DIRECTION	CASE	PEAK (INCHES)			CYCLIC (± INCH)			WAVEFORM RATING
		MEASURED	ANAL.*	%	MEASURED	ANAL.*	%	
X	1	.32 - .39	.34	0	.21 - .28	.34	+ 21	G
Y	1	(.22)	.36	+73	(.12)	.25	+100	–

* PREDICTION BASED ON TOWER CORNER
( ) REXOR PREDICTION

- BEARING ACCELERATIONS AND TOWER DEFLECTIONS ARE CONSERVATIVE
- CAPTURES 10P RESPONSE
- LINEARIZED YAW DRIVE APPEARS TO CAUSE HIGH 2P RESPONSE

Table 2.7-9. Summary of Largest Modal Contributions, Case 1

MODAL LOAD		MAXIMUM	POSN	MODE	MINIMUM	POSN	MODE
3/4 RADIUS BLADE LOCAL LOADS	VX	160.6	159.0	14.00	-165.6	339.0	14.00
	VY	340.4	126.0	13.00	-404.7	36.00	3.000
	VZ	956.2	216.0	3.000	-450.6	21.00	1.000
	MX	113.7	216.0	3.000	-60.56	36.00	3.000
	MY	2273.	339.0	14.00	-3902.	216.0	3.000
	MZ	2633.	126.0	13.00	-2033.	339.0	15.00
1/2 RADIUS BLADE LOCAL LOADS	VX	456.6	159.0	14.00	-467.6	339.0	14.00
	VY	572.3	216.0	3.000	-916.4	36.00	3.000
	VZ	1607.	216.0	3.000	-948.4	21.00	1.000
	MX	330.6	216.0	3.000	-255.7	36.00	3.000
	MY	1.0055E 04	21.00	1.000	-2.2775E 04	216.0	3.000
	MZ	7736.	126.0	13.00	-1.1638E 04	36.00	3.000
1/4 RADIUS BLADE LOCAL LOADS	VX	1023.	159.0	14.00	-1035.	339.0	14.00
	VY	755.2	21.00	2.000	-1441.	36.00	3.000
	VZ	2364.	216.0	3.000	-1450.	21.00	1.000
	MX	629.3	216.0	3.000	-571.8	36.00	3.000
	MY	2.6567E 04	21.00	1.000	-5.2393E 04	216.0	3.000
	MZ	1.3683E 04	21.00	2.000	-2.7165E 04	36.00	3.000
BLADE PITCH BRG REF 3/4R	VX	1607.	159.0	14.00	-1613.	339.0	14.00
	VY	1393.	216.0	3.000	-1813.	36.00	3.000
	VZ	2139.	216.0	3.000	-1449.	21.00	1.000
	MX	678.6	216.0	3.000	-672.1	36.00	3.000
	MY	4.0197E 04	21.00	1.000	-7.4810E 04	216.0	3.000
	MZ	3.6236E 04	216.0	3.000	-4.7345E 04	36.00	3.000
HUB LOADS (ROTATING)	VX	9964.	159.0	14.00	-9964.	339.0	14.00
	VY	6898.	264.0	13.00	-6898.	84.00	13.00
	VZ	2763.	21.00	3.000	-2443.	159.0	4.000
	MX	1.5791E 04	216.0	3.000	-1.5791E 04	36.00	3.000
	MY	6.0962E 04	21.00	2.000	-6.0854E 04	201.0	2.000
	MZ	8806.	159.0	14.00	-1.1474E 04	237.0	6.000
SHAFT LOADS (ROTATING)	VX	1.0382E 04	159.0	14.00	-1.0382E 04	339.0	14.00
	VY	7214.	264.0	13.00	-7214.	84.00	13.00
	VZ	2789.	21.00	3.000	-2576.	159.0	4.000
	MX	2.3652E 04	216.0	3.000	-2.3652E 04	36.00	3.000
	MY	6.1713E 04	21.00	2.000	-6.1634E 04	201.0	2.000
	MZ	8825.	159.0	14.00	-1.1453E 04	237.0	6.000
BEARING B	VX	6373.	321.0	13.00	-9681.	159.0	14.00
	VY	3267.	21.00	2.000	-7120.	36.00	3.000
	VZ	2789.	21.00	3.000	-2576.	159.0	4.000
	MX	4.5128E 04	48.00	1.000	-1.5937E 04	294.0	3.000
	MY	5.3945E 04	18.00	2.000	-1.5170E 04	69.00	13.00
	MZ	8825.	159.0	14.00	-1.1453E 04	237.0	6.000
YAW BEARING LOADS	VX	9499.	321.0	13.00	-1.4468E 04	159.0	14.00
	VY	5289.	21.00	2.000	-1.2960E 04	36.00	3.000
	VZ	4324.	291.0	5.000	-9220.	159.0	4.000
	MX	6.6056E 04	45.00	1.000	-6.9393E 04	30.00	3.000
	MY	1.3947E 05	126.0	13.00	-9.5580E 04	141.0	13.00
	MZ	2.4698E 04	39.00	1.000	-4.3336E 04	36.00	3.000
TOWER ROOT LOADS	VX	1.0250E 04	321.0	13.00	-1.6244E 04	159.0	14.00
	VY	6766.	156.0	13.00	-1.4354E 04	36.00	3.000
	VZ	1.0805E 04	159.0	14.00	-1.0247E 04	159.0	4.000
	MX	7.5356E 05	306.0	13.00	-6.7777E 05	126.0	13.00
	MY	1.1897E 06	159.0	14.00	-7.8726E 05	177.0	14.00
	MZ	8.4899E 05	141.0	13.00	-1.4868E 06	123.0	13.00

Table 2.7-10. Summary of Largest Modal Contributions, Case 4

MODAL LOAD		MAXIMUM	POSN	MODE	MINIMUM	POSN	MODE
3/4 RADIUS BLADE LOCAL LOADS	VX	34.14	162.0	14.00	-35.58	342.0	14.00
	VY	210.0	246.0	5.000	-201.1	12.00	15.00
	VZ	323.0	219.0	1.000	-366.9	39.00	1.000
	MX	31.41	282.0	4.000	-27.98	66.00	5.000
	MY	2522.	27.00	9.000	-1084.	219.0	1.000
	MZ	1183.	246.0	5.000	-1595.	12.00	15.00
1/2 RADIUS BLADE LOCAL LOADS	VX	97.08	162.0	14.00	-100.2	342.0	14.00
	VY	425.2	246.0	5.000	-415.4	102.0	4.000
	VZ	649.8	219.0	1.000	-742.1	39.00	1.000
	MX	95.75	282.0	4.000	-85.09	66.00	5.000
	MY	8405.	21.00	1.000	-7223.	219.0	1.000
	MZ	6026.	246.0	5.000	-5718.	102.0	4.000
1/4 RADIUS BLADE LOCAL LOADS	VX	217.5	162.0	14.00	-221.1	342.0	14.00
	VY	599.1	282.0	4.000	-592.1	102.0	4.000
	VZ	870.2	219.0	1.000	-1014.	21.00	1.000
	MX	195.5	282.0	4.000	-178.7	102.0	4.000
	MY	1.9180E 04	21.00	1.000	-1.6136E 04	219.0	1.000
	MZ	1.2952E 04	246.0	5.000	-1.3218E 04	102.0	4.000
BLADE PITCH BRG REF 3/4R	VX	341.6	162.0	14.00	-344.0	342.0	14.00
	VY	670.6	282.0	4.000	-639.2	102.0	4.000
	VZ	939.8	219.0	1.000	-1124.	39.00	1.000
	MX	217.6	282.0	4.000	-203.0	102.0	4.000
	MY	3.1839E 04	21.00	1.000	-2.7148E 04	219.0	1.000 ←
	MZ	2.0862E 04	282.0	4.000	-2.0035E 04	102.0	4.000
HUB LOADS (ROTATING)	VX	2121.	162.0	14.00	-2121.	342.0	14.00
	VY	1716.	282.0	4.000	-1716.	102.0	4.000
	VZ	1078.	207.0	9.000	-1086.	129.0	4.000
	MX	6147.	282.0	4.000	-6148.	102.0	4.000
	MY	6.9571E 04	39.00	1.000	-6.9444E 04	219.0	1.000
	MZ	3147.	39.00	1.000	-5205.	240.0	6.000
SHAFT LOADS (ROTATING)	VX	2210.	162.0	14.00	-2210.	342.0	14.00
	VY	1742.	282.0	4.000	-1742.	102.0	4.000
	VZ	1056.	207.0	9.000	-1153.	129.0	4.000
	MX	8244.	282.0	4.000	-8245.	102.0	4.000
	MY	7.0764E 04	39.00	1.000	-7.0654E 04	219.0	1.000
	MZ	3128.	39.00	1.000	-5203.	240.0	6.000
BEARING B	VX	645.8	114.0	11.00	-2102.	162.0	14.00
	VY	1566.	111.0	4.000	-1846.	39.00	3.000
	VZ	1056.	207.0	9.000	-1153.	129.0	4.000
	MX	5.4379E 04	60.00	1.000	-9054.	69.00	4.000
	MY	6.1308E 04	27.00	1.000	-6536.	114.0	1.000
	MZ	3128.	39.00	1.000	-5203.	240.0	6.000
YAW BEARING LOADS	VX	990.1	132.0	20.00	-3225.	162.0	14.00
	VY	3554.	111.0	4.000	-3950.	39.00	3.000
	VZ	2519.	108.0	5.000	-4164.	129.0	4.000
	MX	6.3530E 04	57.00	1.000	-2.6543E 04	36.00	3.000
	MY	7.2660E 04	27.00	1.000	-1.2233E 04	228.0	11.00 ←
	MZ	1.3503E 04	291.0	4.000	-1.4370E 04	39.00	3.000
TOWER ROOT LOADS	VX	1369.	132.0	20.00	-3518.	162.0	14.00
	VY	3867.	246.0	5.000	-5207.	39.00	3.000
	VZ	3521.	108.0	5.000	-6014.	105.0	4.000
	MX	6.5428E 04	57.00	1.000	-3.8939E 04	24.00	11.00
	MY	5.9786E 05	102.0	4.000	-3.2415E 05	288.0	5.000
	MZ	3.5594E 05	246.0	5.000	-5.0341E 05	39.00	3.000

is the same mode that causes the maximum edgewise moment (MZ). This mode shape is shown in Figure 2.7-30. It is a cyclic mode with a large coupling between the rotor and the bedplate. Similarly for Case 4, the first elastic mode is the major contributor to the flapwise moment at the blade root. With the yaw drive locked, this mode is the fundamental cyclic flap mode of the rotor and has negligible tower motion, Figure 2.7-31. However, the chordwise moment results from Mode 4 at the 90 degree model position and does show significant interaction with the tower, Figure 2.7-32. At Interface 8 (yaw bearing), the maximum yaw moment (MX) is seen to result from the same mode that causes the maximum flapwise moment at the blade root for Case 1. Although experimental data for these cases are not available at the yaw drive, other data show a large 2P one sided component which is also predicted by the analysis. In Case 1 the large vertical acceleration at 10P is due to the response in Mode 11 (MY at the yaw bearing). This mode is shown in Figure 2.7-33 and involves a resonance of the tower and stairway. These comparisons indicate that the system approach captures the major coupling between the rotor, bedplate and tower structure and predicts the cyclic loading with reasonable accuracy.

The results of the Mod-0 analysis verify the adequacy of the GETSS code. The code provides loads which are generally within 20 percent of experimental measurements and shows good to excellent agreement on harmonic content. A realistic tower shadow based on measured wind tunnel test results can be used directly in the code without modification. An outstanding feature of the code is the prediction of dynamic interactions between various parts of the system in a manner that can be readily traced and understood.

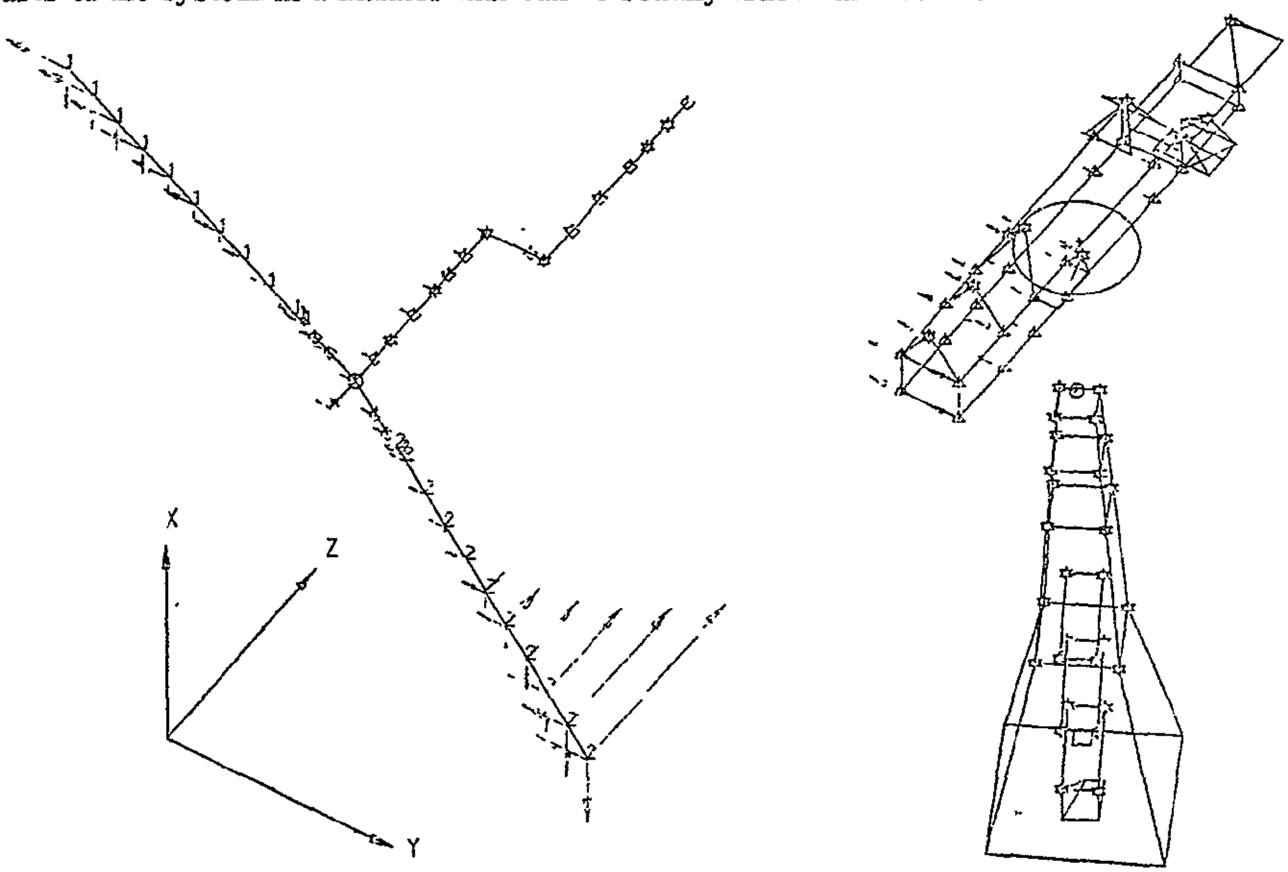


Figure 2.7-30. Mode 3 for Rotor at 45° Position, Case 1

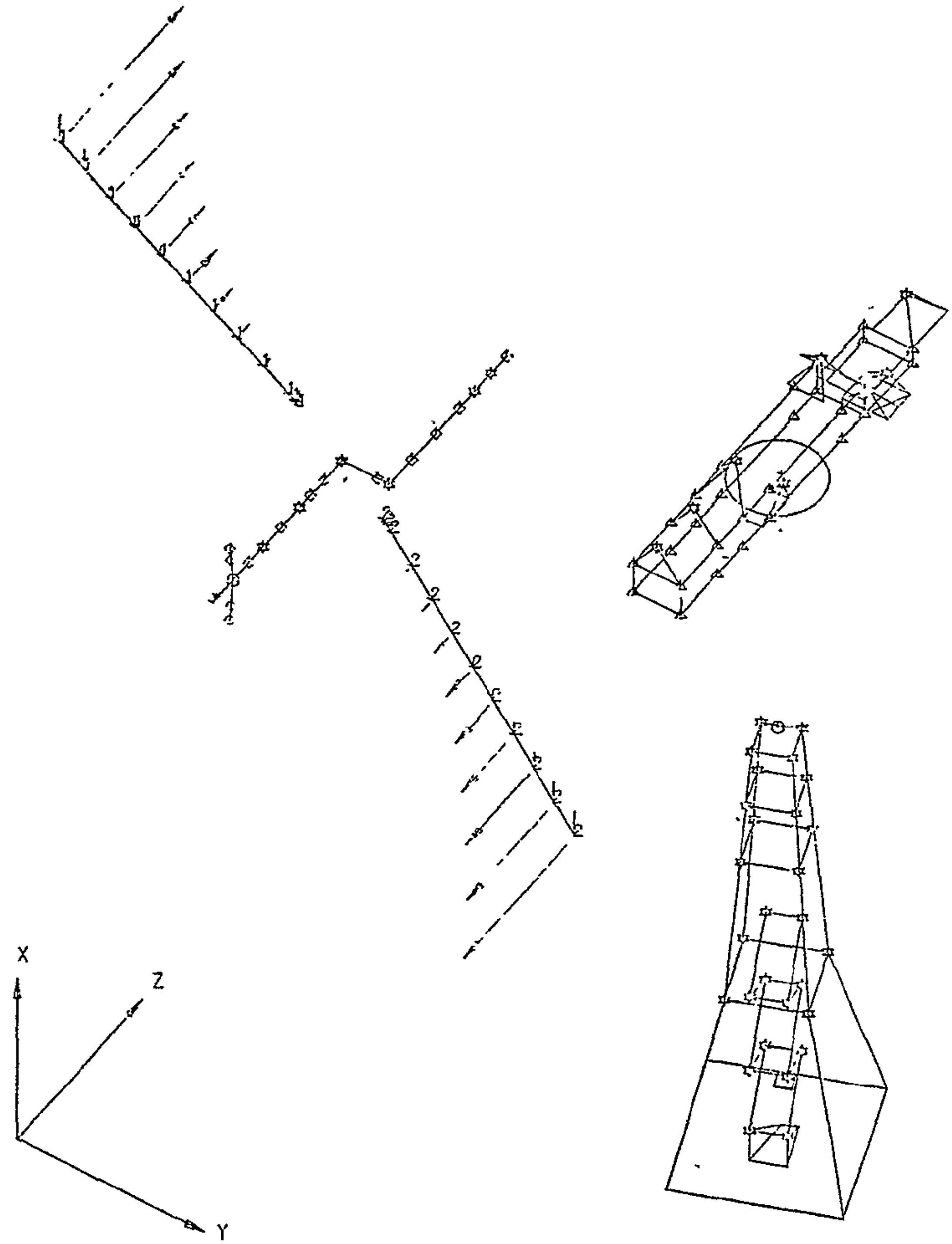


Figure 2.7-31. Mode 1 for Rotor at 45° Position, Case 4

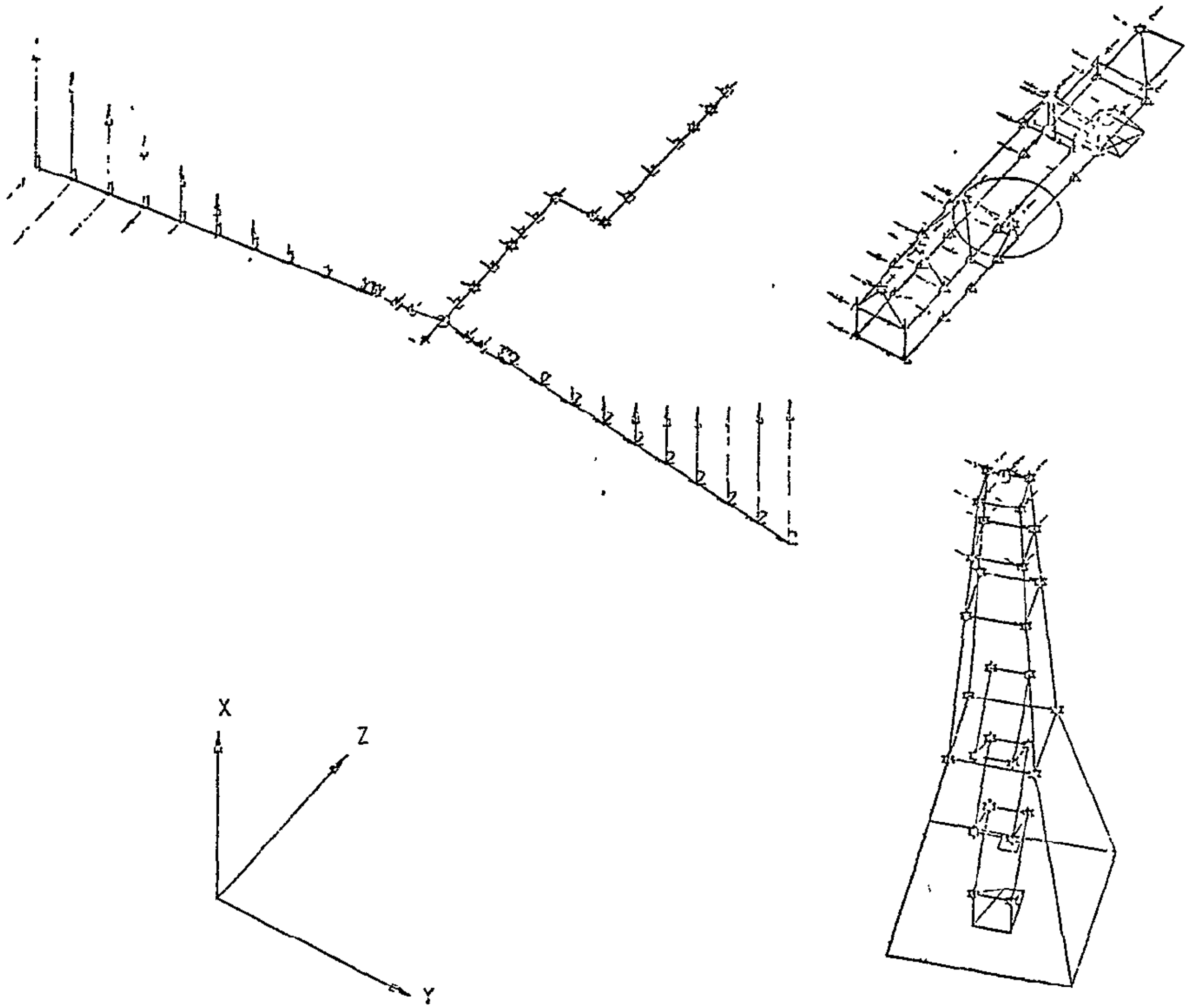


Figure 2.7-32. Mode 4 for Rotor at 90° Position, Case 4

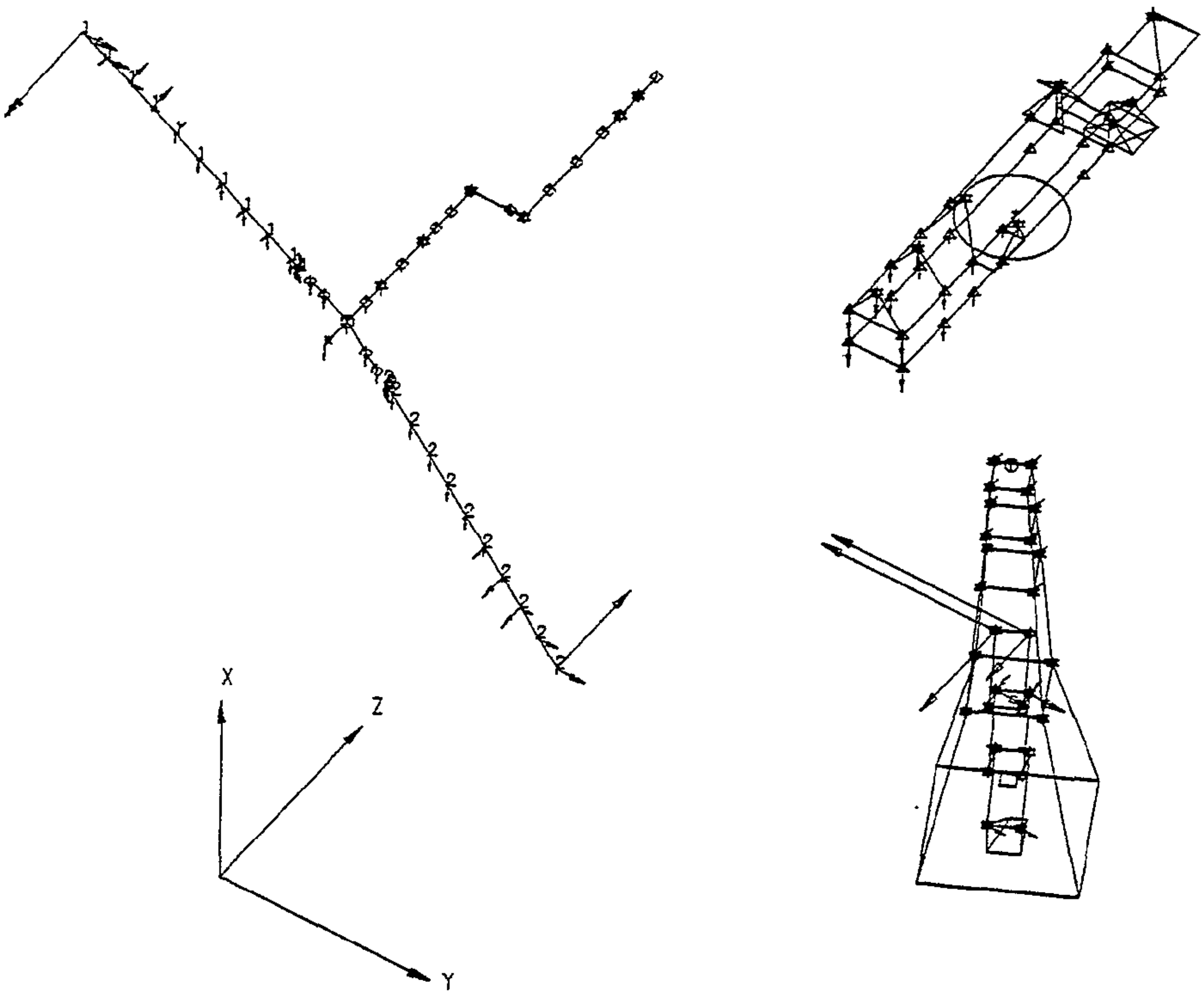


Figure 2.7-33. Tower Stair Mode Excited by 10P

2.8 SENSITIVITY ANALYSIS

Sensitivity analyses are being performed to evaluate several critical parameters in the design loads calculations. These critical parameters are:

1. SOIL FLEXIBILITY - This parameter will be varied to evaluate the change in the base flexibility of the tower and to determine the effects on the tower modal properties and their effect on the interface loads. Soil stiffness calculations will consider foundation geometry and ranges of soil stiffness.

2. TOWER STIFFNESS - This parameter will be varied to examine the change in interface loads with a range of tower stiffness values. A range of tower stiffness values will be selected such that the fundamental bending modes of the tower range from the present value of 2.2P (which meets the specification requirements) to values as high as 3.4P. Evaluation of the change in interface loading with respect to present design loads will be made. The properties of all the other substructure and stiffness links will remain constant.

3. BLADE STIFFNESS - Blade stiffness values will be increased to determine the effect on interface loads with blade stiffness. To enable only the effect of blade stiffness to be made, the tower stiffness value will be placed at about 3.4P to ensure that tower/blade coupling will not obscure blade effects.

Analysis in each of these areas is in progress and some results are expected by the time of the PDR presentation.

2.9 SUMMARY

Structural dynamic analysis are being performed to assure the dynamic adequacy of the WTG system design. Significant results obtained are as follows:

1. An analytical approach is being used which captures the dynamic interactions between various portions of the system. The system is synthesized by substructures which permit portions to be readily rotated to various positions and enable bearing stiffness to be readily varied. By using a series of linear models, the coupling causing maximum dynamic loading can be readily understood and corrective action taken. Damping of the various system modes is identified which provides an early indication of the significant modes relative to coupling with the control system and utility. Loads at major interfaces can be calculated to assure adequate treatment of the structure, yaw drive and other critical elements.

2. Code verification results indicate conservative peak and cyclic load predictions. In applying the AISC code, the peak and cyclic loads are the critical parameters. These loads are generally within approximately 20 percent and are conservative. The analysis uses a realistic tower shadow model representative of wind tunnel data considering the tapered tower geometry and increased shadow width. The

code uses 20 elastic modes of the system enabling higher order responses of the tower to be included. Major dynamic coupling is apparent and readily traceable to system characteristic. Modal test comparisons verify the adequacy of the dynamic modeling of the structure.

3. The Mod-1 tower and nacelle structure have been sized for preliminary strength and stiffness requirements. Although some member sizes are being increased and and local areas are undergoing redesign, the majority of the structure shows positive margins of safety. Of the various loading conditions, the cyclic loads appear to be the most critical because of the reduced range stress. However, conservative phasing of the applied loads should assure the adequacy of the design.

APPENDIX B

DESIGN LOAD DEFINITION

Abstract

This appendix is a copy of G.E. Space Division PIR WTG 1500-77-015; dated November 2, 1977. It defines the design loads at each of the principal WTG interfaces for critical loading conditions, both peak and cyclic. Also included is PIR WTG MOD-1-78-012B, dated March 22, 1979, which details the latest blade loads.

GENERAL ELECTRIC
SPACE DIVISION
PHILADELPHIA

PROGRAM INFORMATION REQUEST / RELEASE

*CLASS. LTR.	OPERATION	PROGRAM	SEQUENCE NO.	REV. LTR.
WTG	1500	77	015	D
PIR NO.	—1R56	— WTG —	498	D

*USE "C" FOR CLASSIFIED AND "U" FOR UNCLASSIFIED

FROM	TO
G. Sardella	W. Lucas & Distribution #8

DATE SENT	DATE INFO. REQUIRED	PROJECT AND REQ. NO.	REFERENCE DIR. NO.
11/2/77			

SUBJECT

DESIGN LOADS FOR WTG; MODEL 400 - STEEL BLADE

INFORMATION REQUESTED/RELEASED

1.0 SUMMARY

This PIR updates and supercedes the Reference 1 Loads PIR. This revision is the first load set using the Dynamic Model 400 series which represents the final tower structure geometry, the plate design bedplate, the first cut Boeing steel blade, and with a system power output rated at 1845 KW at 24.8 mph. Loads were calculated for the wind conditions of Table 1 which remain unchanged from Reference 1. To complete the load set, an approximation for peak loads for Case C, Emergency Feather, is included which shows a significant increase in peak flap moment (My) at the blade retention bearing (hub side) over the peak moment calculated for case B. This load is an approximation based on steady-state assumptions to indicate the magnitude of load that could occur at the Hub and which will require further dynamic analysis. Case D has been updated to reflect blade area changes from the Lockheed configuration. This load set shall be used to check current designs and will be updated when the final steel blade configuration has been selected.

2.0 OPERATING CONDITIONS

The loads analysis was performed for Table 1 operating conditions. These conditions are unchanged from the previous load set except for the new case identification letters A thru D. Case C, Emergency Feather is included since the overspeed which would occur under this condition could give rise to larger flap moments, My, at the blade retention bearing than all of Case B conditions. (This may also be a consideration for blade design to buckling loads). To approximate this condition a 40% overspeed was assumed and the blade angle was trimmed for zero torque. A steady state loads case with 50 mph airspeed (40% gust) was calculated which produced the Table 5 load set. A transient load analysis is under consideration at this time and when performed may provide a more accurate evaluation of this load.

3.0 LOADS ANALYSIS

For this loads analysis, Dynamic Model 413 was used to represent the WTG system. This model includes the current tower structure geometry, the "plate design" bedplate and the first cut Boeing steel blade. This blade is similar to configuration 2 of the blade parametric studies and is rated at 1845 KW at 24.8 mph. The blade weight is approximately 19850 lbs and the system weight corresponds to that reported FW 31 (see Reference 2).

PAGE NO. ____ OF ____

RETENTION REQUIREMENTS

COPIES FOR	MASTERS FOR
☐ 1 MO.	☐ 3 MOS.
☐ 3 MOS.	☐ 6 MOS.
☐ 6 MOS.	☐ 12 MOS.
☐ MOS.	☐ MOS.
☐	☐ DO NOT DESTROY

The WINLD code of the GETSS' program calculated the aerodynamic and gravitational system forces for the Table 1 operating conditions. With these forces, the aero-damping coefficients from the QAERO code, and the Model 413 system modes and frequencies, the system loads were calculated for several interfaces and components. Loads calculated were in terms of shear and moments in the three orthogonal system axes (both rotating and fixed) and also in terms of accelerations and deflections for the same axes. The following interfaces were defined:

a. Loads

(1) Blade, sta. .75R
(2) " " .50R
(3) " " .25R
(4) " " .1R
(5) Blade ret. bg (blade side)
(6) Blade ret. bg (hub side)
(7) Main Rotor bg (Hub side)
(8) " " " (bedplate side)
(9) Yaw Bearing
(10) Tower base

b. Accelerations

(1) Main Rotor brg.
(2) Transmission
(3) Generator
(4) Yaw bearing
(5) Bedplate corner, +Y+Z
(6) " " , -Y+Z
(7) " " , +Y-Z
(8) " " ,-Y-Z

Design loads for the WTG structure are given for the six structural interfaces described above (interfaces 6-10). The Blade stations are not given but can be obtained in plotted form for steady wind speeds in Reference 3. Values of the mean, peak and cyclic components of forces and moment are given at each of the structural interfaces. Accelerations at the major masses listed above are also tabulated along with bedplate corner accelerations and the yaw bearing (tower top). A summary of the data is as follows:

a. Tabulations

Table 1 Wind Conditions for Design
" 2 Fatigue Loads - Case A
" 3A Dynamic and Gravity Loads Case B Upgust, +41° in flow
" 3B " " " " " " , -41° in flow
" 4A " " " " " Downgust, +41° in flow
" 4B " " " " " Downgust, -41° in flow
" 5 Emergency Feather - Case C
" 6 Hurricane - Case D
" 7 Component Accelerations - Fatigue, Case A
" 8A " " - Upgust, Case B +41° in flow
" 8B " " - " " -41° in flow
" 9A " " - Downgust " +41° in flow
" 9B " " - " " -41° in flow

b. Figures

Figure 1 Sign Convention - Rotating System
Figure 2 Sign Convention - Stationary System

4.0 LOADS APPLICATION

The WTG structural design shall be evaluated for each of the four operating conditions given in Table 1. Loads for these cases are presented as described above in Tables 1-9 along with steady state aerodynamic loads given in Table 10. In the application of the loads it is important to note that two set of coordinates are required. For blade loads, the blade retention bearing and the rotating hub, a rotating set of coordinates was required and are defined in Figure 1. This system should be used for load applications at these interfaces. For the hub bearing (B/P side), yaw bearing and tower base interfaces; however, a stationary axes set which is defined in Figure 2 is referenced. This stationary set should be used for application of loads at these interfaces. Acceleration loads are given for component and bedplate locations which are referenced to the stationary axis.

For the fatigue condition of Case A, the cyclic loads given in Table 2 shall be applied to the system. Accelerations calculated for this condition are also given in Table 7 and are provided to complete the load configuration. The maximum value of stress range determined from these loads shall be kept equal to or below the constant amplitude fatigue limit specified by the latest ASSHTO code. This case will cover the range of operating velocities and includes inflow variation and gust dispersion.

Case B will be considered a peak loading case and shall be used to compute peak stresses. These stresses will be checked against the peak allowable stresses of the latest AASHTO code and the appropriate buckling criteria. Since both up and down gusts are considered for each maximum inflow variation each load set of Tables 3 and 4 must be evaluated.

Case C and Case D are extreme loading conditions occurring very infrequently. Peak loads occurring from these conditions shall be evaluated for stresses to the allowables specified by the code for such infrequent loading not to the peak stress allowables used for Case B.

5.0 REFERENCES

1. Sardella, G., "Design Loads for WTG Structure and Component", PIR WTG-1500-77-015C, July 18, 1977.

2. Fuoco, J., "Weight Breakdown, Center of Mass, and Intertias for WTG, FW 31", PIR WTG-1500-77-094, August 19, 1977.

3. "Specification for WTG Rotor Blade", No. 273A6684.

TABLE 1. WIND CONDITIONS FOR WTG STRUCTURE AND COMPONENTS

Case A	Accumulated fatigue over the entire wind duration regime from cut-in to cut-out velocity. Inflow angles to 20° included.
Case B	Max up gust 35 to 50 MPH Max down gust 35 to 20 MPH Blade disk fully immersed. Gust period=7sec. w/profile of Eqn 2, Para 1.3.2.2 Ex B of SOW.
Case C	Emergency Feather RPM: $n_0 < n \leq 1.4 n_0$ — Pitch Rate: 14°/sec RPM: $n < n_0$ — Pitch Rate: 8°/sec
Case D	Hurricane: 120 MPH, FEATHERED

c4

* Ref to blade angle @ 3/4 R
+ Blade coord (rotating)
** Tower coord (stationary)

STEEL BLADE

TABLE 2. FATIGUE LOADS - CASE A

MOD(413

LOAD LBS for SHEAR FT-LBS for MOMENT		MEAN	CYCLIC (±)	PEAK
BLADE RET BRNG (BLADE SIDE)* +	Vx		13 167	
	Vy		26 369	
	Vz		21 541	
	Mx		20 340	
	My		1 139 521	
	Mz		1 001 580	
BLADE RET BRNG (HUB SIDE)+	Vx		13 167	
	Vy		27 780	
	Vz		19 003	
	Mx		20 340	
	My		1 075 275	
	Mz		1 088 295	
HUB BRNG (HUB SIDE)+	Vx		105 165	
	Vy		121 371	
	Vz		26 532	
	Mx		909 225	
	My		1 874 550	
	Mz		250 000	
HUB BRNG (B/P SIDE)**	Vx		26 330	
	Vy		31 424	
	Vz		26 532	
	Mx		755 486	
	My		573 726	
	Mz		250 000	
YAW BRNG**	Vx		82 161	
	Vy		28 046	
	Vz		35 903	
	Mx		984 780	
	My		539 953	
	Mz		150 945	
TOWER BASE**	Vx		139 650	
	Vy		35 745	
	Vz		43 495	
	Mx		1 356 315	
	My		5 390 733	
	Mz		4 239 750	

* Ref to blade angle [illegible]

\+ Blade coord (rotating)

** Tower coord (stationary)

TABLE 3A,

DYNAMIC + GRAVITY LOADS — CASE B

UPGUST: 35 → 50 + 41° INFLOW

MODEL 4

16

LOAD (LBS. for SHEAR, FT-LBS for MOMENT)		MEAN	CYCLIC (±)	PEAK
BLADE RET BRNG (BLADE SIDE)* +	V_x	292,300	8,785	301,085
	V_y	-14,171	19,321	-33,492
	V_z	34,339	23,662	58,001
	M_x	166	16,539	16,705
	M_y	-1,121,800	1,059,100	-2,210,900
	M_z	-801,740	842,050	-1,643,790
BLADE RET BRNG (HUB SIDE)+	V_x	292,300	8,785	301,085
	V_y	-6,809	19,892	-26,701
	V_z	36,528	21,178	57,706
	M_x	166	16,539	16,705
	M_y	-1,277,200	993,850	-2,271,050
	M_z	-535,420	903,530	-1,438,[illegible]
HUB BRNG (HUB SIDE)+	V_x	-349	73,118	-73,467
	V_y	-9	84,290	-84,299
	V_z	-48,775	9,390	-58,165
	M_x	310	665,030	665,340
	M_y	764	2,128,800	2,129,564
	M_z	-1,071,400	95,055	-1,166,455
HUB BRNG (B/P SIDE)**	V_x	-89,730	18,288	-108,018
	V_y	-1,220	22,494	-23,714
	V_z	-48,775	9,390	-58,165
	M_x	338,510	937,470	1,275,985
	M_y	1,395,100	720,530	2,115,630
	M_z	729,945	95,055	-825,000
YAW BRNG**	V_x	-339,730	51,158	-390,888
	V_y	-55	21,051	-21,106
	V_z	-48,533	18,458	-66,991
	M_x	157,300	1,326,800	1,484,100
	M_y	1,274,900	661,170	1,93[illegible],070
	M_z	-728,445	121,360	-849,805
TOWER BASE**	V_x	-688,990	87,548	-776,538
	V_y	-4,632	28,066	-32,698
	V_z	-50,018	30,798	-80,816
	M_x	177,190	1,625,000	1,802,190
	M_y	7,614,200	2,847,400	10,461,600

* Ref to blade angle [illegible]
\+ Blade coord (rotating)
** Tower coord (stationary)

TABLE 3B

DYNAMIC + GRAVITY LOADS – CASE B

UPGUST: 35→50 −41° INFLOW

MODEL 164

LOAD LBS' for SHEAR FT-LBS for MOMENT		MEAN	CYCLIC (±)	PEAK
BLADE RET BRNG (BLADE SIDE)* +	Vx	292 130	7 442	299 572
	Vy	−11 436	16 379	−27 815
	Vz	38 766	14 594	53 360
	Mx	383	10 375	10 758
	My	−1 337 400	845 910	−2 183 310
	Mz	−660 760	610 160	−1 270 920
BLADE RET BRNG (HUB SIDE)+	Vx	292 130	7 442	299 572
	Vy	−4 636	15 402	−20 038
	Vz	39 349	16 314	55 663
	Mx	383	10 375	10 758
	My	−1 397 900	898 780	−2 296 680
	Mz	−420 930	561 770	−982 700
HUB BRNG (HUB SIDE)+	Vx	−181	61 377	−61 558
	Vy	−6	73 679	−73 685
	Vz	−40 736	26 369	−67 105
	Mx	562	492 050	492 612
	My	435	740 180	740 615
	Mz	−985 050	144 290	−1 129 340
HUB BRNG (B/P SIDE)**	Vx	−82 091	13 162	−95 253
	Vy	−4 454	18 782	−23 236
	Vz	−40 736	26 369	−67 105
	Mx	−355 230	411 120	−766 350
	My	392 960	372 150	765 110
	Mz	−680 710	144 290	−825 000
YAW BRNG**	Vx	−334 330	32 367	−366 697
	Vy	−1 750	13 235	−14 985
	Vz	−42 063	36 132	−78 195
	Mx	−436 666	641 460	−1 078 126
	My	49 928	563 190	613 118
	Mz	−702 160	178 090	−880 250
TOWER BASE**	Vx	−632 710	54 880	−687 590
	Vy	−7 780	16 001	−23 781
	Vz	−42 512	51 100	−93 612
	Mx	−631 150	955 210	−1 586 360
	My	5 914 900	6 425 000	12 339 900
	Mz	−1 135 260	2 032 500	−3 167 760

* Ret to blade angle @ 3/4 R
\+ Blade coord (rotating)
** Tower coord (stationary)

TABLE 4A.

DYNAMIC + GRAVITY LOADS — CASE B

DOWNGUST: 35 → 20 MPH; + 41° INFLOW

MODEL 16F

LOAD LBS for SHEAR FT-LBS for MOMENT		MEAN	CYCLIC (±)	PEAK
BLADE RET BRNG (BLADE SIDE)* +	Vx	292 310	9 570	301 880
	Vy	−10 961	19 320	−30 201
	Vz	58 787	12 131	70 918
	Mx	−53	15 121	−15 174
	My	−2 858 800	521 010	−3 379 810
	Mz	−538 400	766 760	−1 305 150
BLADE RET BRNG (HUB SIDE)+	Vx	292 310	9 570	301 880
	Vy	11	20 819	20 830
	Vz	60 600	9 945	70 545
	Mx	−53	15 121	15 174
	My	−2 966 600	446 730	−3 413 330
	Mz	1 722	846 280	848 006
HUB BRNG (HUB SIDE)+	Vx	−452	70 710	−71 162
	Vy	120	81 965	82 085
	Vz	−5 068	5 776	−10 844
	Mx	770	686 580	687 350
	My	918	931 060	931 978
	Mz	2 321	7 423	9 744
HUB BRNG (B/P SIDE)**	Vx	−84 525	21 842	−106 367
	Vy	−1 338	24 415	−25 753
	Vz	−5 068	5 776	−10 844
	Mx	289 500	282 780	572 280
	My	727 560	436 621	1 164 181
	Mz	2 321	7 423	9 744
YAW BRNG**	Vx	−337 610	67 049	−404 659
	Vy	1 286	20 098	21 384
	Vz	−5 386	13 302	−18 688
	Mx	315 920	606 540	922 460
	My	244 670	666 930	911 600
	Mz	1 785	110 060	111 845
TOWER BASE**	Vx	−688 210	113 660	−801 870
	Vy	−4 470	25 440	−29 910
	Vz	−8 937	25 247	−34 184
	Mx	197 230	955 780	1 153 010
	My	1 296 100	2 423 000	3 719 100
	Mz	−221 480	3 006 100	−3 [illegible]

14

* Ret [illegible]
\+ Blade coord (rotating)
** Tower coord (stationary)

TABLE 4B,

DYNAMIC + GRAVITY LOADS – CASE 3

DOWNGUST: 35 → 20 MPH, −41° INFLOW

MODEL 16F

LOAD (LBS for SHEAR, FT-LBS for MOMENT)		MEAN	CYCLIC (±)	PEAK
BLADE RET BRNG (BLADE SIDE)* +	Vx	292 320	10 047	302 367
	Vy	−10 756	18 542	−29 298
	Vz	60 195	11 796	71 991
	Mx	150	13 003	13 153
	My	−2 908 900	529 840	−3 438 740
	Mz	−541 040	728 700	−1 269 740
BLADE RET BRNG (HUB SIDE)+	Vx	292 320	10 047	302 367
	Vy	144	19 269	19 413
	Vz	59 377	12 690	72 067
	Mx	150	13 003	13 153
	My	−2 909 900	572 830	−3 482 730
	Mz	−5 212	763 680	−768 892
HUB BRNG (HUB SIDE)+	Vx	−510	75 533	−76 043
	Vy	3	79 761	79 764
	Vz	−5 203	6 246	−11 449
	Mx	260	628 910	629 170
	My	475	1 283 000	1 283 475
	Mz	−9 812	8 099	−17 911
HUB BRNG (B/P SIDE)**	Vx	−83 993	19 633	−103 656
	Vy	−3 955	23 538	−27 493
	Vz	−5 203	6 246	−11 449
	Mx	−266 990	499 200	−766 190
	My	914 680	391 090	1 305 770
	Mz	−9 812	8 099	−17 911
YAW BRNG**	Vx	−339 460	73 105	−412 565
	Vy	−2 543	20 454	−22 997
	Vz	−6 477	14 420	−20 897
	Mx	−397 970	725 480	−1 123 450
	My	563 610	393 490	957 100
	Mz	−30 116	113 020	−143 136
TOWER BASE**	Vx	−689 550	124 350	−813 900
	Vy	−9 380	25 255	−34 635
	Vz	−9 091	24 998	−34 089
	Mx	−412 840	1 016 500	−1 429 340
	My	1 360 600	2 495 700	3 856 300
	Mz	−852 460	3 091 000	−3 943 460

16

* Ref ... blade angle ...
\+ Blade coord (rotating)
** Tower coord (stationary)

TABLE 5. ~~EMERGENCY FEATHER~~ OVERSPEED LOADS for CASE (
DYNAMIC + GRAVITY LOADS

LOAD LBS for SHEAR FT-LBS for MOMENT		MEAN	CYCLIC (±)	PEAK
BLADE RET BRNG (BLADE SIDE)* +	V_x	573 450	18 619	592 069
	V_y	−35 625	32 900	−68 525
	V_z	116 910	21 319	138 229
	M_x	−295	45 260	−45 555
	M_y	−5 813 200	749 230	−6 562 430
	M_z	−1 779 100	1 362 500	−3 141 600
BLADE RET BRNG (HUB SIDE)+	V_x	573 450	18 619	592 069
	V_y	−255	34 279	−34 534
	V_z	122 700	22 822	145 522
	M_x	−295	45 260	−45 555
	M_y	−6 114 700	836 040	−6 950 740
	M_z	1 071	1 449 300	1 450 371
HUB BRNG (HUB SIDE)+	V_x	244	105 370	105 614
	V_y	−37	114 080	−114 117
	V_z	−5 621	16 891	−22 512
	M_x	−125	1 136 200	−1 136 325
	M_y	1 247	2 058 000	2 059 247
	M_z	4 482	19 684	24 166
HUB BRNG (B/P SIDE)**	V_x	−83 903	44 877	−128 780
	V_y	855	35 885	36 740
	V_z	−5 621	16 891	−22 512
	M_x	68 243	820 830	889 073
	M_y	1 241 300	898 060	2 139 360
	M_z	4 482	19 684	24 166
YAW BRNG**	V_x	−329 950	161 210	−491 160
	V_y	−4 544	23 680	−28 224
	V_z	−7 621	40 905	−48 526
	M_x	197 720	1 640 200	1 837 920
	M_y	791 790	825 730	1 617 520
	M_z	−49 513	125 830	−175 343
TOWER BASE**	V_x	−674 750	273 950	−948 700
	V_y	−14 230	36 674	−50 904
	V_z	−2 173	67 758	−69 931
	M_x	302 520	1 770 600	2 073 120
	M_y	2 066 300	6 435 100	8 501 400
	M_z	−1 [illegible]	4 562 600	−5 559 200

TABLE 6. HURRICANE - CASE D

WT413 BLADES FEATHERED AT 90°, 270°

INTERFACE		60 MPH NORMAL TO BLADE	120 MPH HEADWIND	120 MPH BROADSIDE TO NACELLE	120 MPH HEADWIND MAX CL (1)
BLADE *	Vx	—	—	—	—
RET. BRNG	Vy	—	—	—	-7,822/-4,266
	Vz	-45,742	-24,548	-24,548	-45,742
	Mx	—	—	—	—
	My	1,583,026	663,372	663,372	2,324,863
	Mz		—	—	-332,973/-256,432
HUB BRNG **	Vx	-150,583	-89,995	-89,995	-156,666
(STATIONARY)	Vy		—	—	739
	Vz	—	—	—	11,824
	Mx	—	—	—	-86,409
	My	1,237,200	616,130	616,130	1,696,211
	Mz	—	—	—	—
YAW BRNG **	Vx	-390,436	-338,387	-338,387	-407,058
	Vy	—	—	-38,644	739
	Vz	—	-14,523	—	-2,699
	Mx	—	—	288,638	-86,409
	My	988,525	245,333	138,831	1,793,151
	Mz	—	—	-283,389	—
TOWER BASE **	Vx	-732,756	-680,707	-680,707	-749,288
	Vy	—	—	-138,944	739
	Vz	—	-114,823	—	-102,999
	Mx	—	—	288,638	-86,409
	My	988,525	8,317,466	138,831	8,283,700
	Mz	—	—	-11,597,787	—

* REF TO ROTATING COORD AXES

** REF " STATIONARY " "

(1) VALUES GIVEN ARE FOR THE 90° POSITION BLADE. WHERE THIS VALUE DIFFERS FOR THE 270° POSITION BLADE, THAT VALUE FOR THE 270° BLADE IS GIVEN AS THE SECOND VALUE.

10-17-77

MODEL 4

TABLE 7. COMPONENT ACCELERATIONS FOR FATIGUE, CASE A

		MEAN	CYCLIC (±)	PEAK
HUB BRG	X		± .0214	
	Y		± .1510	
	Z		± .0344	
TRANSMISSION	X		± .0229	
	Y		± .0869	
	Z		± .0340	
GENERATOR	X		± .0369	
	Y		± .1900	
	Z		± .0345	
YAW BEARING	X		± .0192	
	Y		± .0582	
	Z		± .0329	
BEDPLATE CORNER +Y, +Z	X		± .0459	
	Y		± .2491	
	Z		± .0675	
BEDPLATE CORNER −Y, +Z	X		± .0506	
	Y		± .2492	
	Z		± .0759	
BEDPLATE CORNER +Y, −Z	X		± .0201	
	Y		± .1293	
	Z		± .0714	
BEDPLATE CORNER −Y, −Z	X		± .0215	
	Y		± .1293	
	Z		± .0629	

MODEL 413

TABLE 8A. CASE B COMPONENT ACCELERATIONS (g's)

UPGUST: 35 → 50 MPH + 41° INFLOW

		MEAN	CYCLIC(±)	PEAK
HUB BRG	X		±.2098	.0311
	Y		±.1381	-.1526
	Z		±.0438	-.0455
TRANSMISSION	X		±.0220	.0293
	Y		±.1162	.1191
	Z		±.0437	-.0463
GENERATOR	X		±.0354	.0453
	Y		±.2444	.2585
	Z		±.0441	-.0492
YAW BEARING	X		±.0185	.0267
	Y		±.0595	-.0612
	Z		±.0413	-.0435
BEDPLATE CORNER +Y, +Z	X		±.0491	.0626
	Y		±.3189	.3391
	Z		±.0806	-.0904
BEDPLATE CORNER -Y, +Z	X		±.0438	.0551
	Y		±.3188	.3390
	Z		±.0795	-.0848
BEDPLATE CORNER +Y, -Z	X		±.0198	.0299
	Y		±.1151	-.1303
	Z		±.0759	-.0844
BEDPLATE CORNER -Y, -Z	X		±.0209	.0307
	Y		±.1151	-.1303
	Z		±.0743	-.0794

10-18-77

CASE B

TABLE 8B. COMPONENT ACCELERATIONS (g's)

UPGUST: 35 → 50 MPH — 41° INFLOW

		MEAN	CYCLIC (±)	PEAK
HUB BRG	X		±.0123	.0169
	Y		±.1468	−.1481
	Z		±.0522	.0572
TRANSMISSION	X		±.0163	.0218
	Y		±.0856	−.0886
	Z		±.0522	.0571
GENERATOR	X		±.0286	.0369
	Y		±.2007	−.2037
	Z		±.0524	.0575
YAW BEARING	X		±.0117	.0169
	Y		±.0472	−.0476
	Z		±.0495	.0533
BEDPLATE CORNER +Y, +Z	X		±.0392	.0503
	Y	-	±.2702	−.2733
	Z		±.0895	.0921
BEDPLATE CORNER −Y, +Z	X		±.0378	.0482
	Y		±.2701	−.2732
	Z		±.0831	.0897
BEDPLATE CORNER +Y, −Z	X		±.0109	.0167
	Y		±.1222	−.1230
	Z		±.0845	.0873
BEDPLATE CORNER −Y, −Z	X		±.0132	.0178
	Y		±.1222	−.1230
	Z		±.0779	.0837

10-18-77

MODEL 413

CASE B

TABLE 9A. COMPONENT ACCELERATIONS (g's)

DOWNGUST: 35 → 20 MPH ; +41° INFLOW

		MEAN	CYCLIC (±)	PEAK
HUB BRG	X		± .0239	.0345
	Y		± .1602	−.1781
	Z		± .0333	.0427
TRANSMISSION	X		± .0255	.0351
	Y		± .0863	.0919
	Z		± .0326	.0414
GENERATOR	X		± .0413	.0561
	Y		± .1964	.2067
	Z		± .0322	.0391
YAW BEARING	X		± .0215	.0308
	Y		± .0608	−.0714
	Z		± .0315	.0394
BEDPLATE CORNER +Y, +Z	X		± .0566	.0767
	Y		± .2622	.2757
	Z		± .0686	.0696
BEDPLATE CORNER −Y, +Z	X		± .0519	.0707
	Y		± .2621	.2750
	Z		± .0694	.0777
BEDPLATE CORNER +Y, −Z	X		± .0229	.0339
	Y		± .1366	−.1528
	Z		± .0645	.0661
BEDPLATE CORNER −Y, −Z	X		± .0242	.0340
	Y		± .1366	−.1527
	Z		± .0647	.0726

10-18-77

MODEL 413

CASE B

TABLE 9B. COMPONENT ACCELERATIONS

DOWNGUST: 35 → 20 MPH ; −41° INFLOW

		MEAN	CYCLIC(±)	PEAK
HUB BRG	X		±.0261	.0378
	Y		±.1599	−.1614
	Z		±.0336	.0442
TRANSMISSION	X		±.0284	.0392
	Y		±.0825	.0867
	Z		±.0327	.0431
GENERATOR	X		±.0458	.0622
	Y		±.1947	.2049
	Z		±.0339	.0423
YAW BEARING	X		±.0237	.0340
	Y		±.0572	.0579
	Z		±.0319	.0408
BEDPLATE CORNER +Y, +Z	X		±.0623	.0845
	Y		±.2631	.2741
	Z		±.0753	.0832
BEDPLATE CORNER −Y, +Z	X		±.0577	.0786
	Y		±.2630	.2740
	Z		±.0633	.0675
BEDPLATE CORNER +Y, −Z	X		±.0247	.0367
	Y		±.1362	−.1371
	Z		±.0710	.0788
BEDPLATE CORNER −Y, −Z	X		±.0267	.0378
	Y		±.1362	−.1371
	Z		±.0583	.0621

10-18-77

GENERAL ELECTRIC
SPACE DIVISION
PHILADELPHIA

PROGRAM INFORMATION REQUEST / RELEASE

	*CLASS. LTR.	OPERATION	PROGRAM	SEQUENCE NO.	REV LTR.
	WTG	MOD-1	78	012	
PIR NO	U	— 1R44	—MOD-1 —	573	B

*USE "C" FOR CLASSIFIED AND "U" FOR UNCLASSIFIED

FROM	TO
J. Strain	F. Stearns

DATE SENT	DATE INFO. REQUIRED	PROJECT AND REQ. NO.	REFERENCE DIR. NO.
3/22/79			

SUBJECT

BOEING STEEL BLADE LOADS

INFORMATION REQUESTED/RELEASED

1.0 SUMMARY

This PIR presents blade loads for the MOD-1 WTG models 621 and 622. Model 621 was for a 24.8 mph wind, trimmed to rated power (2043 KW at the shaft), and running at 35 RPM. Model 622 was for a 35 mph wind, trimmed to zero power, and running at 38.85 RPM.

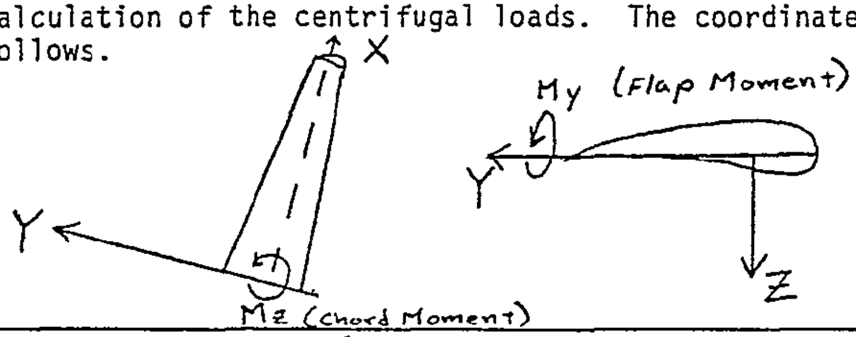
2.0 DISCUSSION

The Boeing steel blade properties which were used in these models were the revision F inertia, area, and elastic axis data, and the weight curve labeled revision G (total weight of 19618 lb. excluding tip weight). Additional weight was distributed by area among the nine blade sections in the model to bring the total to 19917 lb.; 440 lb. of tip weight was included in both models.

After the aforementioned models were assembled, the Boeing blade weight estimate was increased to 20,222 lb. exclusive of the tip weights. To ascertain the effect of this change, the blade was remodeled, distributing the additional 305 lb. by area and the cantilevered frequencies were determined. The frequency change was found to be not significant, and it was decided that a revision of the assembled WTG models was not warranted.

3.0 RESULTS

Table 1 contains the results of the calculations for the model 621 and 622 loads. The means have been corrected to account for the effect of blade flexibility in the calculation of the centrifugal loads. The coordinate system for these loads is as follows.



+Y is always along chord, out trailing edge

Blade rotation is such that Mz is negative

W. Lucas - CCF#7
G. Sardella
J. Strain

F. Stearns
J. Altpater
1R44 File (1)

PAGE NO. ____ OF ____

✓ RETENTION REQUIREMENTS

COPIES FOR	MASTERS FOR
☐ 1 MO	☐ 3 MOS.
☐ 3 MOS	☐ 6 MOS.
☐ 6 MOS.	☐ 12 MOS.
☐ MOS.	☐ MOS.
☐	☐ DO NOT DESTROY

TABLE 1 BLADE BENDING MOMENTS

(in-lb x 10^{-6})

621A

25 MPH	FLAP		CHORD	
35 RPM	CYCLIC	MEAN	CYCLIC	MEAN
.9R	.046	-.031	.033	.004
.8R	.101	-.154	.160	-.016
.7R	.278	-.356	.372	-.076
.5R	.926	-1.693	1.430	-.438
.3R	2.049	-5.019	3.473	-1.520
.1R	4.007	-9.582	6.413	-3.471

622A

35 MPH	FLAP		CHORD	
38.85 RPM	CYCLIC	MEAN	CYCLIC	MEAN
.9R	.082	.243	.057	-.030
.8R	.157	-.320	.202	-.209
.7R	.415	-1.576	.439	-.553
.5R	1.445	-6.558	1.526	-2.069
.3R	3.290	-15.082	3.599	-5.383
.1R	6.141	-25.632	6.442	-10.684

APPENDIX C

SHIPPING AND ASSEMBLY

Abstract

This appendix is a summary of shipping and assembly requirements, used for preliminary planning of these activities.

APPENDIX C

SHIPPING AND ASSEMBLY

1. Introduction

In the design of the MOD-1 WTG, consideration was given to the requirements for shipping and assembly. The general requirements specified by NASA included the following:

(a) The design shall provide for a maximum of shop assembly and a minimum of field assembly prior to erection.

(b) Consideration should be given to transportation via existing surface vehicles.

(c) The installation site may be anywhere from Alaska to the Caribbean.

These requirements were aimed at minimizing shipping and installation costs while avoiding undue technical risks and schedule delays. Once the installation site had been identified as Boone, North Carolina, a significant parameter was eliminated from the equation and it was possible to complete definite cost analyses for competing approaches. The two basic approaches were to ship the nacelle as a complete assembly and erect it on the tower in a single lift, or to ship and erect subassemblies that could be handled by less costly vehicles and lifting equipment. The Boone site favored the latter approach. The detail requirements and procedures for handling the MOD-1 WTG as subassemblies not exceeding 50 tons in weight are described in the following paragraphs.

2. Shipping Plan

The overall shipping and assembly plan is illustrated in Figure 1. Major components are shipped from their point of manufacture to the assembly and test facility in Philadelphia, known as the Riverside facility. Assembly procedures at the test facility will parallel those to be used on site, so that after disassembly the shipping configuration of major sub-assemblies will correspond to the units lifted to the top of the tower, with minor differences noted later. Approximately ten trucks will be required to ship all equipment from the test facility to the site. Major components that are not required at the test facility but will be shipped directly from the point of manufacture to the site will include (a) blades, (b) tower, and (c) substation transformer.

2.1 SHIPPING CONFIGURATION OF MAJOR SUB-ASSEMBLIES

The shipping configurations of major subassemblies are summarized in Table I, listing weight and envelope dimensions. Note that items exceeding 75,000 lb. in weight and 12 feet in width require permits for road travel. Permitted loads may travel only in daylight hours, excluding holidays and weekends and overdemensional loads will require escorts. Where possible, the overwidth loads (bedplate, yaw section, pintte, and fairing) will be shipped in convoy to minimize escort requirements.

2.2 SPECIAL HANDLING AND PACKING REQUIREMENTS

(a) Lifting lugs will be provided to facilitate handling of the bedplate, rotor assembly, and yaw section. The two forward lifting lugs on the bedplate are bolted in place and may be removed during shipment to limit the envelope width. The gearbox, generator, fairing, and ground enclosure are equipped with integral lifting lugs.

(b) Machined mating surfaces, such as the bottom surface of the bedplate, will be protected during shipment by a 3/4 inch thick plywood sheet. Internal packing, bracing, and other special procedures required in preparing for shipment will be performed during disassembly at the test facility, while protective coverings and tiedowns will be provided by the shipper.

(c) Large bearings, such as the yaw bearing, are vulnerable to damage during shipment and must be protected against severe shocks. One precaution will be to completely fill the housing with a light oil.

2.3 BLADE SHIPMENT

Each of the blades will be shipped in special fixtures, tilted at 26° to lower the height for highway movement. Each blade, in its shipping configuration will weight approximately 27,800 lbs. Envelope dimensions will be 101 ft., 3 in. long, 7 ft., 3 in. wide, and 12 ft., 4 in. high. The blade tip will overhang the rear fixture support by 25 feet.

For rail travel from Seattle, Washington to Lenoir, North Carolina, each blade will be centered on one 89 ft. steel deck flat car with a flat deck idler car positioned at the rear to provide clearance for the overhanging blade tip. At Lenoir the blades will be loaded and secured to a truck tractor and a steerable tag vehicle for transport to the site.

3.0 SITE ASSEMBLY PROCEDURE

Assembly of the wind turbine at the site will be conducted in the following stages:

(a) Site preparation and installation of foundation.

(b) Erection of tower.

(c) Lift and installation of major nacelle sub-assemblies.

(d) Installation of control enclosure and utility connection.

(e) Installation of blades.

The nacelle equipment will arrive at the site after erection of the tower is complete. By staging the truck shipments in Boone, each sub-assembly will arrive at the top of Howard Knob in the sequence required for erection. Erection and installation procedures are summarized in Table II. Note that the weight to be lifted may differ slightly from the weight in the shipping configuration.

The oil cooler, removed for shipment, will be installed under the bedplate while the bedplate is suspended or supported at ground level. The yaw slip ring may also be installed at this time, or with equal facility, can be installed after the bedplate has been mounted on the yaw structure. Installation of the rotor assembly with the low speed shaft attached requires removal of the cover plate on the bedplate. The cover plate will then be lifted and installed as a separate unit.

TABLE I

WEIGHT AND SIZE SUMMARY

	WEIGHT (TONS)	SIZE	NOTES
1. Bedplate	40	L=395" (32' - 11") W=150" (12' - 6") H=120" (10')	
2. Rotor Assy	40	L=270" (22' - 6") W=100" (8' - 4") H=130" (10' - 10")	1
3. Gearbox	31.5	L=119" (9' - 11") W= 92" (7' - 8") H=120" (10')	2
4. Generator	7.2	L=103" (8' - 7") W= 62" (5' - 2") H= 72" (5')	3
5. Nacelle Multiplexer Unit (NMU)	1400 lb.	L= 60" (5') W= 36" (3') H= 92" (7' - 8")	
6. Yaw Section	25	L=177" (14' - 9") W=170" (14' - 2") H= 91" (7' - 7")	
7. Slipring	600 lb.	L= 90" (7' - 6") W= 40" (3' - 4") H= 30" (2' - 6")	4
8. Pintle	20.5	L=186" (15' - 6") W=186" (15' - 6") H=129" (10' - 9")	

TABLE I

WEIGHT AND SIZE SUMMARY

(CONTINUED)

	WEIGHT (TONS)	SIZE	NOTES
9. Fairing	6500 lb	L=400" (33' - 4") W=158" (13' - 2") H=131" (10' - 11")	5
10. Ground	17	L=337" (28' - 1") H=128" (10' - 8") W=127" (10' - 7")	
11. Miscellaneous #1	2000 lb	L=136" (11' - 4") W=120" (10') H= 24" (2')	6
12. Miscellaneous #2	1800 lb	L=136" (11' - 4") W= 96" (8') H= 48" (4')	7

NOTES TO TABLE I

1 Includes rotor hub, adapter structure, low speed shaft and flexible couplings, pitch mechanism, fixture.

2 Includes shipping stand, low-speed rigid coupling, high-speed shaft assembly.

3 Includes cooling air manifold, resistor, but not a shipping crate.

4 Includes shipping crate.

5 Dimensions are for assembled fairing.

6 Includes oil cooler, bedplate crosswalk, braces, yaw slip ring shield, generator exhaust duct, rotor slip ring, and structural interface bolts.

7 Includes pintle decking, ladders, wireways, and junction boxes.

TABLE II

PROPOSED ASSEMBLY SEQUENCE

ITEM NO.	ASSEMBLY	WEIGHT LIFTED (LB)	ASSEMBLY AIDS REQUIRED	ACCESS REQUIRED	ATTACHMENT REQUIRED BEFORE REMOVING CRANE	INTERFACE CONNECTION AND INSTALLATIONS AFTER CRANE REMOVAL
1	Pintle(upper section of tower)	41,000	• Alignment pins	• Top of tower legs for access to bolts	24-1¼ dia. bolts	• Torque bolts • Install floor • Install hand rails • Install personnel lift
2	Yaw Structure	47,800	• Temporary alignment pins • Portable pump to pressurize yaw brakes	• Temporary stand on upper level of tower for access to bolts	32-1¼ dia. bolts	• Retorque all bolts • Install lower ladder
3	Bedplate	87,400	• Temporary alignment pins • Guide to protect yaw motors	• Through yaw structure for access to interior bolts • Temporary scaffold from upper level of tower for access to exterior bolts	26-1¼ dia. bolts (interior)	• Install exterior bolts (26) • Retorque all bolts • Remove guide fixtures • Install upper ladder • Install slip ring • Connect cables to slip ring • Hydraulic lines (10)

5	Gearbox	60,700	• Tool to rotate gearbox high speed shaft for alignment • Alignment dowels	• Top of bedplate through yaw structure (note: safety railings in place)	8-1½ bolts	• Retorque all bolts • Attach rigid coupling to shaft & remove shaft support • Pull rotor cables through shaft & gearbox & connect to slip ring • Install flex coupling between gearbox & lube pump • Remove safety railing
	Generator	14,400	• Tool to rotate generator • Alignment dowels	• Top of bedplate	4-1¼ dia. bolts	• Attach rigid coupling to high speed shaft • Remove shaft support
7	Fairing	5,500	• Alignment lips on bedplate	• Nacelle exterior (blind fasteners)	• Approx. 80 bolts	• Install weather seals at rotor adapter • Install generator cooling duct
8	Slip Ring	520	• Sling through hole in bedplate	• Interior of yaw structure		• Connect cables
9	Blades (2)	20,000	• Temporary alignment pins	• Rotor hub interior through access opening	56-1¼ bolts	• Torque bolts • Connect instrumentation wiring

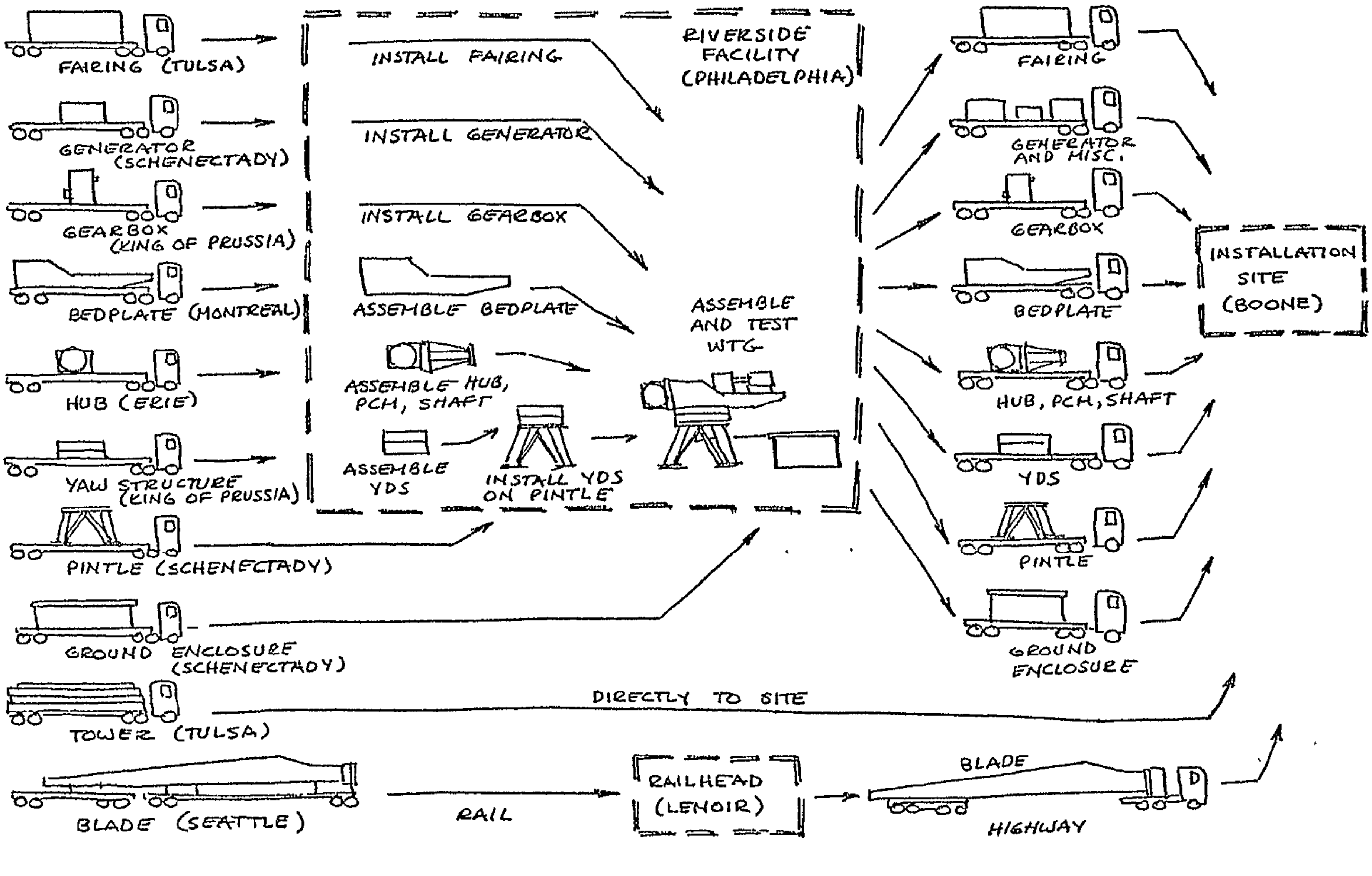


Figure 1 Shipping Plan

APPENDIX D

METAL BLADE STUDY

Abstract

This appendix is an edited version of the Task Completion Report submitted by Lockheed - California for the Mod 1 Metal Blade Study Program. The editing has deleted reference data, supplementary notes, and presentation material that were considered superfluous. A complete set of performance data has been included only for the selected blade configuration (5.X. RF1).

TASK COMPLETION REPORT

MOD. 1 METAL BLADE PROGRAM

INTRODUCTION

Effort on this program was initiated in March 21, 1977. The objective of the contract at that time was preliminary design, detail design and fabrication of two back-up metal blades. On April 29, 1977, the program was redirected to eliminate the detail design and fabrication of blades and perform a limited parametric analysis to determine the design criteria which would lead to a minimum cost blade for a 1500 kW WTG. Thus, on April 29, the objective was to perform a parametric analysis as shown in Figure 1.1. The results were to be reviewed on May 23rd and a final configuration selected from the parametric candidates.

At the parametric analysis review with GE on May 23, 1977, it was decided to determine the preliminary design characteristics of a minimum cost 110 foot metal blade which would produce 2000 kW shaft power at a rated windspeed of 22 mph (30 ft. height).

On June 2, 1977, GE was notified in telephone conference that the requirement of 35 rpm design rotational speed creates conditions which result in a heavy and expensive blade. Because, the rotational speed cannot be reduced, it was decided to revert back to a shorter blade which would produce 2000 kW shaft power at a rated windspeed higher than 22 mph.

The format of this report is as follows:

1. Introduction
2. Selection of Blade Geometry
3. Load Analysis
4. Structures
5. Dynamic Analysis
6. Producibility
7. General Recommendations
8. Blade Data
9. Presentation Material (DELETED)

OBJECTIVES

- HELP G.E. ESTABLISH DESIGN REQUIREMENTS FOR MOD 1 METAL BLADES

- EVALUATE EFFECTS OF RPM, AIRFOIL SECTION, TWIST, AND LENGTH ON FABRICATION COST OF MOD 1 METAL BLADES

- METAL BLADE REQUIREMENTS MUST MINIMIZE MODIFICATIONS TO PRESENT DESIGN OF THE REST OF THE WTG SYSTEM

CAVEATS

- PARAMETRIC STUDY - NOT A DESIGN
- ALL COSTS ARE ENGINEERING PARAMETRIC ESTIMATES AND DO NOT INCLUDE SOME ITEMS COMMON TO ALL CONFIGURATIONS.
- BLADE DESIGN REQUIREMENTS TRADE-OFFS ARE SUB-OPTIMIZED FOR BLADE ONLY

FIGURE 1.1

VARIABLE DESIGN PARAMETERS

Parametric Configuration	Airfoil	RPM		Blade Twist		Nominal Blade Length		
		31.5	35	Opt.	None	95'	100'	104'
1. (2.6.1,2)	44XX		X	X				X
2. (2.6.3)	44XX	X		X				X
3. (5.X.3)	LS-1		X	X			X	
4. (5.X.4)	LS-1	X		X		X		
5. (5.X.6)	LS-1	X			X	X		

CUT-OUT WINDSPEED = 35 mph AT 9 METER HEIGHT

SECTION 2

2. Selection of Blade Geometry

Optimizing the torque force along the major working portions of the blade (outboard of 40% of radius) with consideration of realistic constraints may be accomplished in the following general manner. Achieving low induced drag losses and high lift-to-drag ratios (low profile drag losses) at structurally acceptable low thickness ratios, 21 to 12%, results in a planform taper ratio of approximately 3 to 1 for the chords measured at the 40% and the 100% radius stations, respectively. Selection of the airfoil series determines the C_l's for high lift-to-drag ratios and the margins between high operating C_l's and the stall C_l's. This aspect of the design selection will be addressed later in this section.

Structurally acceptable narrow chord blades, in combination with the initial matrix of rpm's and diameters, make it feasible to achieve near maximum efficiency at rated and mean wind speed regions (top speed ratio 8 to 11) with blades having a two segment linear twist (approximately -8° from centerline to 40% radius; approximately -5°, 40% to 100% radius) and a two segment thickness ratio distribution with a break in the thickness distribution at 40% radius stations. Metal blade fabrication does not require straight line spanwise elements, but special spanwise curvatures are not especially warranted for performance.

Two airfoil families have been considered in the preliminary study, the NACA 44XX airfoil series, and the NASA LS (1) -04XX series. Figure 2.1 shows a comparison of the two airfoil families in the thickness ratio range of 12 to 21%. The particular advantages of the LS (1) -04XX series occur in the region nearer the lower thickness ratios of this range, which also exhibits higher design C_l capability with roughness, Figure 2.2.

Candidate configurations with both airfoil families have been studied with allowance for a manufacturing joint at the break in linearity of twist and thickness ratios. In addition to a preliminary requirement of 1670 kW shaft power a cut-in power of 100 to 150 kW was considered as a reasonable power level. A subsequent lower level of 90 kilowatts was still considered reasonable.

Configuration 2.6.1, (candidate 1 in Figure 1.1) was taken as a first configuration which was known to meet the requirements and was shown to be overdesigned for the rated power when the 44XX airfoil is used in place of the 230XX airfoil. Configuration 2.6.2 is an alternate version with the manufacturing points and breaks in linearity of twist and thickness ratio moved to 48% for ease in manufacturing with this blade length.

Configuration 2.6.3, (candidate 2 in Figure 1.1) is the same as configuration 2.6.1, but operating at 31.5 instead of 35 RPM. This configuration could be considered representative of the 44XX airfoil series blades.

The remaining effort has been directed to the LS (1) -04XX's airfoils which permits smaller blades for the rated and cut-in powers. At rated power, the LS (1) -04XX series allows higher operating C_l's, higher $L/_D$'s and higher margins to stall C_l's. Configuration 5.X.3, (candidate 3 in Figure 1.1) provides the smallest blade to meet the performance and is designed for 35 RPM. An alternate configuration, 5.X.9, with a higher thickness ratio at 10% radius, .40 versus .294, was considered for meeting frequencies and structural loads, but did not meet rated power at rated wind speed. Configuration 5.X.4, (candidate 4 in Figure 1.1) was the next smallest blade that could meet the rated power but at the lower RPM of 31.5. Configuration 5.X.6, (candidate 5, Figure 1.1) shows the increase in diameter required to meet rated power of 1670 kW at 31.5 RPM of the blade has zero twist.

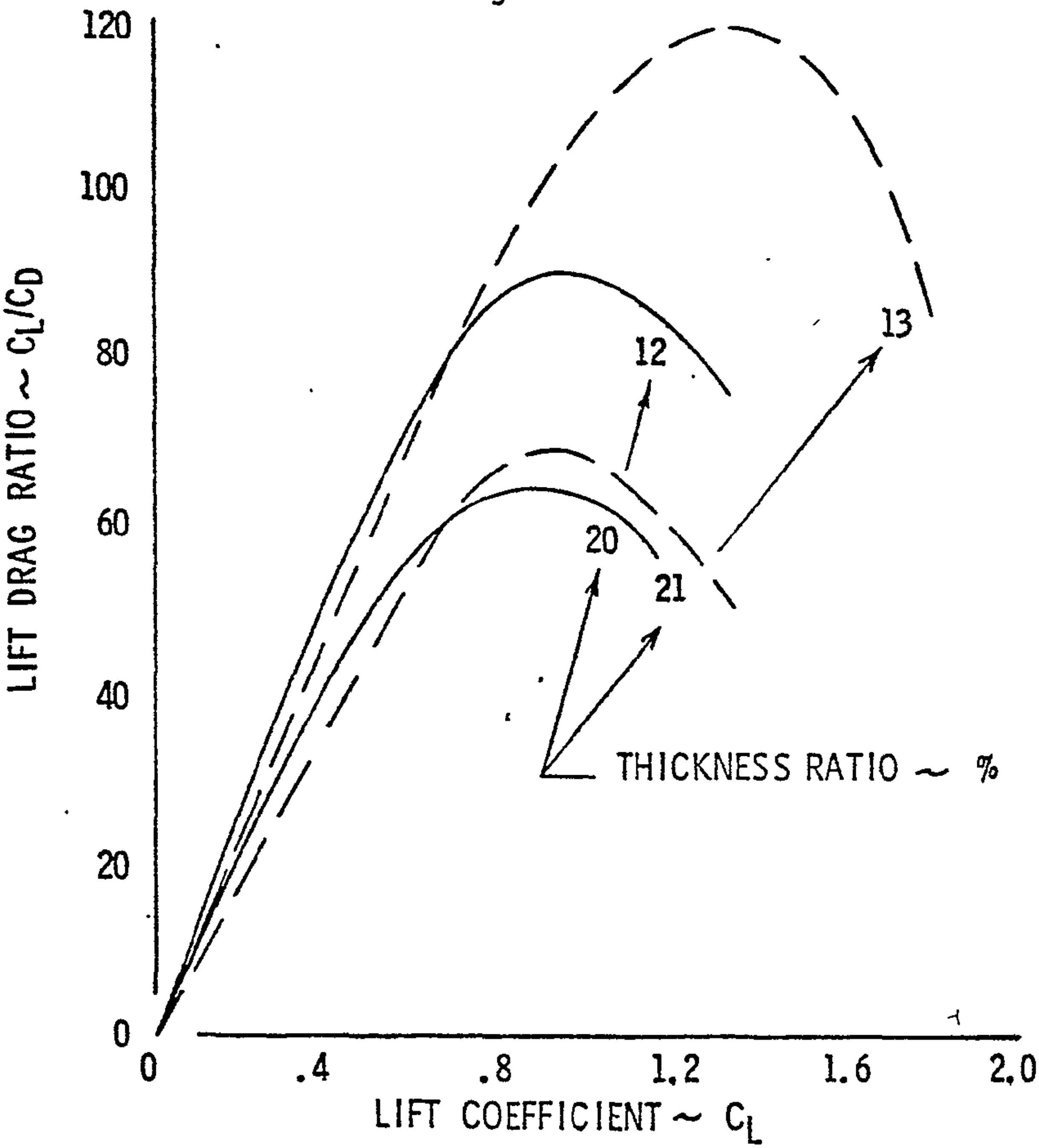
Redirection in target design resulted in a requirement of 2000 kW shaft power at rated speed with a blade length of 110 ft. at 35 RPM. Structural examination of this configuration, 5.X.F13, showed that attempts to refine this design resulted in a continually increasing weight, and operation at C_l's which did not really take advantage of LS (1) -04XX airfoil series characteristics.

A second redirection in design resulted, allowing the rated wind speed to be increased from 22 to 23.74 MPH. This design includes the thickness ratio distribution of 5.X.9 and the slighly increased chords of 5.X.RF1, but at a blade length of 99.6 ft. This configuration is the smallest blade which meets the performance, frequencies and loads. It takes better advantage of the performance characteristics of the LS (1) -04XX airfoil series.

9

FIG. 2.1 COMPARISON OF AIRFOIL PROPERTIES

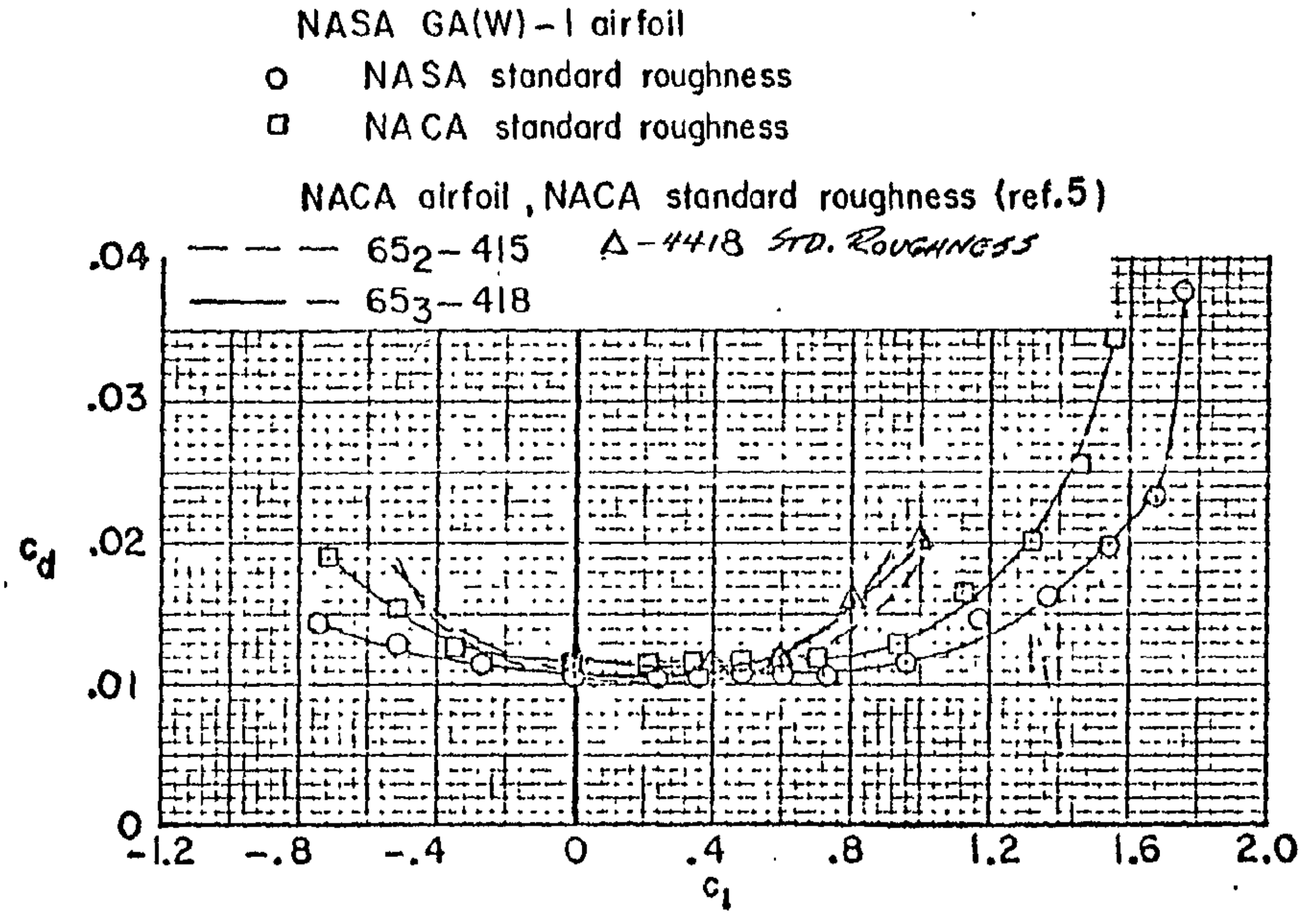
Airfoil	Surface	Approx. RN
NACA 44XX(1)	Mid Smooth & NACA Std Rough	6×10^6
NASA LS(1)04XX(2)	NASA Std Rough	6×10^6



(1) Processed from NACA Rpt 824

(2) Data Source: NASA TM X-72843

COMPARISON OF SECTION CHARACTERISTICS



(b) Variation of c_d with c_l.

FIGURE 2.2

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CONFIGURATION	AIRFOIL SERIES	TYPE OF TWIST	BLADE LENGTH, FT	SOLIDITY PER BLADE: GEOMETRIC	SOLIDITY PER BLADE: THRUST BASED	SOLIDITY PER BLADE: POWER BASED	ACTIVITY FACTOR, PER BLADE	RPM	30 FT HT WIND SPEEDS, MPH: RATED	30 FT HT WIND SPEEDS, MPH: CUT IN	SHAFT POWER, KW AT 140 FT HT WIND SPEEDS: RATED	SHAFT POWER, KW AT 140 FT HT WIND SPEEDS: CUT IN
SUMMARY GEOMETRY INFORMATION								SUMMARY OPERATING CONDITIONS AND PERFORMANCE				
① 2.6.1,	Ⓐ NACA 44XX	2 SEGMENT LINEAR	103.75	.02071	.01515	.01380	33.88	35.	22.0	11.0	1752	99
2.6.2			103.75	.02071	.01515	.01380	33.88	35.			1714	92
② 2.6.3			103.75	.02071	.01515	.01380	33.88	31.5			1685	145
③ 5.X.3	Ⓑ NASA LS(1)-04XX		99.6	.01825	.01335	.01216	29.84	35.			1670	111
5.X.9			99.6	.01825	.01335	.01216	29.84	35.			1625	107
④ 5.X.4			99.6	.02157	.01578	.01438	35.29	31.5			1670	132
⑤ 5.X.6		ZERO	102.4	.02098	.01535	.01399	34.32	31.5			1671	110
5.X.F13		1 SEGMENT LINEAR	110.0	.01696	.01242	.01131	27.76	35.			2010	95
5.X.RE1		2 SEGMENT LINEAR	99.6	.01873	.01372	.01249	30.66	35.	23.74		2016	103.6

WIND TURBINE ROTOR, 2 METAL BLADES,

12 DEG CONE ANGLE, AIRFOIL CONTOUR: 10. TO 100% BLADE LENGTH

NOTES ON SURFACE CONDITION ASSUMPTIONS:

Ⓐ NACA 44XX AIRFOIL SERIES: MID SMOOTH & NACA STD ROUGHNESS @ $RN = 6 \times 10^6$

Ⓑ NACA LS(1)-04XX AIRFOIL SERIES: NASA ROUGHNESS @ $RN = 6 \times 10^6$

FIGURE 2.3 GEOMETRY & PERFORMANCE SUMMARY.

3. LOADS ANALYSIS

An initial loads analysis was performed for the GE fiberglass blade so that a common point of departure would be established and that the equivalence of frequency calculating methods at GE and Lockheed would be verified. The results of this analysis proved the equivalence of the two methods.

The loads analysis consists of the following:

- Frequency spectrum
- Case descriptions
- Blade tip deflection
- Blade flapwise and chordwise bending moment distributions for each design case as specified in the statement of work

Under the statement of work, the frequency placement requirement was a major structural design parameter. Those configurations which did not meet the frequency requirement were eliminated and, in most cases, loads were not calculated. Table 3.1 is a summary of the configurations analyzed and the frequency placement of the first flap and first in-plane modes. Loads analyses were performed on the following configurations: 2.6.1, 2.6.3, 5.X.9 (alternate configuration to 5.X.3 to try and meet the frequency requirement), and 5.X.RF1.

The loads data for each configuration analyzed are included in this report under the particular configuration number.

DESIGN LOADS

METHOD OF ANALYSIS

LOCKHEED COMPUTER PROGRAM "WINTUR"

A FULLY COUPLED FLAPWISE-CHORDWISE BLADE DYNAMIC RESPONSE ANALYSIS METHOD ~ INCLUDES BLADE DYNAMIC TORSION ANALYSIS

A SUB-SET COMPUTES COUPLED FLAPWISE-CHORDWISE NATURAL FREQUENCIES

APPLICATION AND CREDIBILITY

APPLIED TO ALL HELICOPTER ROTORS DEVELOPED AND BUILT BY LOCKHEED

RESULTS COMPARED IN A HELICOPTER INDUSTRY-WIDE LOADS PREDICATION COMPARISON STUDY AT 1974 AHS/NASA-AMES ROTOR SPECIALIST'S MEETING

MOD-0 100 kW METAL BLADES FOR NASA-LEWIS

MOD-0 CORRELATION OF ANALYTICAL AND ACTUAL LOADS DATA FOR NASA-LEWIS

MOD-0A 200 kW METAL BLADES FOR LAS/NASA-LEWIS

WIND ENVIRONMENT

WIND SPEEDS

SPECIFIED BY THE STATEMENT OF WORK (SOW)

WIND SHEAR PROFILE

$V = V_o \left(\frac{H}{H_o}\right)^n$; AS SPECIFIED BY SOW

WHERE: $n = 0.167$

$H_o = 30$ FEET

V_o = SPECIFIED WIND SPEED AT $H = 30$ FEET

TOWER SHADOW

30° AZIMUTHAL SECTOR BASED ON A 29% TOWER SHADOW

FIGURE 3.1

MOD 1 FREQUENCY

	R ins	CHORD @ 10%	t/c 10%	t/c 40%	t/c 100%	RPM	TWIST	FREQUENCY ~P IF	IT
2.6.1	1245	129.6	.294	.216	.120	35.	OPT.	2.5	4.4
2.6.2	1245	129.73	.385	.254	.120	35.	OPT.		
2.6.3	1245	129.6	.294	.216	.120	31.5	OPT.	2.7	4.9
5.X.3	1195.2	109.68	.294	.216	.120	35.0	OPT.	2.3	4.1
5.X.4	1195.2	129.6	.294	.216	.120	31.5	OPT.	2.6	4.8
5.X.5	1245	129.6	.294	.216	.120	31.5	OPT.	—	—
5.X.6	1228.8	129.6	.294	.216	.120	31.5	0	—	—
5.X.7	1195.2	109.68	.36	.239	.120	35.	OPT.	—	—
5.X.8	1195.2	109.68	.36	.239	.120	35.	OPT	—	—
5.X.9	1195.2	109.68	.40	.216	.120	35.	OPT	2.6	4.6
5.X.F13	1320	112.44	.33	.216	.120	35.	OPT	2.8	4.3
5.X.RF1	1195.2	112.44	.40	.216	.120	35.	OPT	2.7	4.8

BLADE LOADS SUMMARY/COMPARISON @ STATION r/R = 0.[illegible]

GE SOW CASE NO.	LOCKHEED CASE NO	WIND SPEED MPH*	CONFIGURATIONS: 2.6.1 RPM = 35 TWIST = OPT AIRFOIL = 44XX		2.6.3 RPM = 31.5 TWIST = OPT AIRFOIL = 44XX		RPM = TWIST = AIRFOIL =		2.6.1 RPM = 35 TWIST = OPT AIRFOIL = 44XX		2.6.3 RPM = 31.5 TWIST = OPT AIRFOIL = 44XX		RPM = TWIST = AIRFOIL =	
			FLAP BENDING ~ 10^6 IN-LB						CHORD BENDING ~ 10^6 IN-LB					
			MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC
2.1.2.1	1A	15	-5.70	±1.10					0.20	±1.60				
NORMAL	1B	20	-3.20	±1.35	-2.30	±1.40			0.45	±1.60	0.30	±2.05		
OPERATION	1C	25	-5.00	±1.85					-0.20	±1.60				
	1D	30	-6.20	±2.20					-0.75	±1.60				
	1E	33	-6.75	±2.35					-1.05	±1.75				
	1F	35	-7.00	±2.60	-5.60	±2.60			-1.25	±1.75	-1.00	±2.20		
2.1.2.2 SEVERE	2D	35 +15	-2.25	±3.90	-1.10	±3.85			-1.85	±1.80	-1.90	±2.70		
GUSTING	2E	35 -15	-11.00	±1.30					-2.05	±1.75				
HURRICANE WIND	5B	120	9.50	±0.2[illegible]					0.20	±1.65				
FULL FEATHER	3A	18	-14.30	±1.80	-11.30	±1.60			-10.50	±0.30	-8.50	±2.10		

* ALL WIND SPEEDS ARE AT 30 FOOT ELEVATION.

BLADE LOADS SUMMARY/COMPARISON @ STATION r/R = 0.4[illegible]

GE SOW CASE NO.	LOCKHEED CASE NO	WIND SPEED MPH*	CONFIGURATIONS: 2.6.1, RPM = 35, TWIST = OPT, AIRFOIL = 44XX		5.X.9, RPM = 35, TWIST = OPT, AIRFOIL = LS-1		RPM =, TWIST =, AIRFOIL =		2.6.1, RPM = 35, TWIST = OPT, AIRFOIL = 44XX		5.X.9, RPM = 35, TWIST = OPT, AIRFOIL = LS-1		RPM =, TWIST =, AIRFOIL =	
			FLAP BENDING ~ 10^6 IN-LB						CHORD BENDING ~ 10^6 IN-LB.					
			MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC
2.1.2.1	1A	15	-5.70	±1.10	-3.03	±0.82			0.20	±1.60	0.11	±1.08		
NORMAL	1B	20	-3.20	±1.35	-1.30	±1.01			0.45	±1.60	0.04	±1.19		
OPERATION	1C	25	-5.00	±1.85	-2.11	±1.41			-0.20	±1.60	-0.29	±1.15		
	1D	30	-6.20	±2.20	-3.47	±1.71			-0.75	±1.60	-0.64	±1.14		
	1E	33	-6.75	±2.35	-4.10	±1.90			-1.05	±1.75	-0.85	±1.25		
	1F	35	-7.00	±2.60	-4.53	±2.01			-1.25	±1.75	-0.96	±1.25		
2.1.2.2 SEVERE	2D	35 +15	-2.25	±3.90	-0.30	±3.25			-1.85	±1.80	-1.70	±1.60		
GUSTING	2E	35 -15	-11.00	±1.30	-8.00	±1.00			-2.05	±1.75	-1.60	±1.30		
HURRICANE WIND	5B	120	9.50	±0.35	7.50	±0.20			0.20	±1.65	0.11	±1.10		
FULL FEATHER	3A	18	-14.30	±1.80	-10.30	±1.10			-10.50	±0.30	-7.30	±0.30		

* ALL WIND SPEEDS ARE AT 30 FOOT ELEVATION.

D-36

BLADE LOADS SUMMARY/COMPARISON @ STATION r/R = 0.6

GE SOW CASE NO.	LOCKHEED CASE NO	WIND SPEED MPH*	FLAP BENDING ~ 10^6 IN-LB: 5.X.9, RPM = 35, TWIST = OPT, AIRFOIL = LS-1		FLAP BENDING: 5.X.4, RPM = 31.5, TWIST = OPT, AIRFOIL = LS-1		FLAP BENDING: RPM = , TWIST = , AIRFOIL =		CHORD BENDING ~ 10^6 IN-LB: 5.X.9, RPM = 35, TWIST = OPT, AIRFOIL = LS-1		CHORD BENDING: 5.X.4, RPM = 31.5, TWIST = OPT, AIRFOIL = LS-1		CHORD BENDING: RPM = , TWIST = , AIRFOIL =	
			MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC
2.1.2.1	1A	15	-3.03	±0.82	-5.95	±1.19			0.11	±1.08	-0.29	±1.88		
NORMAL	1B	20	-1.30	±1.01	-4.62	±1.47			0.04	±1.19	-0.22	±1.98		
OPERATION	1C	25	-2.11	±1.41	-4.50	±1.85			-0.29	±1.15	-0.58	±2.04		
	1D	30	-3.49	±1.71	-5.81	±2.31			-0.64	±1.14	-1.20	±2.11		
	1E	33	-4.10	±1.90	-6.32	±2.53			-0.85	±1.25	-1.51	±2.22		
	1F	35	-4.33	±2.01	-6.61	±2.68			-0.96	±1.25	-1.72	±2.31		
2.1.2.2 SEVERE	2D	35 +15	-0.30	±3.25	-1.60	±4.06			-1.70	±1.60	-2.48	±3.16		
GUSTING	2E	35 -15	-8.00	±1.00	-9.95	±1.37			-1.60	±1.30	-2.16	±2.20		
HURRICANE WIND	5B	120	-7.50	±0.20	-8.88	±0.49			0.11	±1.10	0.44	±1.89		
FULL FEATHER	3A	18	-10.30	±1.10	-10.05	±2.04			-7.30	±0.30	-9.60	±0.75		

* ALL WIND SPEEDS ARE AT 30 FOOT ELEVATION.

BLADE LOADS SUMMARY/COMPARISON @ STATION r/R = 0.0

GE SOW CASE NO.	LOCKHEED CASE NO	WIND SPEED MPH*	2.6.1 RPM = 35. TWIST = OPT AIRFOIL = 44XX		2.6.3 RPM = 31.5 TWIST = OPT AIRFOIL = 44XX		RPM = TWIST = AIRFOIL =		5.X.9 RPM = 35 TWIST = OPT AIRFOIL = LS-1		5.X.4 RPM = 31.5 TWIST = OPT AIRFOIL = LS-1		RPM = TWIST = AIRFOIL =	
			PITCH CONTROL MECHANISM LOADS / BLADE ~ 10^3 IN-LB											
			MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC	MEAN	CYCLIC
2.1.2.1	1A	15	255.	±106.					105.	±44.	125.	±95.		
NORMAL	1B	20	225.	±91.	191.	±83.			88.	±34.	106.	±87.		
OPERATION	1C	25	274.	±93.					106.	±37.	117.	±86.		
	1D	30	329.	±93.					135.	±44.	156.	±101.		
	1E	33	356.	±100.					148.	±51.	176.	±112.		
	1F	35	369.	±107.					158.	±56.	189.	±121.		
2.1.2.2 SEVERE	2D	35 +15	262.	±116.	197.	±108			67.	±88.	66.	±120		
GUSTING	2E	35 -15	527.	±144.					297.	±83.	328.	±142		
HURRICANE WIND	5B	120	-1330.	±62.					-912.	±52.	-1269.	±76.		
FULL FEATHER	3A	18	2268.	±316.	1696.	±449			1366.	±177.	1406.	±272.		

* ALL WIND SPEEDS ARE AT 30 FOOT ELEVATION.

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4. STRUCTURES

4.1 Structural Considerations

The structure is sized to achieve a target life of 30 years without appreciable fatigue damamge, cracks $<$.25".

At such time as a crack appears, from whatever source (material flaw, corrosion, stress corrosion, fatigue), the crack propagation rate must be low enough to maintain adequate residual strength capability between inspection periods.

The above fundamental criteria determines the selection of the structural configuration, material and the sizes of the structural members.

4.2 Fatigue and Fail Safe Criteria

The fatigue criteria is defined in the GE work statement: paragraph 1.4, 2.1.2.1 and 2.1.2.4. It requires the determination of the average mean and cyclic loading for wind speeds up to the cut off wind speed (35 MPH). The cyclic loading is established at the predominant frequency which is assumed at one per rev.

The number of cycles which occur over a 30 year period distributed in accordance with the velocity duration curve (work statement Exhibit B) is shown below in Figure 4.2.1. It should be noted that any cycles which number above 10^6 require the allowable to be below the endurance limit. On this basis, operation at the higher speeds sizes the blade.

The variation of cyclic loading at a specified wind speed is required to be log normal distributed having a mean cyclic load equal to the calculated value. The flatwise and edgewise cyclic loads are required to have a standard deviation such that the 3σ loads are 1.85 and 1.2 times the mean cyclic load, respectively. The cummulative damage shall be determined using the dominant frequency of cyclic loading.

The above criteria for a log normal distribution applies factors (1.85 and 1.2) to cover loadings other than those occurring from steady wind conditions. It is assumed these factors take care of transients such as gusts, cross winds and yaw motions.

The ratio of the transient load to steady load levels, will vary with the absolute value of the steady wind speed and the specific dynamic characteristics of the generator system. The flatwise bending 3σ value was the overriding factor which established the member sizing of the blade main structural box section.

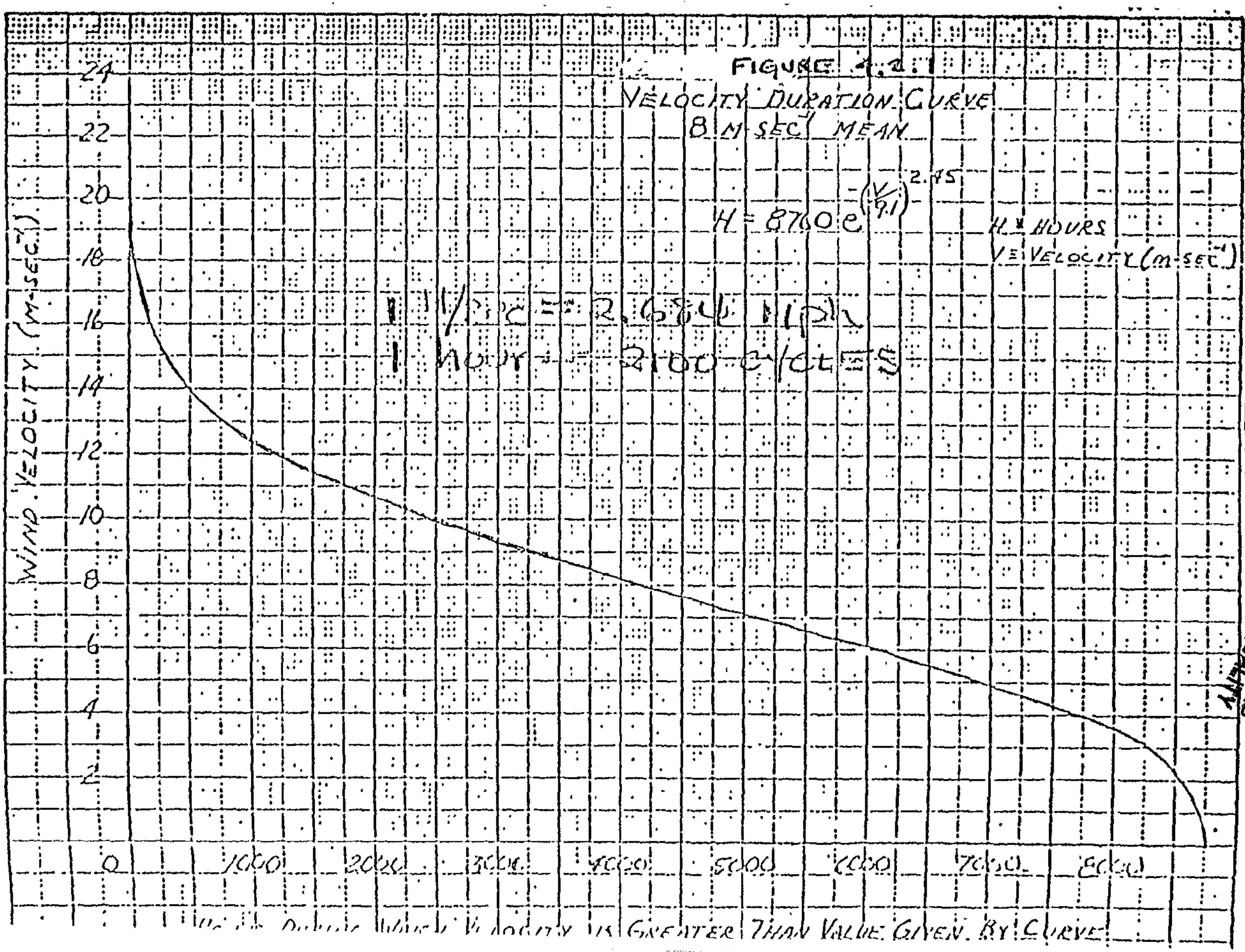

EXHIBIT 3

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The basic log normal distribution formulae are shown in Figure 4.2.2. An example of distribution used in the blade analyses in terms of mean cyclic loading and number of cycles is shown in Figure 4.2.3.

In order to determine fatigue damage using the Palmgren-Miner theory of linear cumulative damage the allowable fatigue stresses (S-N curve) must be determined and a realistic level of hardware fatigue quality must be assumed.

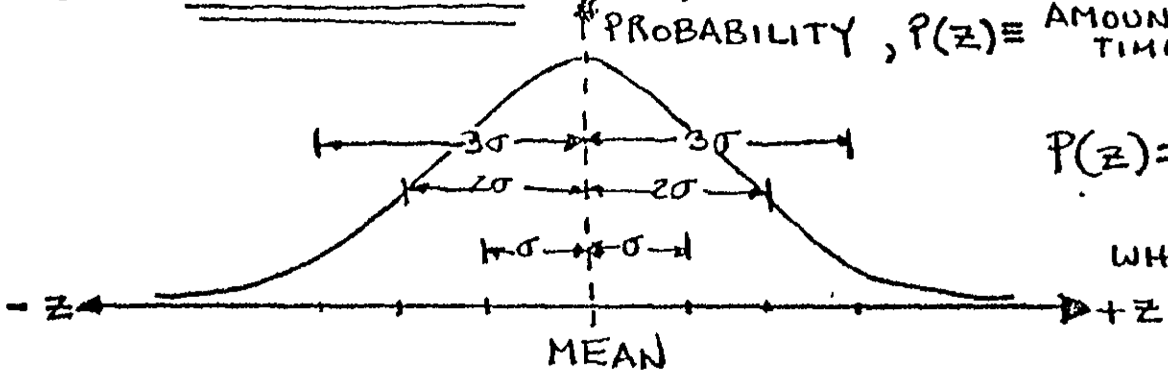
For the sizing of the members required before mass and stiffness properties can be calculated, a fatigue quality index of 5 was assumed. To achieve this value in the actual hardware will require careful attention to the detail design of joints and fittings. The actual fatigue quality index achieved in the hardware must be substantiated by tests covering the cyclic range required 10^7.

A typical allowable curve for the aluminum alloys used in the sizing of the structure is shown below in Figure 4.2.4. The values shown are for a 50% probability. These values must be reduced to arrive at the allowables for sizing of the blade, approximately 20% in the high cycle range.

Fail Safe requirements are established to give a damage tolerance level which would result in a capability to carry maximum design loads after the failure of a single number.

FIGURE 4.2

- STATISTICAL DISTRIBUTION OF CYLIC LOADING USING A CUMULATIVE LOG NORMAL PROBABILITY CURVE



*PROBABILITY, P(z) ≡ AMOUNT OF TOTAL TIME AT PARAMETER z

$$P(z) = \frac{1}{\sqrt{2\pi}} e^{-\frac{z^2}{2}}$$

WHERE $z = \frac{x}{\sigma}$

$x = \log_{10}(\text{LOAD})$

$$\sigma = \text{STANDARD DEVIATION} = \sqrt{\frac{\Sigma(x-\bar{x})^2}{n-1}}$$

$3\sigma_{LOG} = X$ MEANS THAT AT +3 STANDARD DEVIATIONS FROM THE MEAN VALUE, THE LOAD WILL BE X TIMES THE MAGNITUDE OF THE MEAN LOAD. THE SAME DISTRIBUTION WILL YIELD A LOAD MAGNITUDE OF (1/X) OF THE MEAN LOAD AT -3 STANDARD DEVIATIONS FROM THE MEAN

FOR THE CUMULATIVE FREQUENCY, $f(x)$

$f(x)$ = THE AREA UNDER THE ABOVE CURVE

AT	A	LOAD	LEVEL	3σ	BELOW	THE	MEAN,	THE	CUMULATIVE	FREQUENCY	IS 0.13%
"	"	"	"	2σ	BELOW	"	"	"	"	"	" 2.28%
"	"	"	"	σ	BELOW	"	"	"	"	"	" 15.87%
"	"	"	"	0	——	"	"	"	"	"	" 50%
"	"	"	"	σ	ABOVE	"	"	"	"	"	" 84.13%
"	"	"	"	2σ	ABOVE	"	"	"	"	"	" 97.72%
"	"	"	"	3σ	ABOVE	"	"	"	"	"	" 99.87%

*NOTE: FOR EACH WINDSPEED THERE EXISTS A SEPARATE DISTRIBUTION BASED ON VELOCITY DURATION CURVE (FIG.   ) AND THE CALCULATED MEAN CYCLIC LOAD

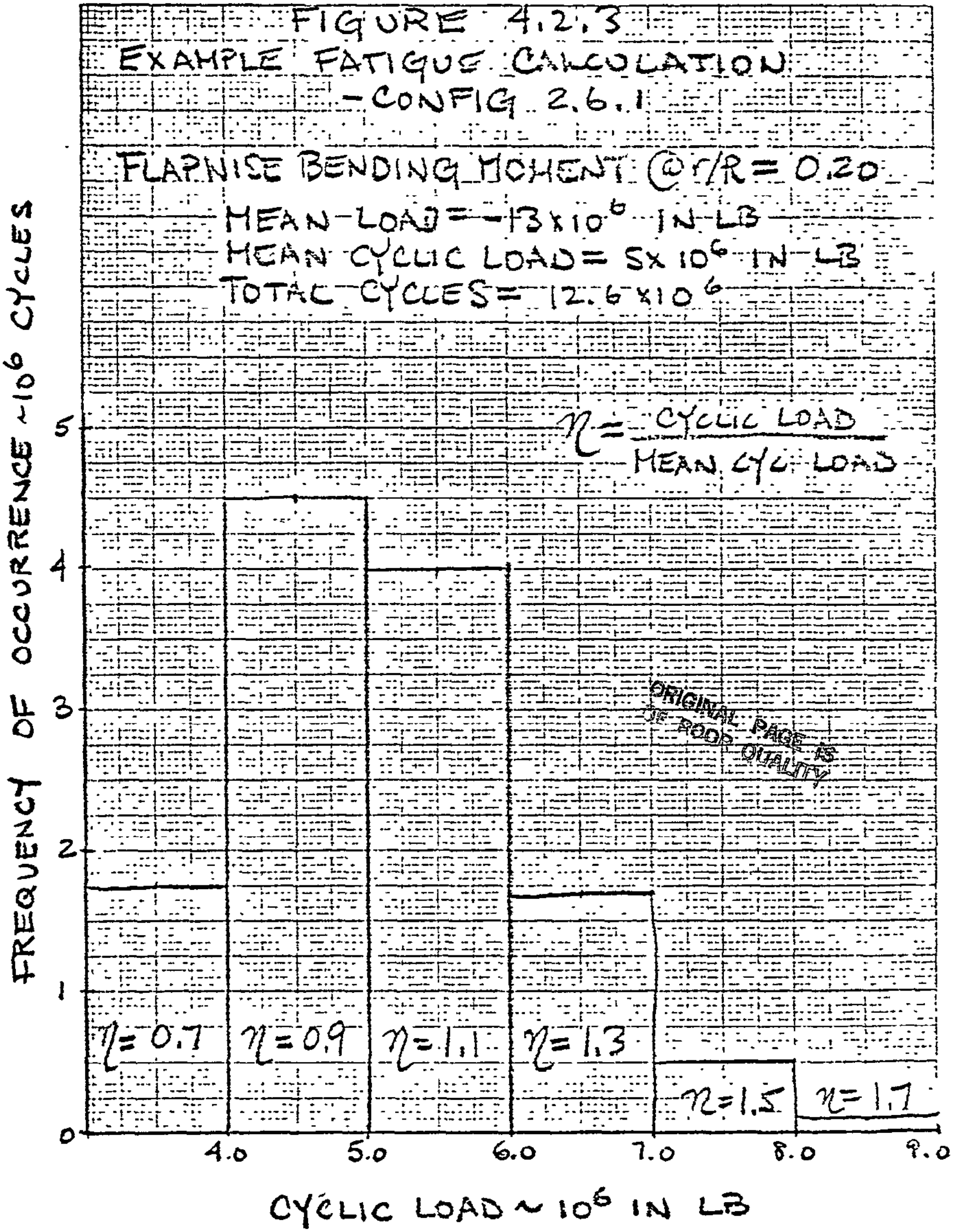


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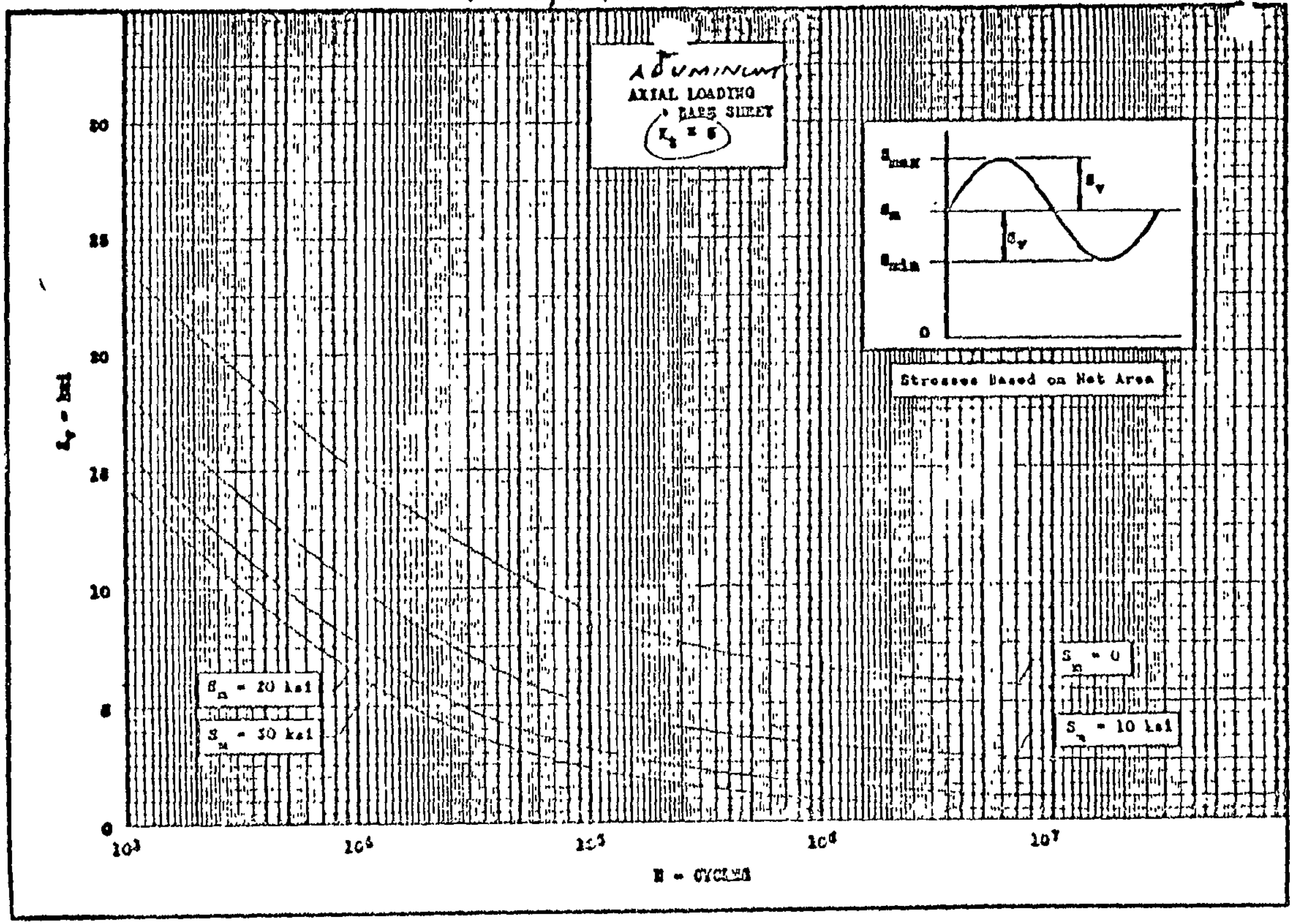
FIGURE 1.2.1



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7

4.3 Structural description

The basic blade primary structural section is composed of an aluminum, mechanically attached trapezoidal shaped box composed of heavy skins and corner caps.

To this basic section is attached the leading edge nose skin and trailing edge stringer stiffened skin to form the basic airfoil. This structural configuration was sized to carry the maximum design load, and comply with fatigue and frequency requirements. The basic geometry, weight, structural properties, mass and stiffness properties, resulting from the sizing exercise are given in Figures 4.3.1 thru 4.3.8.

The blade has been designed to carry all the design loads specified in the work statement and shown in the loads section of this report.

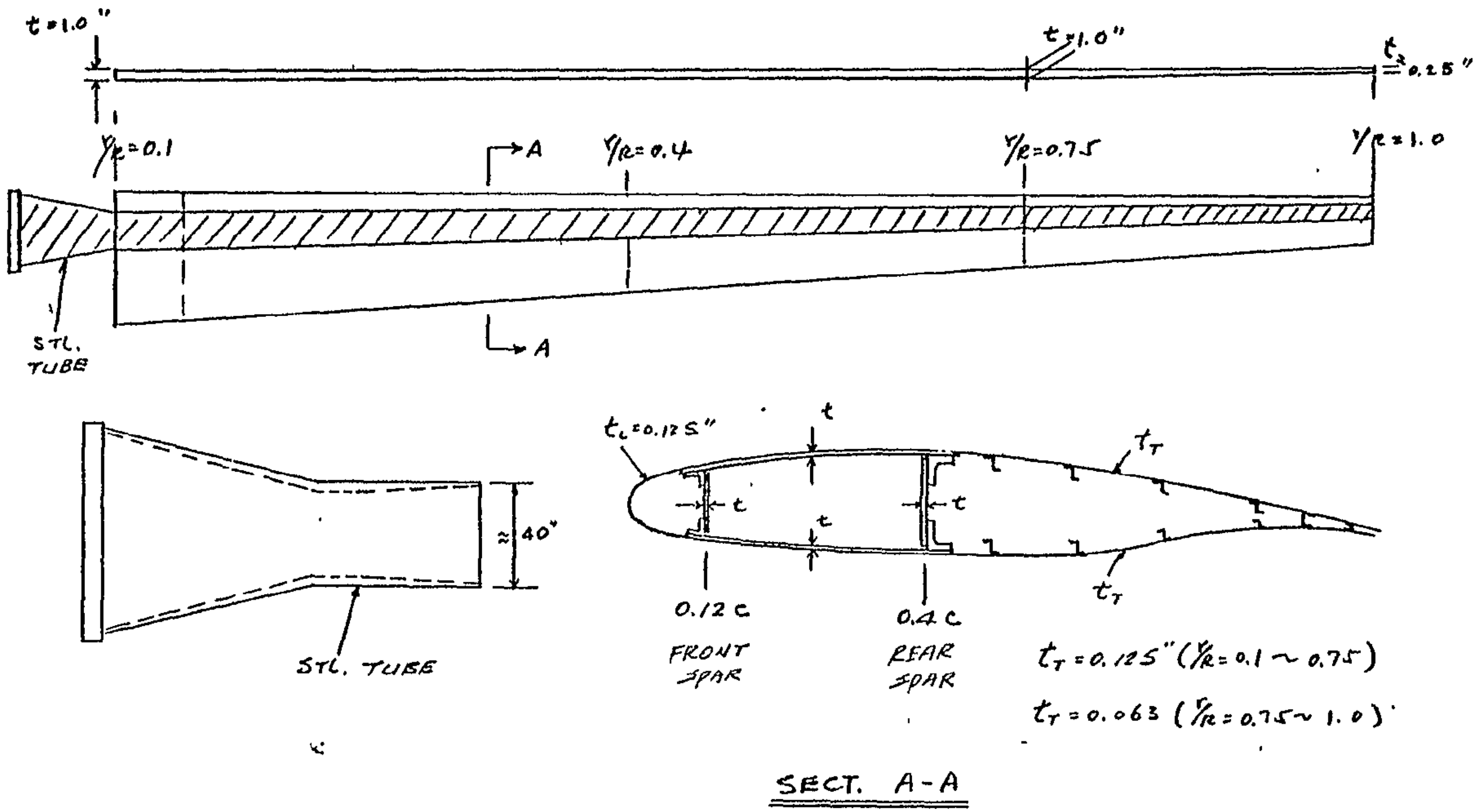

FIGURE 4.3.1 BLADE GEOMETRY

MOD 1 BLADE LAYOUT FOR CONF. 5.X.RF 1

CONFIGURATION S.X.RFI

6/21

r/R	r FEET	r INCHES	c FEET	c INCHES	c/R	t/c	t INCHES	θ DEG
0	0	0	10.15	121.80	.1019	.461	56.19	10.92
.05	4.98	59.76	9.76	117.12	.0980	.431	50.44	9.92
.10 ✓	9.96	119.52	9.37	112.44 ✓	.0941	.400 ✓	44.98 ✓	8.92
.15	14.94	179.28	8.98	107.76	.0902	.369	39.80	7.92
.20 ✓	19.92	239.04	8.59	103.08 ✓	.0863	.339 ✓	34.91 ✓	6.92
.25	24.90	298.80	8.20	98.40	.0823	.308	30.31	5.92
.30	29.88	358.56	7.91	93.72	.0784	.277	25.99	4.92
.35	34.86	418.32	7.42	89.04	.0745	.247	21.96	3.92
Ⓐ .40 ✓	39.84	478.08	7.03	84.36 ✓	.0706	.216 ✓	18.22 ✓	2.92
.45	44.82	537.84	6.64	79.68	.0667	.208	16.57	2.50
.50	49.80	597.60	6.25	75.00	.0628	.200	15.00	2.08
.55	54.78	657.36	5.86	70.32	.0588	.192	13.50	1.67
.60 ✓	59.76	717.12	5.47	65.64 ✓	.0549	.184 ✓	12.08 ✓	1.25
.65	64.74	776.88	5.08	60.96	.0510	.176	10.73	0.83
.70	69.72	836.64	4.69	56.28	.0471	.168	9.46	0.42
.75 ✓	74.70	896.40	4.30	51.60 ✓	.0432	.160 ✓	8.26 ✓	0.00
.80	79.68	956.16	3.91	46.92	.0393	.152	7.13	-0.42
.85	84.66	1015.92	3.52	42.24	.0353	.144	6.08	-0.83
.90 ✓	89.64	1075.68	3.13	37.56 ✓	.0314	.136 ✓	5.11 ✓	-1.25
.95	94.62	1135.44	2.74	32.88	.0275	.128	4.21	-1.67
1.00 ✓	99.60	1195.20	2.35	28.20 ✓	.0236	.120 ✓	3.38 ✓	-2.09

PHYSICAL BLADE AIRFOIL

Ⓐ BREAK IN LINEARITY OF t/c AND θ @ 40% RADIUS

FIG 4.3.2 BLADE GEOMETRY - TABULATED

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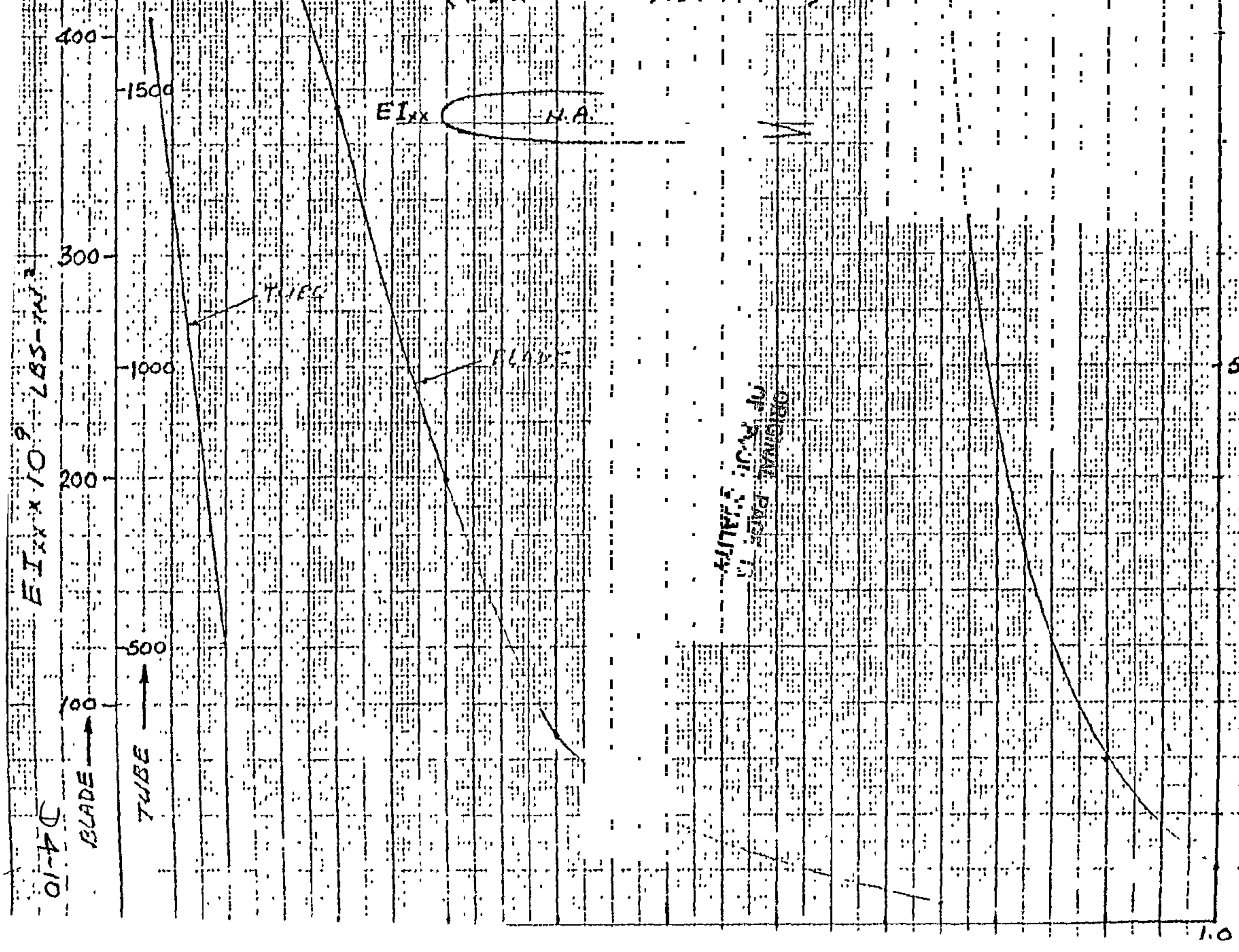
FIG. 4.3.3

STRUCTURAL STIFFNESS – EI_{xx} CURVES

(BLADE CONF. ... ... I)



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PAGE
MODEL S.X.R.E.I
REPORT NO

FIG. 4.3.4
STRUCTURAL STIFFNESS – EI_{zz} CURVE
(BLADE CONF. 5.X.RF1)

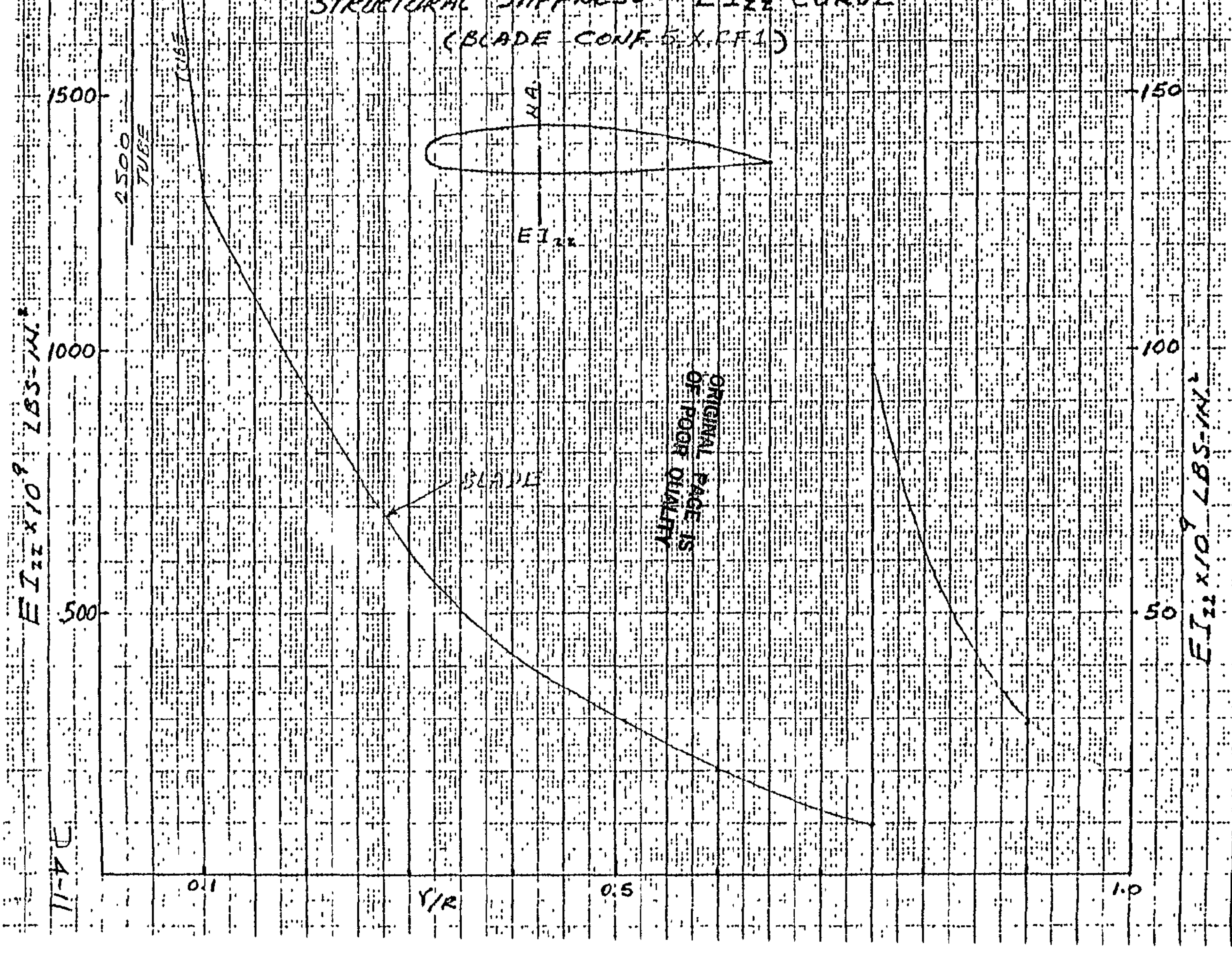
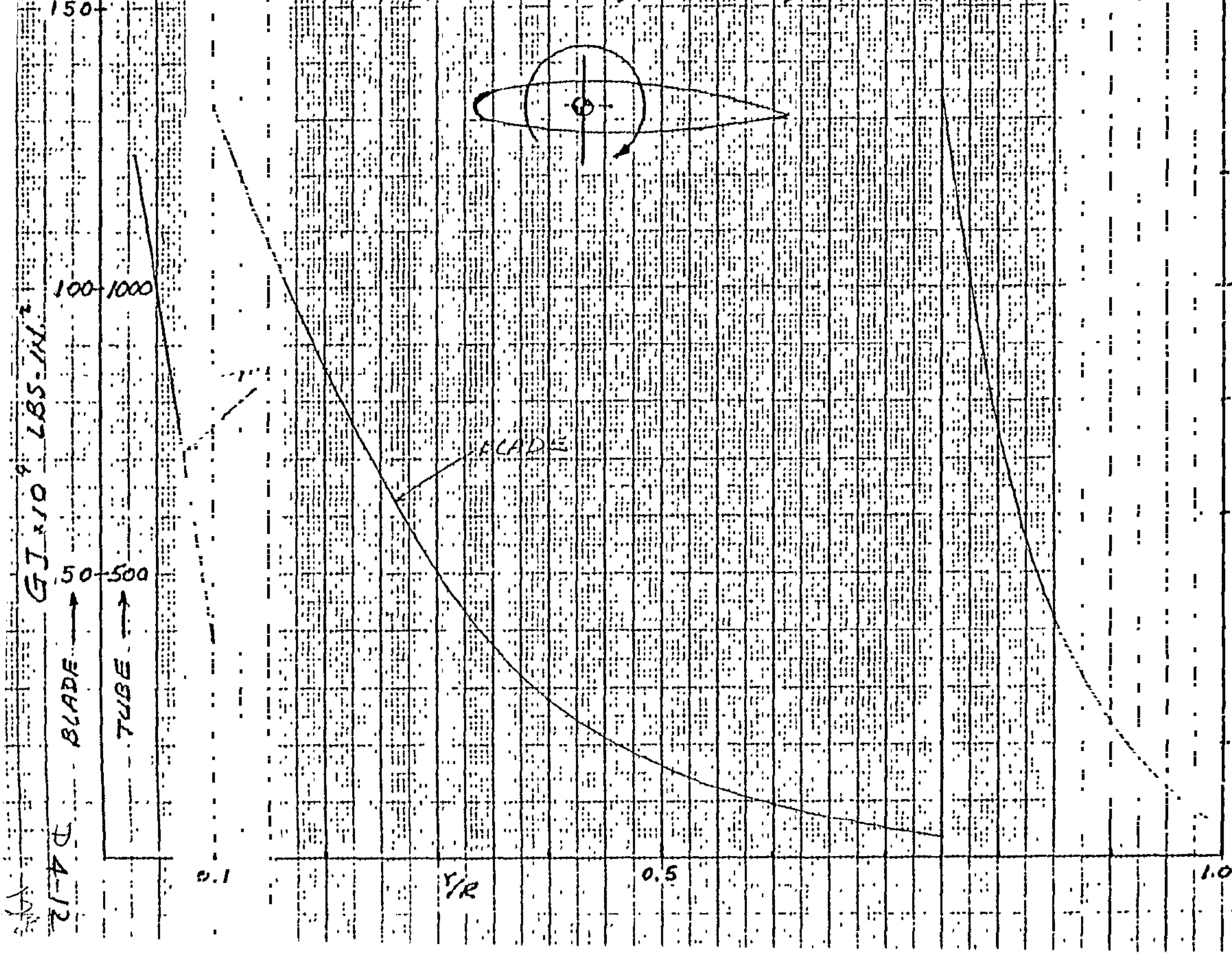


FIG. 4.3.5

STRUCTURAL STIFFNESS - GJ CURVE
(BLADE CONF 5.X. EE 1)



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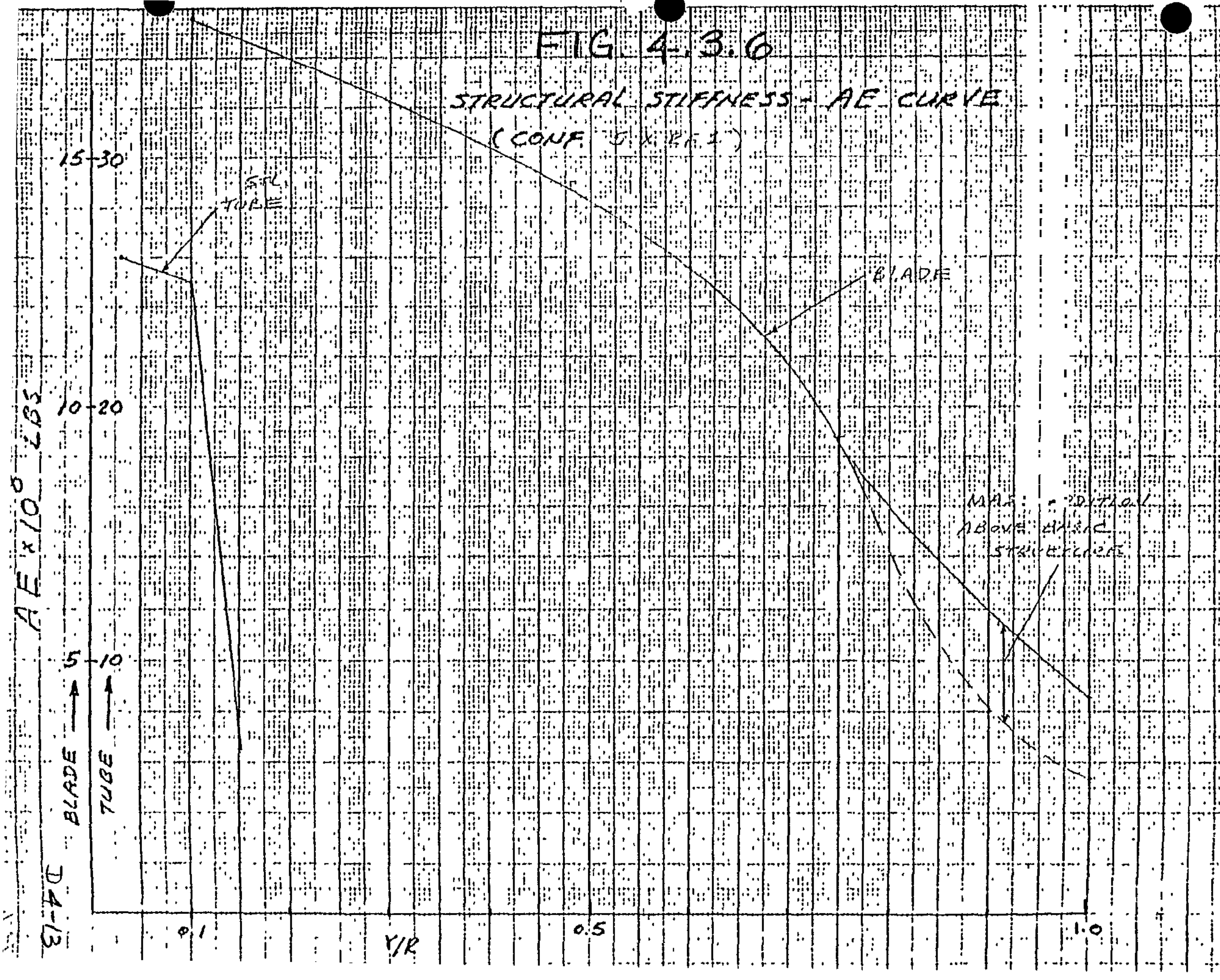
PAGE 3 of 6
MODEL 5.X. EE1
REPORT NO ________



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PAGE 4 OF 6
MODEL 5.X.RF
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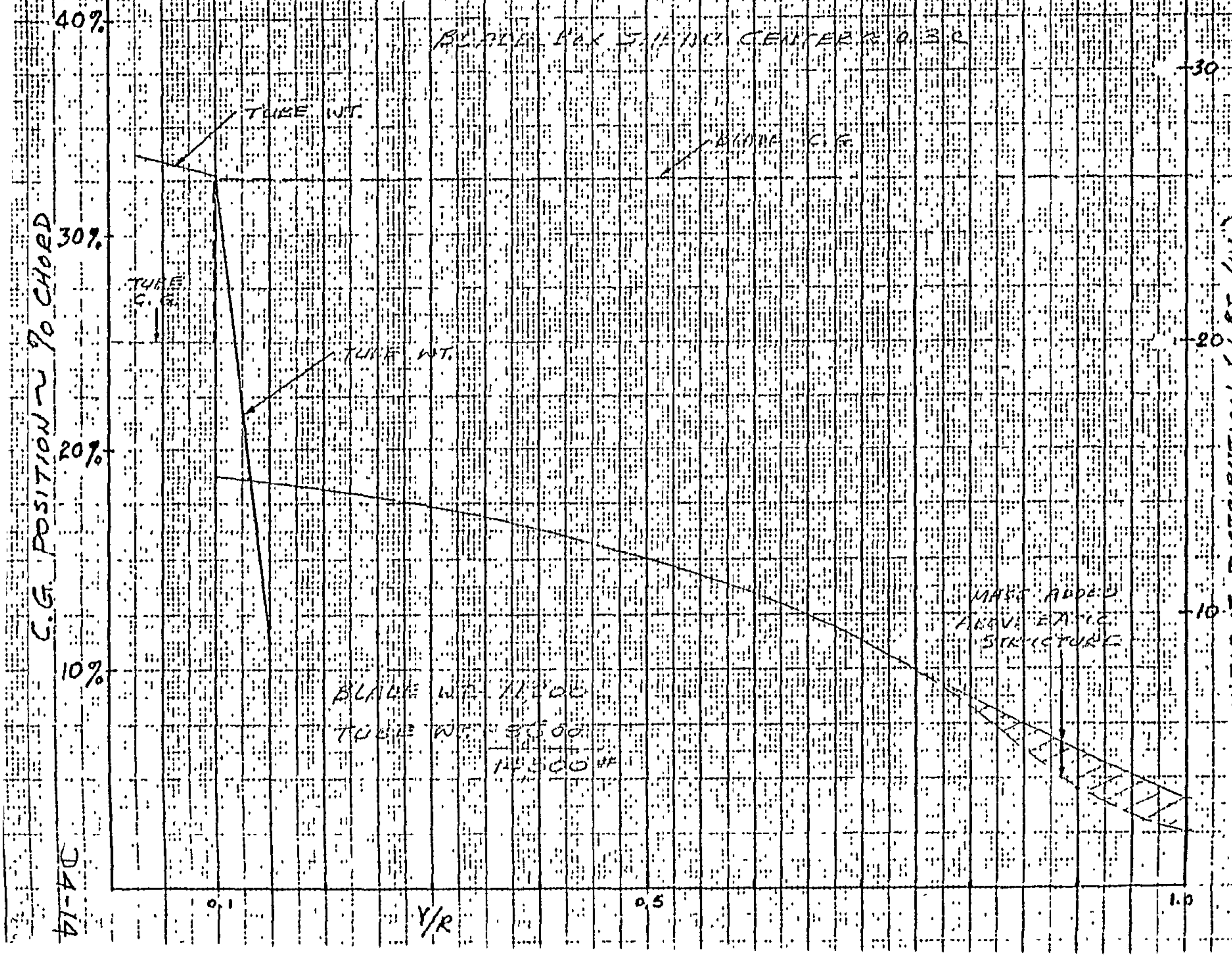
FIG. 4.37

C.G. POSITION (% CHORD) AND
WEIGHT DISTRIBUTION (#/IN.) CONF 5 X RF 1



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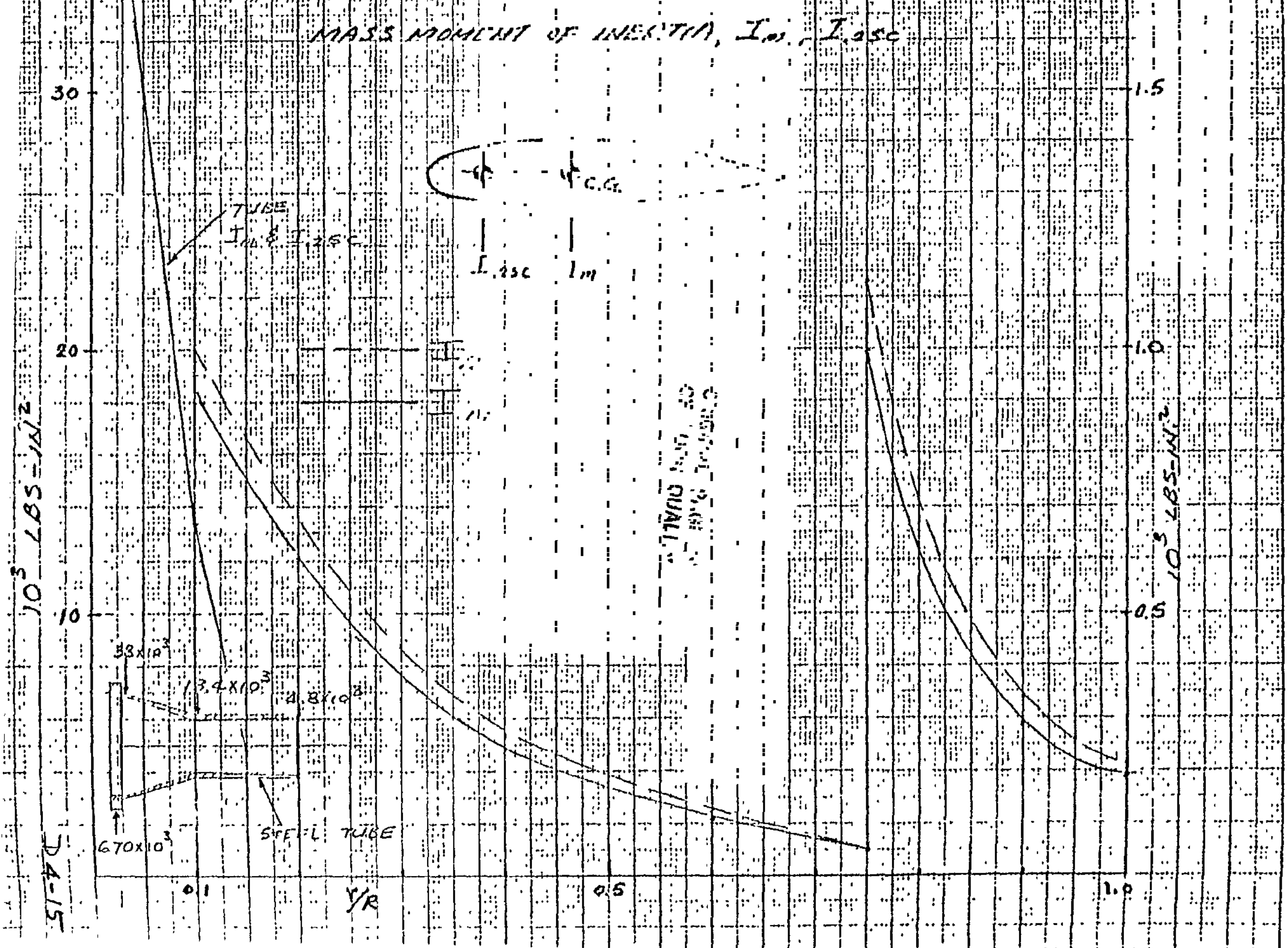
MODEL
REPORT NO

FIG. 4.3.8



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PAGE 6 OF 6
MODEL EX. REF.
REPORT NO

5. DYNAMIC ANALYSES

5.1 Flutter and Divergence Analyses

Flutter and divergence analyses were performed on configuration 2.6.1 for a wind speed of 65 MPH. The subsequent paragraphs report on these analyses.

The basic data used in the analyses are presented in Table 5.1. The calculated vibration analysis modes which are presented in Tables 5.2 and 5.3 were input to the flutter and divergence analyses. The flutter and divergence analyses performed are summarized in Table 4. These analyses investigated the collective and cyclic boundary conditions for variations in:

Control system stiffness

Static mass balance

Tower fore and aft bending modes

As shown in Table 5.4, these flutter and divergence analysis results are presented in Figures 23,24 and 25. (Figures 1 thru 22 and 26 thru 33 have been deleted.)

No analyses were conducted for the configuration 5.X.RF1. It is estimated, though, that the 5.X.RF1 cantilevered blade torsion mode frequency (infinite pitch control stiffness) is only slightly lower than the 22.10 HZ of the 2.6.1 configuration. The 2.6.1 configuration local centers of gravity are between the 36 to 41% chord for an elastic axis at the 34% chord. The 5.X.RF1 local centers of gravity are located at the 32% chord for an elastic axis at the 30% chord. The torsion/bending modes for the 5.X.RF1 configuration are more inertially and aerodynamically decoupled than are the torsion/bending modes for the 2.6.1 configuration.

Stall flutter could be encountered (based on control system stiffness/inertia given) in an emergency feathering condition or in the case of gusts. Therefore, there exists a potential for severe stall induced loads to be developed unless a minimum requirement of .7 reduced frequency is satisfied and maintained. Figure 34 shows blade limit cycle angle versus reduced frequency. Placed on this curve is the configuration 5.X.RF1 estimated reduced frequency.

5.2 Wind Turbine Transient Response and Low Frequency Stability

In the design of minimum cost large wind turbines it is necessary to distribute carefully the blade structural stiffness and mass to achieve acceptable dynamic behavior. System stability in all normal and emergency operating conditions and minimal loads amplification must be achieved.

System stability, for a designed system, may be assessed by use of the rotary wind flutter and the REXOR W.T. computer programs. The flutter program determines the stability of the aerodynamically coupled higher frequency blade modes by a roots extraction procedure. As such it employs linear constant coefficient differential equations with or without insteady aerodynamics.

REXOR W.T. determines the stability of the low frequency blade modes coupled with lower motions and the blade feathering control systems. It employs a step by step integration of the non-linear, periodic coefficient differential equations of the fully coupled modes. Pitch-flap-lag stability is determined by "plucking" the blades in the in-plane direction and noting the decay of the oscillations, so induced.

REXOR W.T.can also be used, for stall flutter analyses. The first dynamic torsion blade mode and possibly a second flap mode should be included for reasonable results.

Steady and quasi-steady oscillatory loads are calculated by WINTWR and rapid transient responses are calculated by REXOR W.T. An example of the latter is the Emergency Feather Condition.

5.3 Pitch-Flap-Lag Stability

The following degrees of freedom were employed to model the Mod 1 wind turbine in the REXOR W.T. program:

- Blade No. 1 flatwise cantilever mode
- Blade No. 2 flatwise cantilever mode
- Rotor cyclic edgewise mode
- Rotor rigid body torsion mode
- Generator armature torsion mode
- Tower first lateral mode
- Tower first fore-aft mode
- Tower first torsion mode

In addition, the blades are free to twist in a quasi-static torsion feathering mode.

The flatwise and edgewise modes of an individual blade are non-linearly coupled due to different bending stiffnesses in the two directions; products of edgewise forces and flatwise deflections and vice-versa lead to feathering moments. Flatwise-edgewise coupling is also furnished by coriolis accelerations.

The wind turbine is driven by a wind shear profile of index =.167 with the reference velocity specified at the 9.0 meter height. A tower shadow of 30.0 degree sector centered on the tower center line was employed with a 21% uniform velocity decrement within the sector.

Structural damping of approximately 3% was employed in the tower and generator shaft and 4% in the blades. Windage damping of a further 3% was applied to the generator armature. A synchronous spring between the armature and tower was employed.

The non-linear edgewise-flatwise coupling noted above contributes to pitch-flap-lag instability. Since the feathering moment terms are functions of blade mean flapping deflection and collective pitch which in turn are functions of power extracted and windspeed, it is necessary to investigate stability over the full range of power available, including transient power ranges at each windspeed-rpm point.

In the present analyses pitch-flap-lag stability investigations were made at the synchronous rpm at cut-in, design and cut-out wind speeds at zero power, design power and double design power respectively. At the design wind speed, zero power was also checked.

A steady state case was run to stimulate a gust at 65 MPH at design power and rpm.

The pitch-flap-lag cases are summarized in Figure 35 and time histories are presented for each of the cases cited. No pitch-flap-lag instabilities were found; though low damping was found in some cases as reported in the presentation to GE of May 23, 1977.

A complete analysis would include investigation at all rpm from zero to overspeed with approximate transient power (both positive and negative) and windspeed considered.

Figure 36 lists the natural frequencies (on primitive frequencies) of the wind turbine components employed in the analysis.

5.4 Emergency Feather

Emergency feather produces significant flap bending loads and large flapping deflections that are of importance in setting tower-blade tip clearance.

A gust of wind is assumed to drive the rotor torque to a level that forces the generator out of sync. This causes the power output to drop to zero and the rotor rpm to accelerate. At the same time the system initiates nose down blade feathering at maximum rate that continues until the blades reach the full feathered stop.

The rotor accelerates for approximately a second until the angle of attack reaches zero. After that the blade lift reverses and a negative torque is produced that reduces rpm.

The rpm reduces rather slowly, however, and after the passage of a further second or two the blade stalls upside down and remains stalled until the rpm approaches zero.

Emergency feather cases listed in Figure 35 were run at the design wind speed under the assumptions that the gust caused loss of sync and then disappeared. A possibly more severe case, with the gust velocity remaining at its maximum value should be examined. Tracings of the emergency feather cases are included in the section of this report dealing with configuration 2.6.1.

Case 4 and 5 were run with 14°/sec. maximum feathering rate. Case 4 employed steady non-linear section aerodynamics suitable for stall flutter analyses. Case 5 showed considerably more dynamic blade motion than Case 4 but could not have shown stall flutter had it existed because of a lack of a blade dynamic torsion mode in the analysis.

Case 10 was run with 8°/sec. for comparative purposes. Maximum loads were not significantly less than those of Case 4. The shapes of th e two sets of curves were different however suggesting that the blade azimuth at emergency feather initiation may have a significant effect on maximum loads.

TABLE 5.1

FLUTTER AND DIVERGENCE ANALYSIS DATA
CONFIGURATION 2.6.1

DATA DESCRIPTION	FIGURE NUMBER
WEIGHT DISTRIBUTION	26
CENTER OF GRAVITY DISTRIBUTION	27
MASS MOMENT OF INERTIA DISTRIBUTION	28
BALANCE WEIGHT DISTRIBUTION	29
EI_{XX} DISTRIBUTION	30
EI_{ZZ} "	31
GJ "	32
BLADE TWIST & CHORD LENGTH DISTRIBUTION	33

BEARING STIFFNESS = 2.35×10^{10} IN-lbs/RAD.
ELASTIC AXIS LOCI = 34% CHORD
NOMINAL CONTROL SYSTEM
STIFFNESS, PER BLADE = 150×10^{6} IN-lbs/RAD
CONTROL SYSTEM WEIGHT = 4000 lbs ON A 20" ARM

TABLE 5.2

MODE DESCRIPTIONS AND FREQUENCIES
CANTILEVER BOUNDARY CONDITIONS
CONFIGURATION 2.6.1

MODE NUMBER	MODE DESCRIPTION	FREQUENCY - Hz		
		UNBALANCED		BALANCED
		.5 RPM	35 RPM	.5 RPM
1	1ST FLAPPING	1.23	1.45	0.97
2	1ST IN-PLANE	2.70	2.72	2.15
3	2ND FLAPPING	3.60	3.84	2.90
4	3RD FLAPPING	7.78	8.01	6.50
5	2ND IN-PLANE	10.07	10.13	8.49
6	4TH FLAPPING	14.00	14.21	11.87
7	1ST TORSION	22.10	22.11	18.60

TABLE 5.3

MODE DESCRIPTION AND FREQUENCIES
COLLECTIVE BOUNDARY CONDITIONS
AT .5 RPM AND MASS UNBALANCED
CONFIGURATION 2.6.1

MODE No.	MODE DESCRIPTION	FREQUENCY ~ Hz CONTROL SYSTEM STIFFNESS AS A PERCENT OF NOMINAL			
		50	37.5	12.5	5
1	1ST IN-PLANE*	.50	.50	.50	.50
2	1ST FLAPPING	1.21	1.21	1.21	1.21
3	1ST TOWER-F&A	1.57	1.57	1.57	1.57
4	2ND FLAPPING	3.61	3.61	3.54	3.65
5	2ND IN-PLANE	5.23	6.23	5.23	5.23
6	3RD IN-PLANE	7.22	7.22	7.22	7.22
7	1ST TORSION	7.46	6.67	4.01	2.46

* 1ST IN-PLANE IS A GENERATOR MODE

TABLE 5.4

SUMMARY OF FLUTTER AND DIVERGENCE ANALYSES - CONFIGURATION 2.6.1

COLLECTIVE ANALYES - NO POWER WINDSPEED = 65 MPH				
FIGURE No.	CONTROL SYSTEM STIFFNESS % NOMINAL	STATIC * MASS BALANCED	TOWER FORE & AFT FREQUENCY	TOWER GENERATOR FREQUENCY = 1.87 Hz
1	50	YES	1.69 Hz	
2	37.5	YES	1.69	
3	25	YES	1.69	
4	12.5	YES	1.69	
5	5	YES	1.69	
6	50	NO	1.69	
7	37.5	NO	1.69	
8	25	NO	1.69	
9	12.5	NO	1.69	
10	5	NO	1.69	
11	50	NO	9.06	
12	50	NO	11.19	

CYCLIC ANALYSIS - NO POWER - 65 MPH		
FIGURE NO.	CONTROL STIFFNESS % NOMINAL	STATIC * MASS BALANCED
13	100	YES
14	75	YES
15	50	YES
16	25	YES
17	10	YES
18	100	NO
19	75	NO
20	50	NO
21	25	NO
22	10	NO

* "YES" MEANS THE BLADE IS BALANCED AT 25% CHORD FOR 50% OF THE OUTER SPAN.

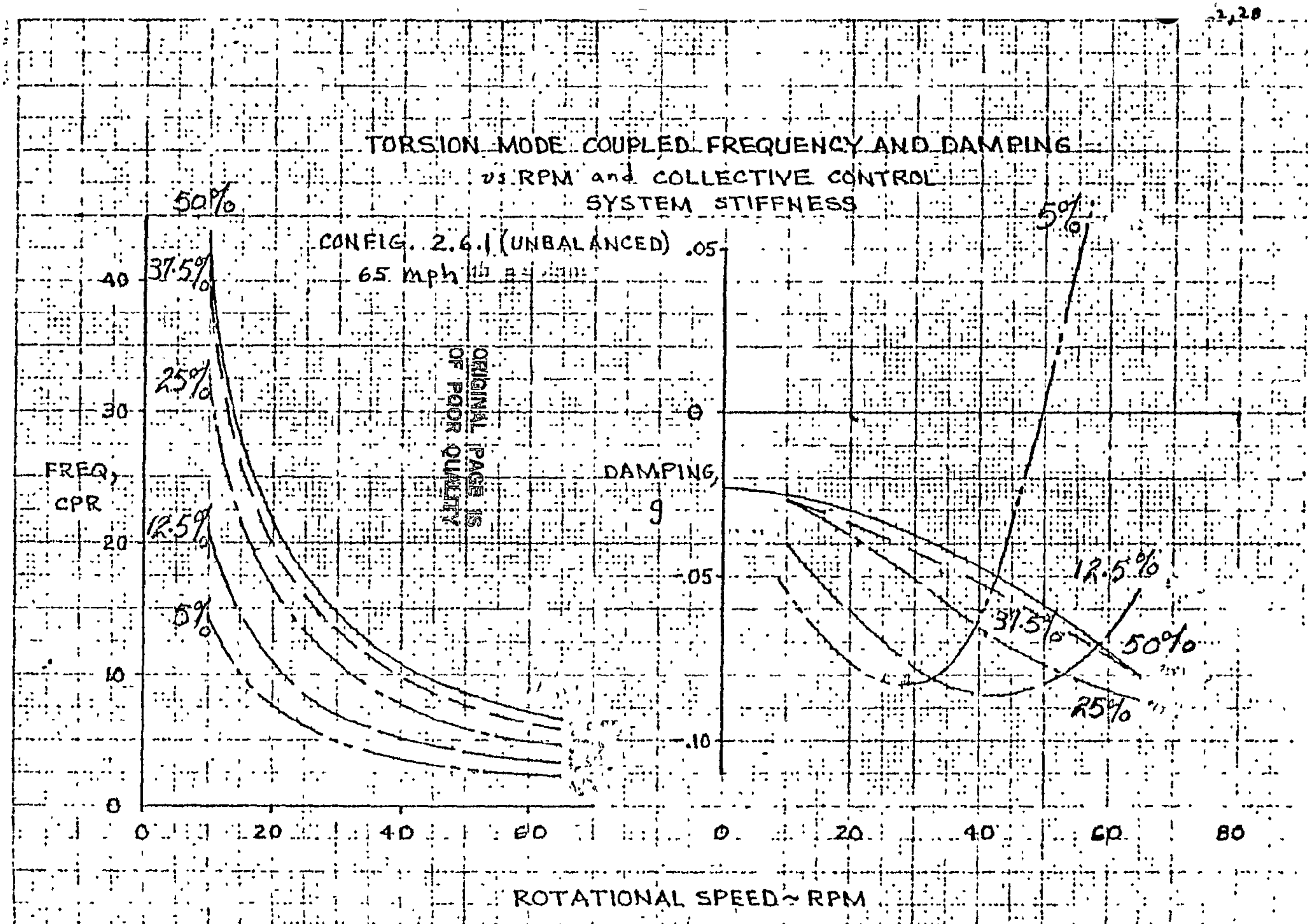
TORSION MODE COUPLED FREQUENCY AND DAMPING
vs RPM and COLLECTIVE CONTROL
SYSTEM STIFFNESS
CONFIG. 2.6.1 (UNBALANCED)
65 mph
FREQ, CPR
DAMPING, g
50%
37.5%
25%
12.5%
5%
ROTATIONAL SPEED ~ RPM
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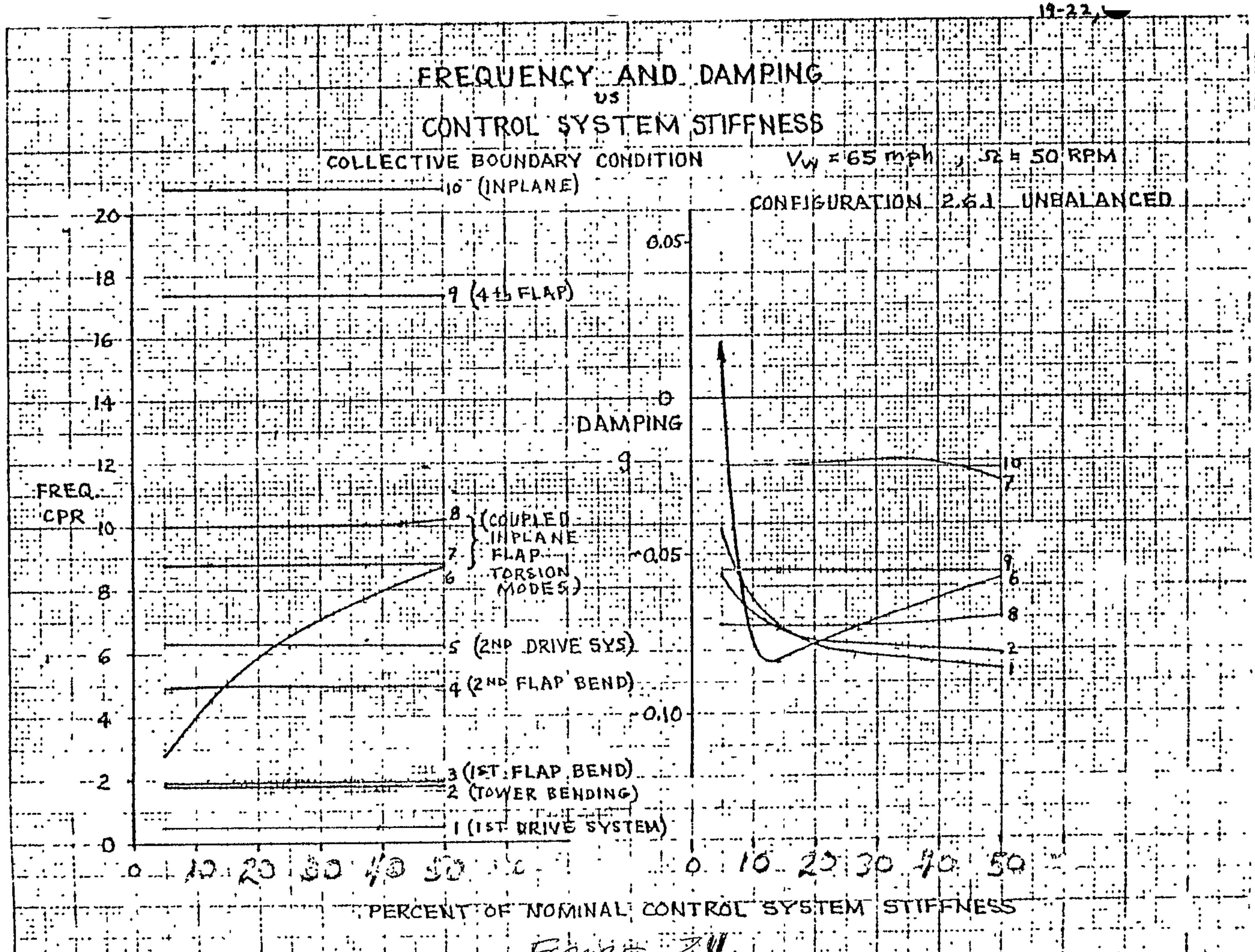
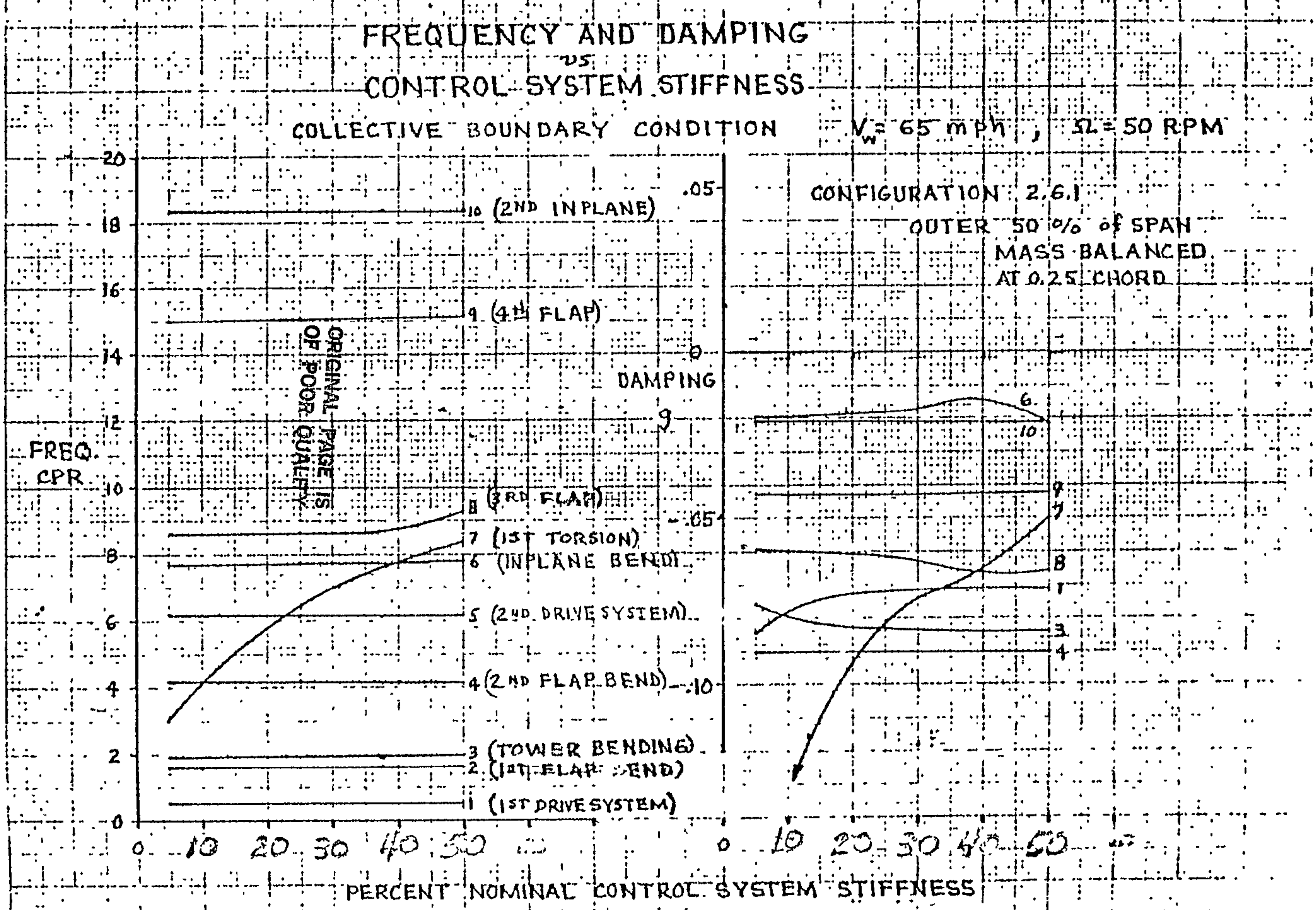
FIGURE 23


FREQUENCY AND DAMPING
VS
CONTROL SYSTEM STIFFNESS
COLLECTIVE BOUNDARY CONDITION
V_w = 65 mph , Ω = 50 RPM
CONFIGURATION 2.61 UNBALANCED
10 (INPLANE)
9 (4th FLAP)
8
7
6
(COUPLED INPLANE FLAP TORSION MODES)
5 (2ND DRIVE SYS)
4 (2ND FLAP BEND)
3 (1ST FLAP BEND)
2 (TOWER BENDING)
1 (1ST DRIVE SYSTEM)
FREQ. CPR
20
18
16
14
12
10
8
6
4
2
0
DAMPING
g
0.05
0
-0.05
-0.10
0 10 20 30 40 50
PERCENT OF NOMINAL CONTROL SYSTEM STIFFNESS


FREQUENCY AND DAMPING
vs
CONTROL SYSTEM STIFFNESS
COLLECTIVE BOUNDARY CONDITION
V_w = 65 mph ; Ω = 50 RPM
CONFIGURATION 2.6.1
OUTER 50 % of SPAN
MASS BALANCED
AT 0.25 CHORD
FREQ. CPR
DAMPING
g
10 (2ND INPLANE)
9 (4th FLAP)
8 (3RD FLAP)
7 (1ST TORSION)
6 (INPLANE BEND)
5 (2ND DRIVE SYSTEM)
4 (2ND FLAP BEND)
3 (TOWER BENDING)
2 (1ST FLAP BEND)
1 (1ST DRIVE SYSTEM)
PERCENT NOMINAL CONTROL SYSTEM STIFFNESS
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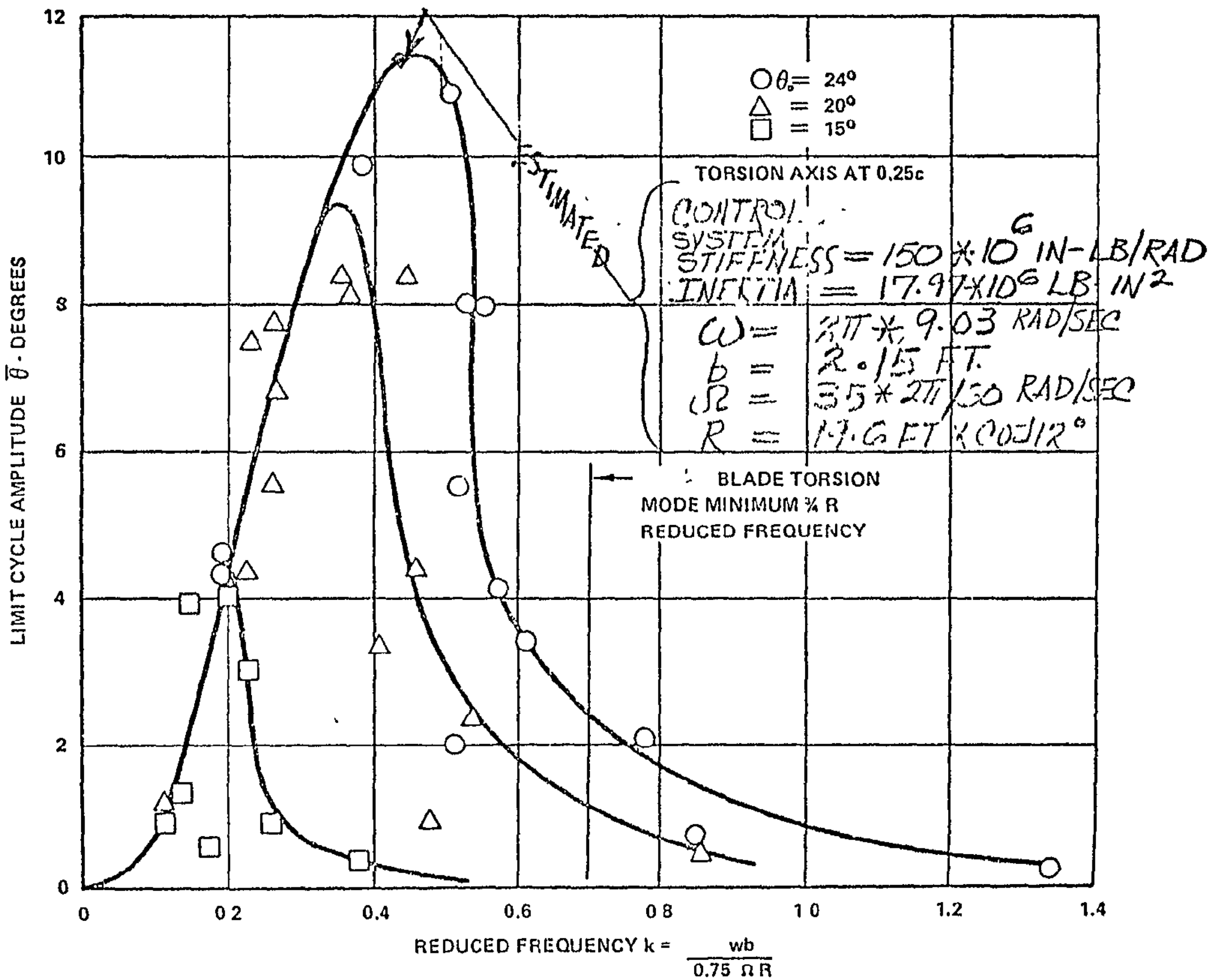

Figure 3 11 Limit Cycle Ampl de vs Reduced Frequency

FIGURE 35 MOD 1 - WIND TURBINE CONFIGURATION 2.6.1

SUMMARY OF REXOR W.T. CASES

CASE No.	DESCRIPTION	WIND SPEED AT 30 FT (MPH)	RPM	MEAN ELECT. OUTPUT POWER KW	FEATHER RATE $\dot{\Theta}_0$ DEG/SEC	COMMENT
2	PITCH-FLAP-LAG	22	35	1500	—	
4	EMERG. FEATHER	22	35*	1500*	14.0	
5	EMERG. FEATHER	22	35*	1500*	14.0	STALL AER
6	PITCH-FLAP-LAG	35	35	1500	—	
7		35	35	1200		
8		10	35	0	—	
9	PITCH-FLAP-LAG	35	35	3000	—	
10	EMERG. FEATHER	22	35*	1500*	8.0	
11	PITCH-FLAP-LAG	65	35	1500	—	
13	PITCH-FLAP-LAG	35	35	0	—	

* INITIAL VALUE

FIGURE 36 MOD 1 - WIND TURBINE CONFIGURATION 2.6.1

NATURAL FREQUENCIES OF SYSTEM COMPONENTS

COMPONENT	MODE TYPE	FREQUENCY		
		HZ	RAD/SEC	PER REV AT 35 RPM
BLADE	1ST EDGEWISE (CANTILEVER)	2.57*	16.12*	4.40P
BLADE	1ST FLATWISE (CANTILEVER)	1.47*	9.21*	2.51P
ROTOR-GENERATOR	1ST TORSION (RIGID ROTOR)	.37	2.33	.64P
ROTOR-GENERATOR	2ND TORSION (RIGID ROTOR)	3.79	23.8	6.49P
TOWER	1ST LATERAL	1.69	10.60	2.89P
TOWER	1ST FORE-AFT	1.69	10.63	2.90P
TOWER	1ST TORSION	5.31	33.36	9.10P

SUMMARY OF CONDITIONS ANALYZED BY CONFIGURATION

Configuration	Flutter	Divergence	Pitch-Flap Lag Stability			Transient Dynamics
			mph	rpm	pwr	
2.6.1	o 0-50 rpm o 65 mph o Cyclic o Collective o Cantilever o 10 - 100% Control Stiff. o Mass Balance	o 40 - 160 mph o Wind over T.E. at 45 o Control Stiff. 10 - 100%	10 35 22 35 35 65 65	35 35 35 35 35 35 50	0 0 1500 1500 3000 1500 1500	1500 kW 35 rpm 22 mph 8 /sec and 14 /sec
5. X	o 0 - 50 rpm o 65 mph o Cyclic o Collective o Cantilever o 10 - 100% Control Stiff. o Mass Balance	o 40 - 160 mph	None			None
5. X NT	o 0 - 50 rpm o 65 mph o Cyclic o Collective o Cantilever o 10 - 100% Control. Stiff. o Mass Balance	None	35 65 65	35 35 50	1500 1500 1500	None

PITCH - FLAP - LAG STABILITY ANALYSIS SUMMARY

CONDITION / CONFIGURATION	V_W, mph	RPM	kW	DAMPING	
				g	g LESS ASSUMED STRUCT. DAMP.
2.6.1	10	35	0	0.11	.07
	35	35	0	0.10	.06
	35	35	1200	0.055	.015
	22	35	1500	0.053	.013
	35	35	1500	0.040	.00
	35	35	3000	0.06	.02
	65	35	1500	-	-
	65	50	1500	-	-
5. X .4	35	35	1500	0.06	.02
(NO TWIST)	65	35	1500	0.06	.02
	65	50	1500	-	-

6. MOD 1 BLADE PRODUCIBILITY

The producibility features of the blade are best demonstrated in the trade studies shown on Pages A B and C. The initial decision to use a heavy plate main spar box resulted in simplifying the structure by eliminating the stringers, ribs and web stiffeners which are usually associated with box spar designs. Elimination of these non-recurring costs has a heavy impact on actual blade costs when dealing with small production quantities.

The trade study (pg. A) compares "D" spar versus box spar structure configurations and steel versus aluminum. The primary disadvantage of the "D" spar configuration is the difficulty of forming the thick surface panels to the tight leading edge radii. The principle disadvantages of steel are the difficulties of forming the panels, the span time for machining the spar caps, the increased weight and the high cost.

Configuration III (aluminum box spar) is clearly the preferred candidate having four methods for skin forming, and is lower in cost by as much as $459,000 for the second blade set.

The trade study (pg. B) compares the box spar skin panel fabrication approach as applied to the aluminum box spar. Candidate I, stretch formed skins is a prime candidate for higher production quantities. The high tooling cost ($259,000 for stretch dies), however, eliminates stretch forming from consideration for small production quantities. Candidate II, shot peen forming, is eliminated for this particular blade configuration because the shot peen process cannot produce the tight contour radius required in the area of the front beam (12% chord).

Candidate III, Die formed skins. This forming process consists of progressively moving the material through an 8-foot long forming die set. This could be a viable candidate for greater production quantities but is more costly than candidate IV for the quantities considered here.

Candidate IV, Extruded skins, is the preferred skin panel fabrication method. These panels require no forming and no machining other than edge trim and is the lowest in cost of the viable candidates.

The trade study (pg. C) compares blade configurations. Except for candidate I, the cost differences are primarily the result of size and weight variations. These candidates are comparable from the standpoint of producibility. Candidate I, in addition to being the largest and heaviest, is also the most complex having a three spar box as compared to the 2 spar box of the other candidates.

The cost figures shown on all trade studies are parametric, developed by Engineering for comparative purposes only, and do not include some items which are common to all candidates. These figures cannot be used to determine the actual cost of any candidate.

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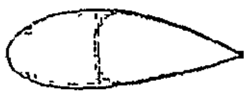
TRADE STUDY: MOD I WIND TURBINE BLADE STRUCTURE

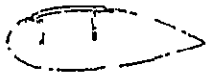
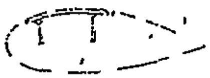
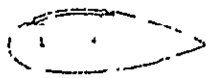
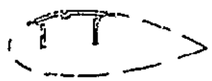
CANDIDATE	CONFIGURATION	ADVANTAGES	DISADVANTAGES	Δ DELTA COST PER BLADE SET (COMPARED TO CONF 2.5)				ORDER OF PREFERENCE
				TOOLING	MATERIAL	LABOR	TOTAL PER BLADE SET	
I CONFIG 2.2 STEEL BOX SPAR ALUM. LEADING & TRAILING EDGES		1 FEWER FASTENERS (THRU USE OF WELDING) THAN 2.5 OR 2.6 2 SMALLER T.E. THAN 2.3 OR 2.6	1 INCREASED WEIGHT 2 WEIGHTS NEEDED TO BALANCE CG 3 CORROSION PROTECTN MORE DIFFICULT 4 REQUIRES LONGER LEAD TIME & SPAN TIME 5 HIGHER COST 6 NOT ADAPTABLE TO ALTERNATE METHODS OF SKIN FORMING (RE CONFIG 2.5)	+$19,200	+$18,638	+$241,620	+$279,458	3
II CONFIG 2.3 STEEL D SPAR ALUM TRAILING EDGE		1 OPTIMUM C.G. CHARACTERISTICS (EASIEST TO BALANCE) 2 FEWER FASTENERS (WELDED STRUCT) THAN 2.5 OR 2.6	1 INCREASED WEIGHT 2 CORROSION PROTECTN MORE DIFFICULT 3 COMPLEX TRANSITION, ROOT TO BLADE 4 T.E. STRUCTURE CARRIES GREATER LOAD 5 LARGER MORE COMPLEX TRAILING EDGE 6 REQUIRES LONGER LEAD TIME & SPAN 7 HIGHER COST 8 NOT ADAPTABLE TO ALTERNATE METHODS OF SKIN FORMING (RE CONFIG 2.5)	+$103,112	-$31,600	+$106,500	+$178,012	4
III CONFIG 2.5 ALUMINUM BOX SPAR ALUM. LEADING & TRAILING EDGES		1 SMALLER T.E. THAN CONF 2.3 OR 2.6 2 BACKUP METHODS AVAIL. FOR PRODUCING CONTOURED BOX SURFACE PANELS i.e. EXTRUSION, SHOT PEENING PROCRESSIVE DIE FORMING (SEE SURFACE PANEL TRADE STUDY)	1 WEIGHTS NEEDED TO BALANCE C.G. 2 LARGER NUMBER OF FASTENERS	θ	θ	θ	θ (FURTHER REDUCTION OF -$180,000 WITH ALTERNATE SKIN FORMING METHOD)	1
IV CONFIG 2.6 ALUMINUM D SPAR ALUM TRAILING EDGE		1 GOOD C.G. CHARACTERISTICS 2 FEWER FASTENERS THAN 2.5	1 DIFFICULT FORMING OF LEAD EDGE 2 COMPLEX TRANSITION, ROOT TO BLADE 3 T.E. STRUCTURE CARRIES GREATER LOAD 4 LARGER, MORE COMPLEX TRAIL EDGE 5 HIGH TOOLING COSTS 7 NOT ADAPTABLE TO ALTERNATE 8 METHODS OF SKIN FORMING (RE CONFIG 2.5)	+$83,912	-$40,960	+$37,980	+$80,932	2

PAGE A D6-3

WE-1-TS-14

TRADE STUDY: MOD I WIND TURBINE BLADE

ALUMINUM SKIN SURFACES - BOX SPAR

CANDIDATE	CONFIGURATION	ADVANTAGES	DISADVANTAGES	RISKS	DELTA COST PER BLADE SET (COMPARED TO CANDIDATE I) TOOLING	MATERIAL	LABOR	TOTAL COST	ORDER OF PREFERENCE
I STRETCH FORMED SKINS		FEWER FASTENERS THAN CONFIG IV	EXPENSIVE TOOLING 15% MATERIAL WASTE GREATER NUMBER OF FASTENERS DUE TO ONE ADDITIONAL CHORDWISE SPLICE 5 TO 6 MONTHS DELIVERY OF FORMED PANELS CREATES SCHEDULE IMPASSE		0	0	0	0	3
II SHOT-PEEN FORMED SKINS		FEWER FASTENERS THAN CONFIG IV EDGE TRIM PRIOR TO FORMING SAVES MATERIAL	SMALL LEADING EDGE RADIUS CANNOT BE FORMED BY THIS METHOD		-$207,134	+$975	+$25,540	-$180,619	4
III DIE FORMED SKINS		FEWER FASTENERS THAN CONFIG IV POSSIBLE EDGE TRIM PRIOR TO FORMING SAVES MATERIAL		REQUIRES DEVELOPMENT TO ACHIEVE ACCURATE CONTOURS & TWIST ABILITY TO FORM THICK ALUM SKINS TO REQ'D CONTOUR USING THE BIRDSBORO HAS NOT BEEN DEMONSTRATED	-122,000	0	+11,520	-110,480	2
IV EXTRUDED SKINS		NO FORMING REQUIRED FAIL SAFE SURFACES	GREATER NUMBER OF FASTENERS BECAUSE OF LONGITUDINAL SKIN SPLICE 15% MATERIAL WASTE		-$220,134	+$17,670	+$8550	-$193,884	1

[illegible] INCLUDE SOME ITEMS COMMON TO ALL CANDIDATES

WE-1-TS-23

TRADE STUDY: MOD. 1 WIND TURBINE BLADE

BLADE CONFIGURATION

CANDIDATE	DESCRIPTION	SHAFT POWER - KW RATED 22 MPH	SHAFT POWER - KW CUT-IN 11 MPH	BLADE DESIGN EFFORT	BASIC WEIGHT/SET (BALANCE WTS NOT INCL)	DYNAMICS	RELATIVE COST ⚠ TOOLING	RELATIVE COST ⚠ 1ST SET (NO TOOLING)	RELATIVE COST ⚠ 1ST SET (INCL TOOLING)	RELATIVE COST ⚠ 2ND SET	ORDER OF PREFERENCE
1 ([illegible])	44XY AIRFOIL 35 RPM TWIST 103 75 R - 3 SPAR ROOT CD 10% = 125 6 TIP CD = 32 4	1752	99	120%	32,340	4TH	-0-	-0-	-0-	-0-	
2 ([illegible])	44XX AIRFOIL 31 5 RPM TWIST 103 75 R - 2 SPAR ROOT CD 10% = 123 6 TIP CD = 32 4	1685	145	118%	30,600	3RD	-61,600	-72 300	-133,900	-64,300	
3 ([illegible])	LS 1 AIRFOIL 35 RPM TWIST 99 7 R - 2 SPAR ROOT CD 10% = 109 7 TIP CD = 27 3	1670	111	100%	23,000	1ST	-67,000	-110,300	-177,300	-98,100	
4 ([illegible])	LS 1 AIRFOIL 31 5 RPM TWIST 99 7 R - 2 SPAR ROOT CD 10% = 123 0 TIP CD = 32 4	1670	131	105%	25,000	2ND	-69,500	-106,000	-175,500	-94,300	
5 ([illegible])	LS 1 AIRFOIL 31 5 RPM NO TWIST 102 4 R - 2 SPAR ROOT CD 10% = 119 6 TIP CD = 32 4	1671	110	108%	25,600	2ND	-69,500	-98,400	-167,900	-97,500	
6 ([illegible])	44XX AIRFOIL 35 RPM TWIST-DEEP ROOT 103 75 R - 2 SPAR ROOT CD 15% = [illegible] TIP CD = 32 4	1714	92	118%	29,000	3RD	-62,400	-74200	-136,600	-66,000	

⚠ COSTS ARE PARAMETRIC, DEVELOPED BY ENGINEERING FOR COMPARATIVE PURPOSES ONLY & DO NOT INCLUDE SOME ITEMS COMMON TO ALL CANDIDATES

7. GENERAL RECOMMENDATIONS

A. Dynamics substantiation analysis of the coupled system is recommended following completion of the detailed design. This is essential to further the assurance of a successful design.

B. The pitch control mechanism system integrity is critical to the stability margins of the system, both with respect to classical flutter and stall flutter. It is currently estimated that 600 million inch-lbs./rad stiffness will be required to prevent serious load amplification due to stall unless damping is present or added to the pitch circuit.

C. Slop in either the blade root bearing system and/or pitch circuit will result in self higher harmonic tuning, which produces high loadings and may produce limit cycle oscillations.

D. The design must offer the widest possible latitude for tuning adjustments of dynamic components so that movement within the constraints is still possible. Possible variations in design versus actual test results obtained, as may be expected in any hardware development program, must be considered as basic required latitudes for which such provisions are made.

E. Proposed aeroelastic and dynamic stability criteria for Mod-1 are:

   a. Limited available pitch-flap-lag stability solutions indicate that the nominal stiffness of 150 million inch-lbs./rad is marginal.

   b. The coupled system predicted loads are highly dependent on the accuracy of the support description and must be current therefore with respect to the fabricated design, since harmonic loadings are amplified due to inner harmonic coupling effects.

   c. Structural fatigue life is a direct function of stress concentrations associated with the details of both the sub-structure, as well as the super structure of the detail design. A K_t of 5 is recommended as a target for a well executed design.

E. (Continued)

d. Emergency conditions commonly produce the highest transient loading cases, based on experience, and these are functions of the logic applied to the control circuits. System control logic must constantly be regarded with respect to structural adequacy when adoption of operational modes and subsequent failure modes are considered.

e. Limit Design Rotor Speed - The maximum anticipated rotor overspeed as used for design. Establishment of this overspeed shall be based on due consideration of all control system failure modes.

Flutter and Divergence-Rotating - The rotor system shall be shown to be free by analysis from aerodynamically induced flutter and divergence to in excess of 1.25 times limit design rotor speed. (Also, by test, the system shall be shown to be stable to 1.05 times limit design rotor speed.)

Pitch-Flap-Lag Stability - By analysis the system stability shall be shown to no less than the assumed structural damping within the operational envelope and to limit design rotor speed. (Also, by test, the system shall be stable over the operating envelope with a margin of not less than a g=0.015 and shall be stable at 1.05 times limit design rotor speed.)

Flutter and Divergence-Non-Rotating - The system shall be shown by appropriate analytical methods to be free from aerodynamically induced flutter and divergence to in excess of 1.2 times the maximum hurricane design wind speed with the wind coming from any direction.

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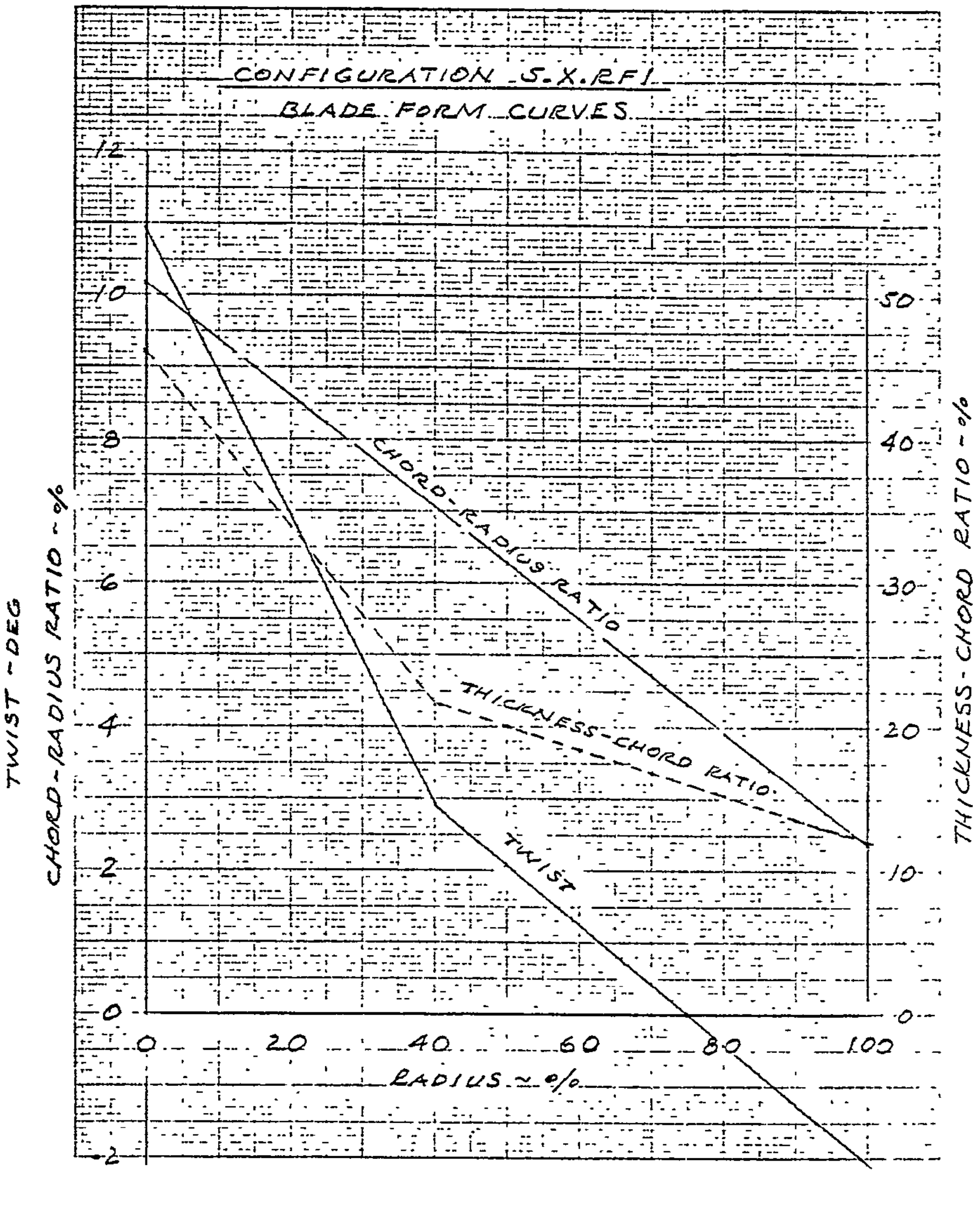

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CONFIGURATION S.X.-RFI Wg Coty 6-2-77 6/2/17

r/R	x FEET	x INCHES	c FEET	c INCHES	s/R	z/c	z INCHES	θ DEG
0	0	0	10.15	121.80	.1019	.461	56.19	10.92
.05	4.98	59.76	9.76	117.12	.0980	.431	50.44	9.92
.10	9.96	119.52	9.37	112.44	.0941	.400	44.98	8.92
.15	14.94	179.28	8.98	107.76	.0902	.369	39.80	7.92
.20	19.92	239.04	8.59	103.08	.0862	.339	34.91	6.92
.25	24.90	298.80	8.20	98.40	.0823	.308	30.31	5.92
.30	29.88	358.56	7.81	93.72	.0784	.277	25.99	4.92
.35	34.86	418.32	7.42	89.04	.0745	.247	21.96	3.92
Ⓐ .40	39.84	478.08	7.03	84.36	.0706	.216	18.22	2.92
.45	44.82	537.84	6.64	79.68	.0667	.252	16.57	2.50
.50	49.80	597.60	6.25	75.00	.0628	.200	15.00	2.08
.55	54.78	657.36	5.86	70.32	.0588	.192	13.50	1.67
.60	59.76	717.12	5.47	65.64	.0549	.184	12.08	1.25
.65	64.74	776.88	5.08	[illegible]	.0510	.176	10.73	0.83
.70	69.72	836.64	4.69	[illegible]	.0471	.168	9.46	0.42
.75	74.70	896.40	4.30	[illegible]	.0432	.160	8.26	0.00
.80	79.68	956.16	3.91	[illegible]	.0393	.152	7.13	-0.42
.85	84.66	1015.92	3.52	42.24	.0353	.144	6.08	-0.83
.90	89.64	1075.68	3.13	37.56	.0314	.136	5.11	-1.25
.95	94.62	1135.44	2.74	32.88	.0275	.128	4.21	-1.67
1.00	99.60	1195.20	2.35	28.20	.0236	.120	3.38	-2.08

PHYSICAL BLADE AIRFOIL

Ⓐ: BREAK IN LINEARITY OF z/c AND θ @ 40% RADIUS

TO: DICK POOR - MOD I PROG.

D8-2

FORM LAC 38

WORK SHEET

6-3-77
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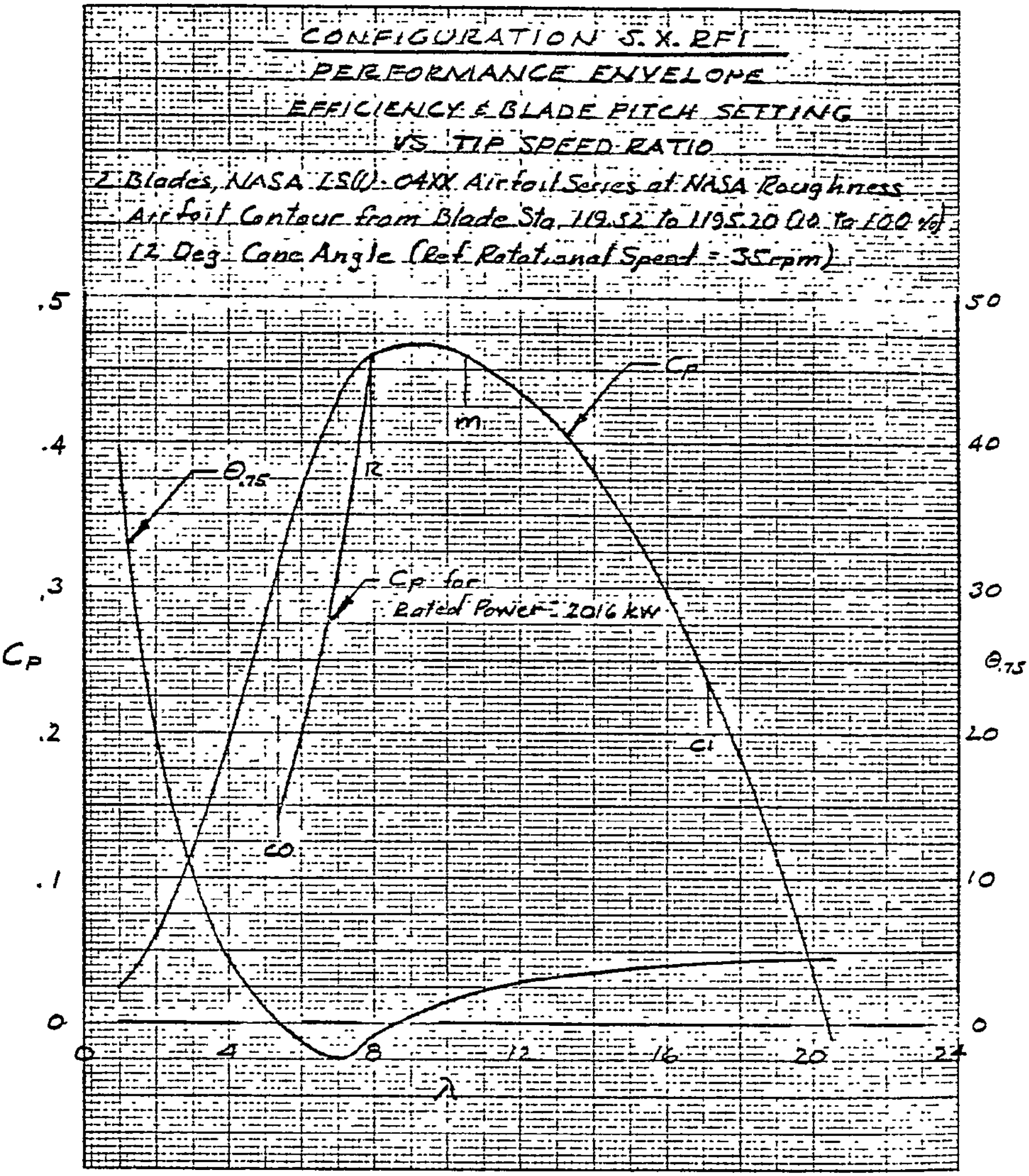

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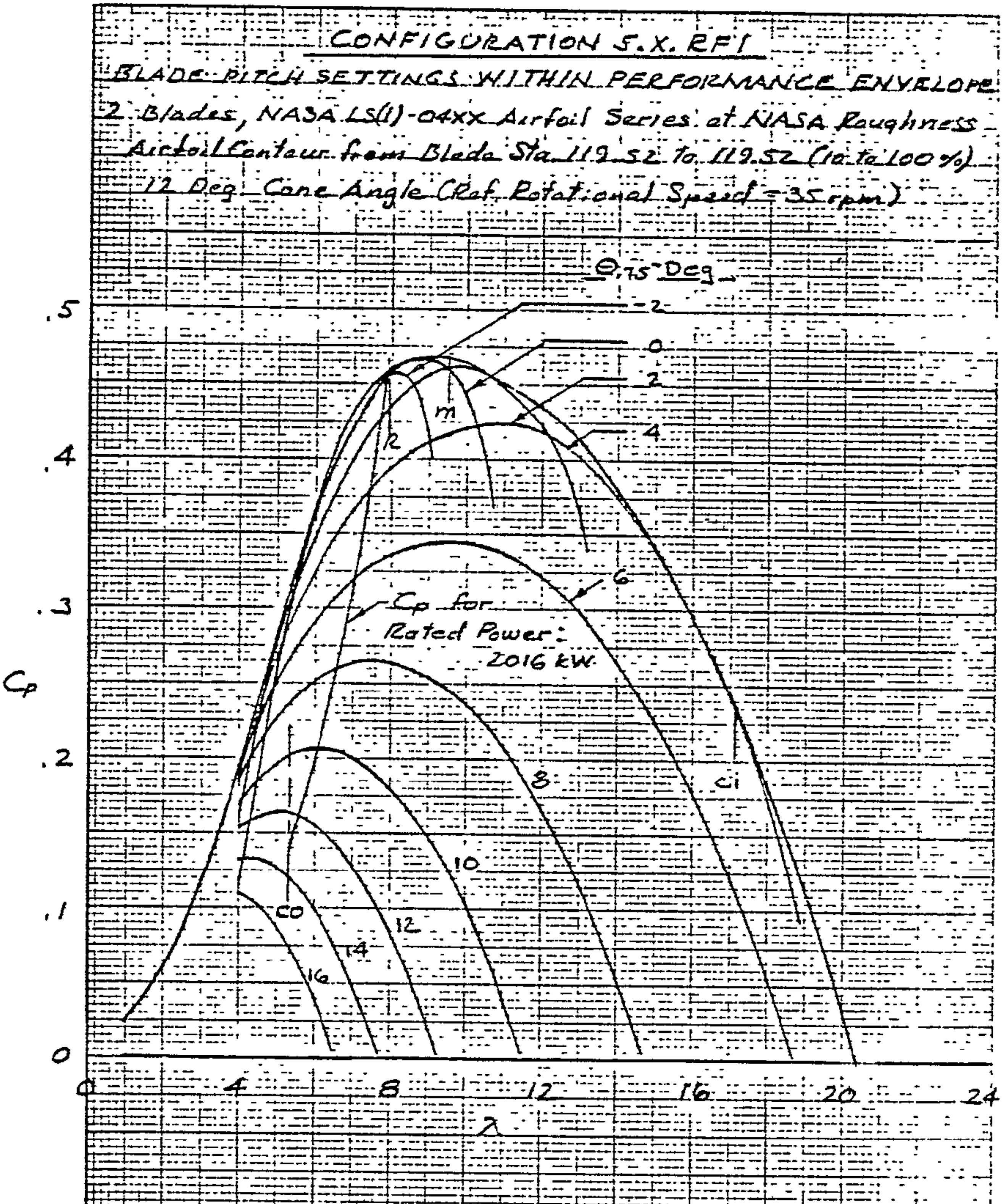

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J.X. RF1

6-3-77
BSW

	DATA FOR PERFORMANCE ENVELOPE				CP'S AT RATED POWER WITHIN OR AT ENVELOPE		
λ	$\theta_{.75}$	① $C_{P_{TAB}}$	② $C_{P_{TAB}}/(\cos^2 12°)$	KW	② $C_{P_{TAB}}/(\cos^2 12°)$	KW	λ
1	39.3	.02263	.0237				
1.5	27.3	.03733	.0390				
2	19.1	.05697	.0595				
3	10.4	.1124	.1175				
4	4.2	.1854	.1938				
5	1.2	.2689	.2810				
5.378	-.1	.3011	.3147	4425	.1434	2016	5.378
5.5	-.3	.3113	.3254	4277	.1534	↓	5.5
6	-1.0	.3506	.3664	3711	.1990	↓	6.
6.5	-2.2	.3860	.4034	3214	.2530	↓	6.5
7	-2.5	.4130	.4317	2753	.3161	↓	7
7.5	-1.9	.4304	.4498	2333	.3887	↓	7.5
7.930	-.8	.4397	.4596	2016	.4596	↓	7.930
7.954	-.8	.4403	.4602				
8	-.7	.4413	.4612				
8.5	-.2	.4461	.4663				
8.557	-.1	.4462	.4664				
9	.5	.4479	.4681				
9.5	1.0	.4476	.4678				
10	1.5	.4435	.4635				
10.458	1.8	.4379	.4577				
11	2.3	.4294	.4488				
12	2.8	.4173	.4362				
13	3.2	.3948	.4126				
14	3.5	.3641	.3806				
15	3.8	.3265	.3413				
16	4.0	.2837	.2965				
17	4.2	.2333	.2438				
17.112	4.2	.2272	.2375				
17.114	4.2	.2270	.2373				
18	4.3	.1749	.1828				
19	4.4	.1082	.1131				
20	4.5	.03294	.0344				
20.4	4.6	.0002666	.0003				
20.5	4.6	$\overline{.008078}$	$\overline{.0085}$				

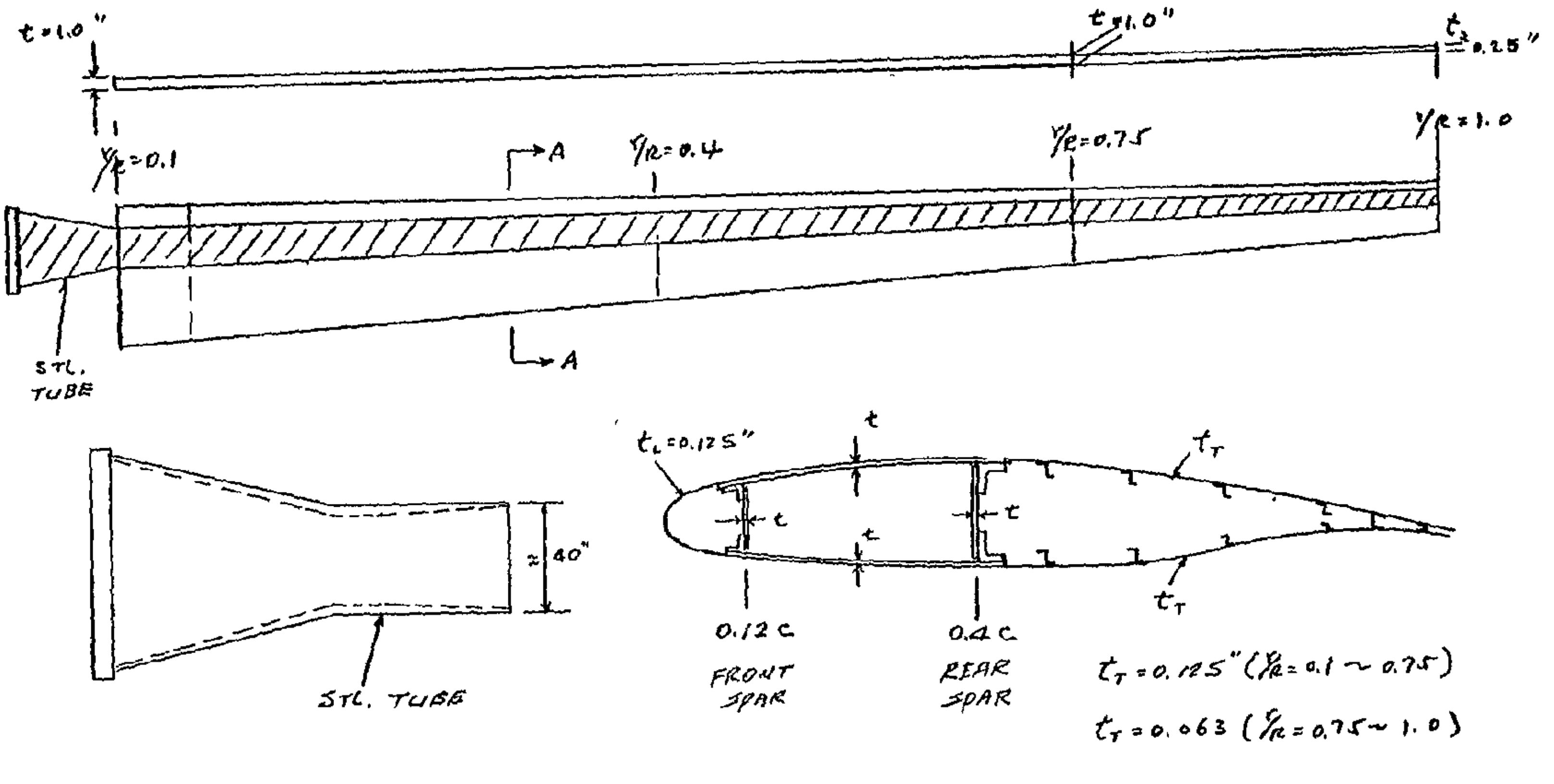
① $C_{P_{TAB}} = P/[\frac{\rho}{2} V^3 \pi S^2]$, WHERE S = BLADE LENGTH

② $C_{P_{PROJECTED\ AREA}} = C_{P_{TAB}}/(\cos^2 12°)$, WHERE 12° = CONE ANGLE

D 8-6

FORM LAC 38

WORK SHEET



SECT. A-A

NO. 1 BLADE LAYOUT FOR CONF. 5.X.RF 1

CONFIGURATION S.X.RF1 6-2-78

6/21

P. 5 of

r/R	r FEET	r INCHES	c FEET	c INCHES	c/R	t/c	t INCHES	θ DEG
0	0	0	10.15	121.80	.1019	.461	56.19	10.92
.05	4.98	59.76	9.76	117.12	.0980	.431	50.44	9.92
.10 ✓	9.96	119.52	9.37	112.44 ✓	.0941	.400 ✓	44.98 ✓	8.92
.15	14.94	179.28	8.98	107.76	.0902	.369	39.80	7.92
.20 ✓	19.92	239.04	8.59	103.08 ✓	.0862	.339 ✓	34.91 ✓	6.92
.25	24.90	298.80	8.20	98.40	.0823	.308	30.31	5.92
.30	29.88	358.56	7.81	93.72	.0784	.277	25.96	4.92
.35	34.86	418.32	7.42	89.04	.0745	.247	21.96	3.92
(A) .40 ✓	39.84	478.08	7.03	84.36 ✓	.0706	.216 ✓	18.22 ✓	2.92
.45	44.82	537.84	6.64	79.68	.0667	.208	16.57	2.50
.50	49.80	597.60	6.25	75.00	.0628	.200	15.00	2.08
.55	54.78	657.36	5.86	70.32	.0588	.192	13.50	1.67
.60 ✓	59.76	717.12	5.47	65.64 ✓	.0549	.184 ✓	12.08 ✓	1.25
.65	64.74	776.88	5.08	60.96	.0510	.176	10.73	0.83
.70	69.72	836.64	4.69	56.28	.0471	.168	9.46	0.42
.75 ✓	74.70	896.40	4.30	51.60 ✓	.0432	.160 ✓	8.26 ✓	0.00
.80	79.68	956.16	3.91	46.92	.0393	.152	7.13	-0.42
.85	84.66	1015.92	3.52	42.24	.0353	.144	6.08	-0.83
.90 ✓	89.64	1075.68	3.13	37.56 ✓	.0314	.136 ✓	5.11 ✓	-1.25
.95	94.62	1135.44	2.74	32.88	.0275	.128	4.21	-1.67
1.00 ✓	99.60	1195.20	2.35	28.20 ✓	.0236	.120 ✓	3.38 ✓	-2.09

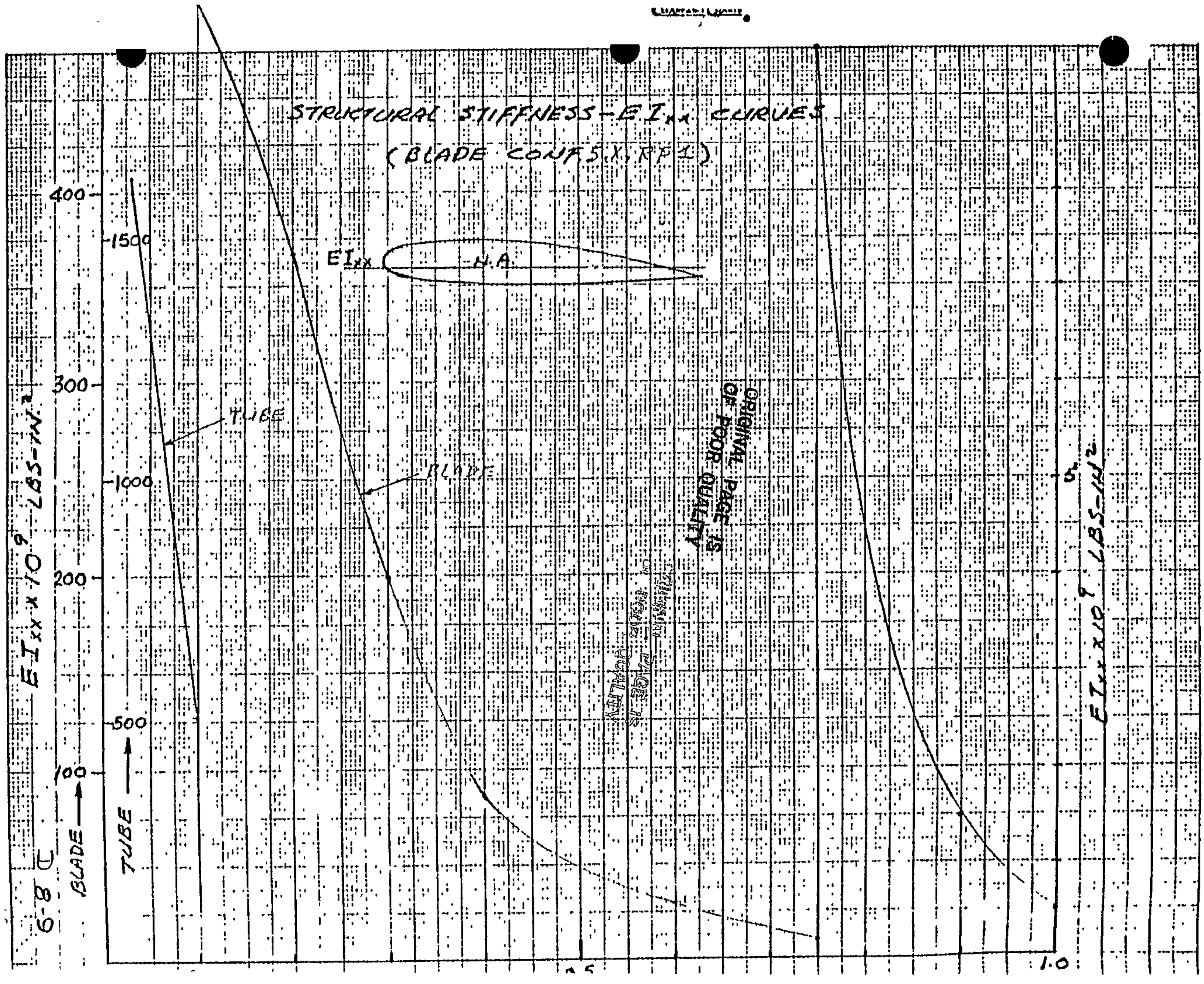
PHYSICAL BLADE AIRFOIL

(A) BREAK IN LINEARITY OF t/c AND θ @ 40% RADIUS

P 8-8

FORM LAC 3D

WORK SHEET



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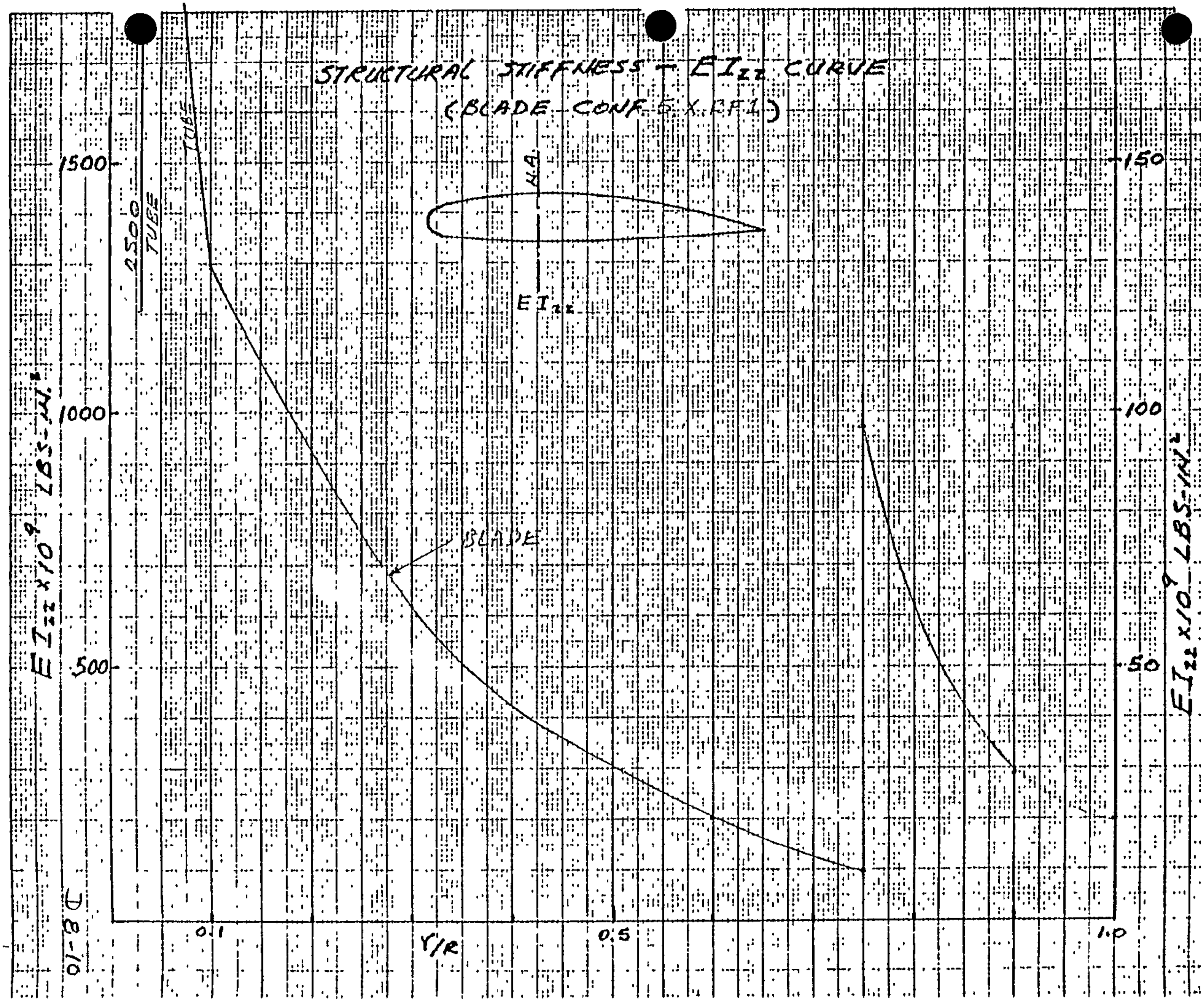
PREPARED BY MIKE NIU
DATE 6-3-77
CHECKED BY

PAGE 1 OF 6
MODEL 5.X RF1
REPORT NO



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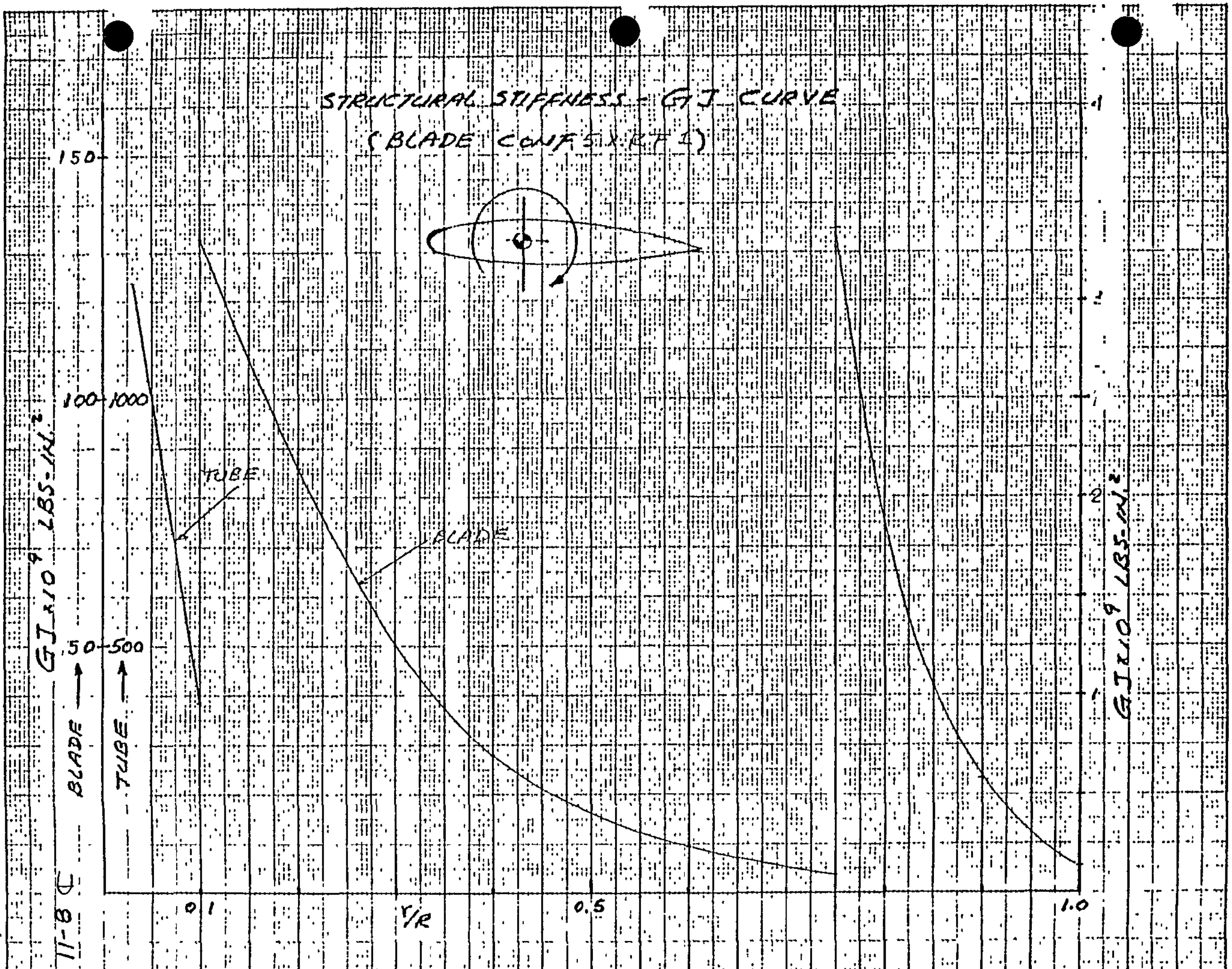
PAGE 2 of 6
MODEL 5 X RF1
REPORT NO ____



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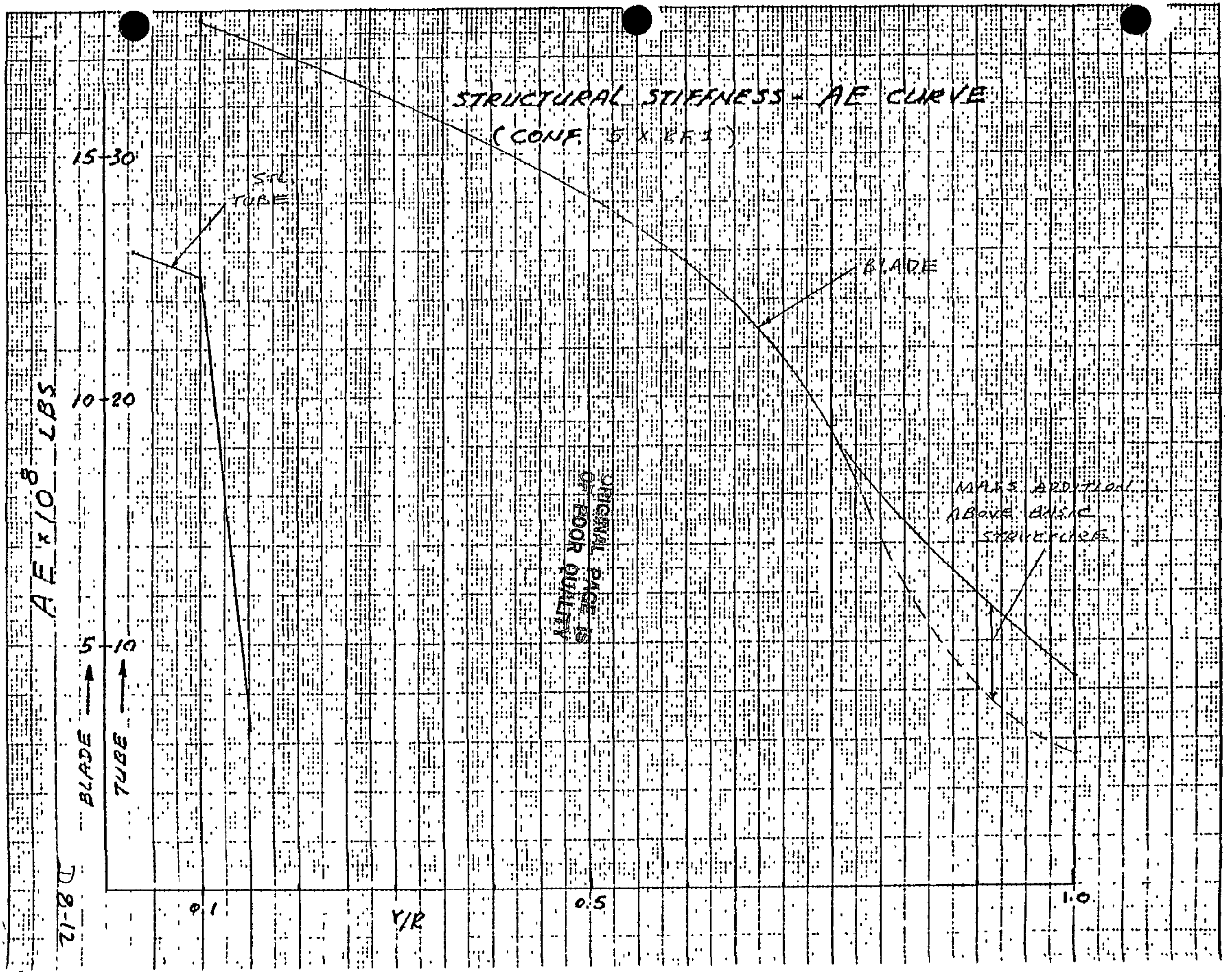
PAGE 3 of 6
MODEL 5 X RF1
REPORT NO.



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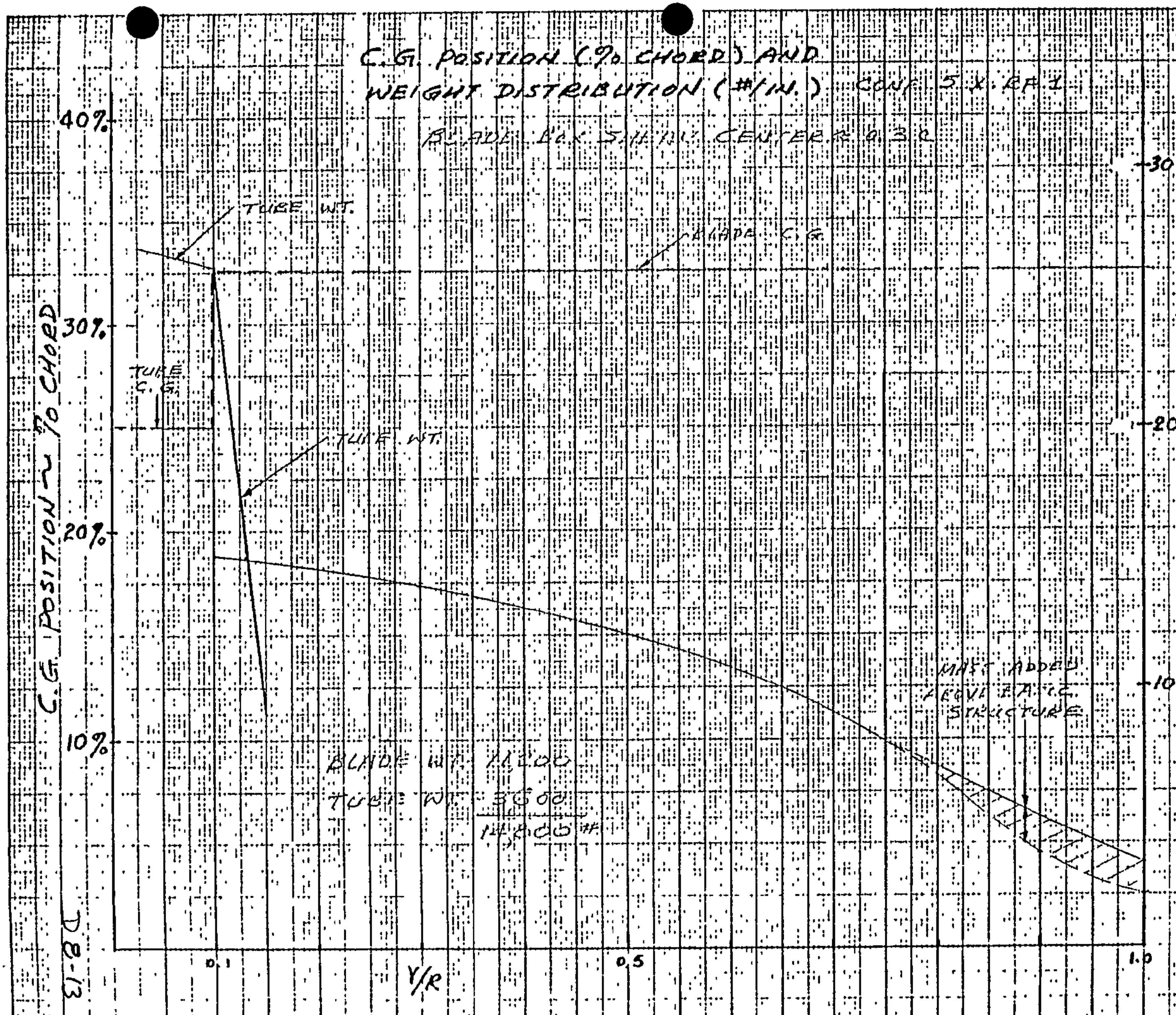
PAGE 4 OF 6
MODEL 5 X RF1
REPORT NO



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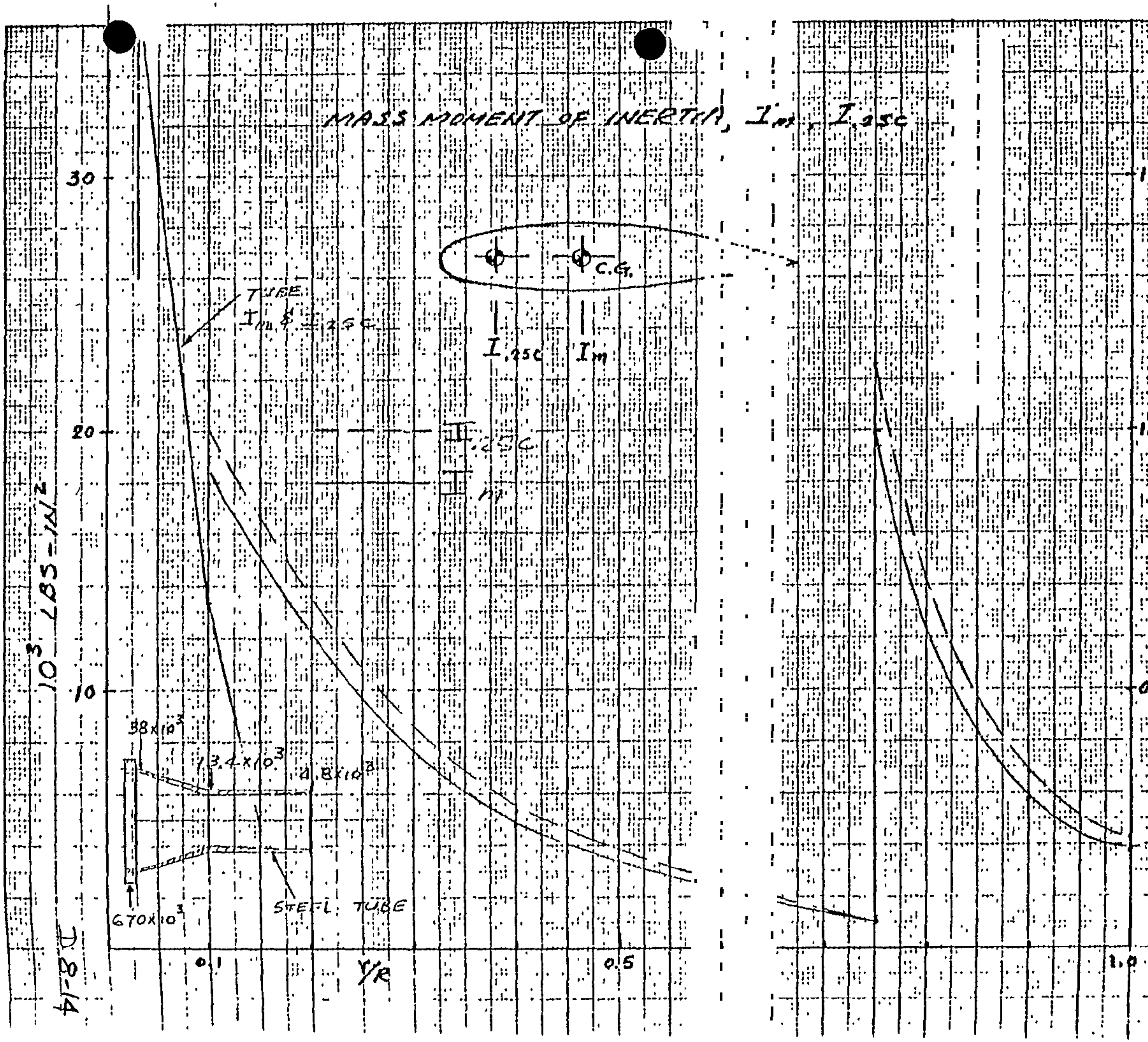
PAGE 5 OF 6
MODEL
REPORT NO



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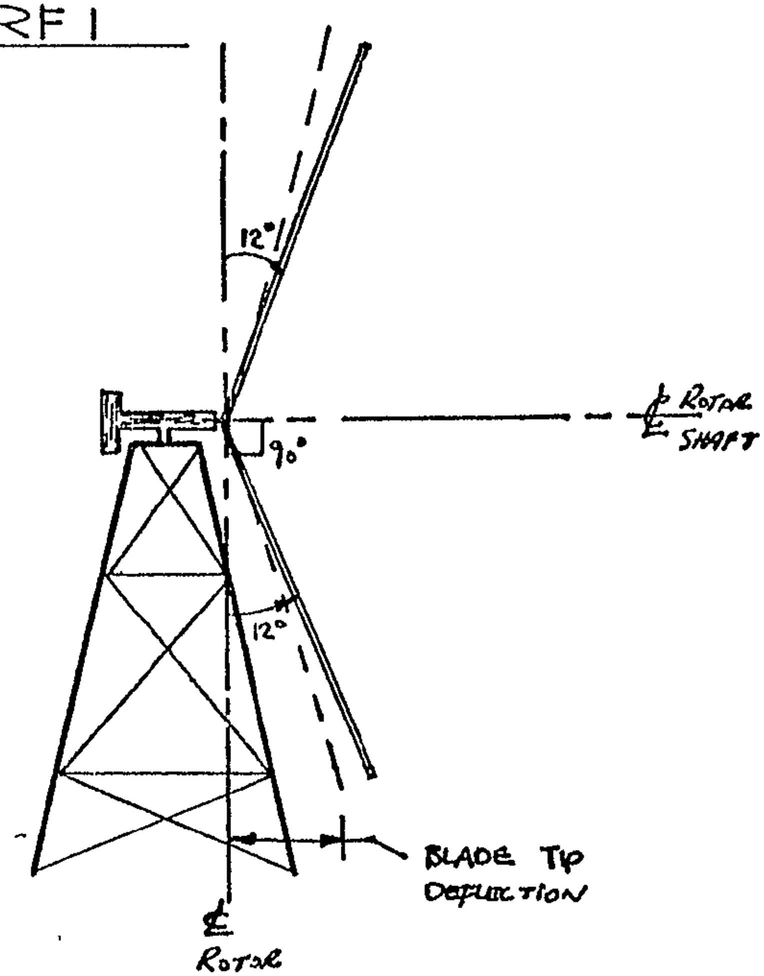
MODEL EX REL
REPORT NO.

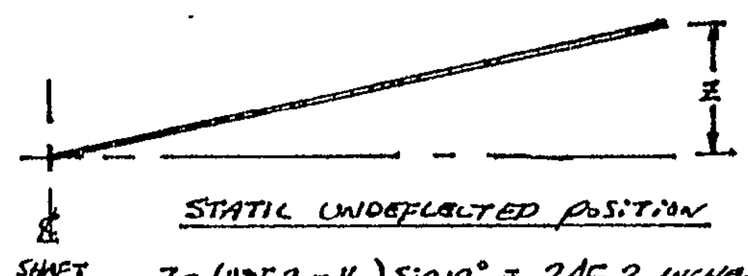
MOD I – METAL BLADE

CONFIGURATION 5.X.RF1

LOCKHEED CASE NO.	RPM	VEL. @ 30' (MPH)	$\Theta_{3/4R}$ (DEG.)	BLADE TIP DEFLECTION W/ RESPECT TO ℄ ROTOR ~ INCHES ~ MEAN	CYCLIC
101 (1A)	35	15	-5.6	198.3	± 6.1
102 (1B)	35	20	-4.7	206.5	± 8.1
103 (1C)	35	25	-4.7	212.5	± 10.2
104 (1D)	35	30	-10.1	202.3	± 12.1
105 (1E)	35	33	-12.4	198.9	± 13.3
106 (1F)	35	35	-13.7	197.1	± 13.9
204 (2D)	35	50.	-13.7	220.8	± 22.0
205 (2E)	35	20.	-13.7	177.8	± 6.7
301 (3A)	35	23.7	-90.	237.7	± 0.8
502 (5B)	0	120	0.	280.5	± 2.6

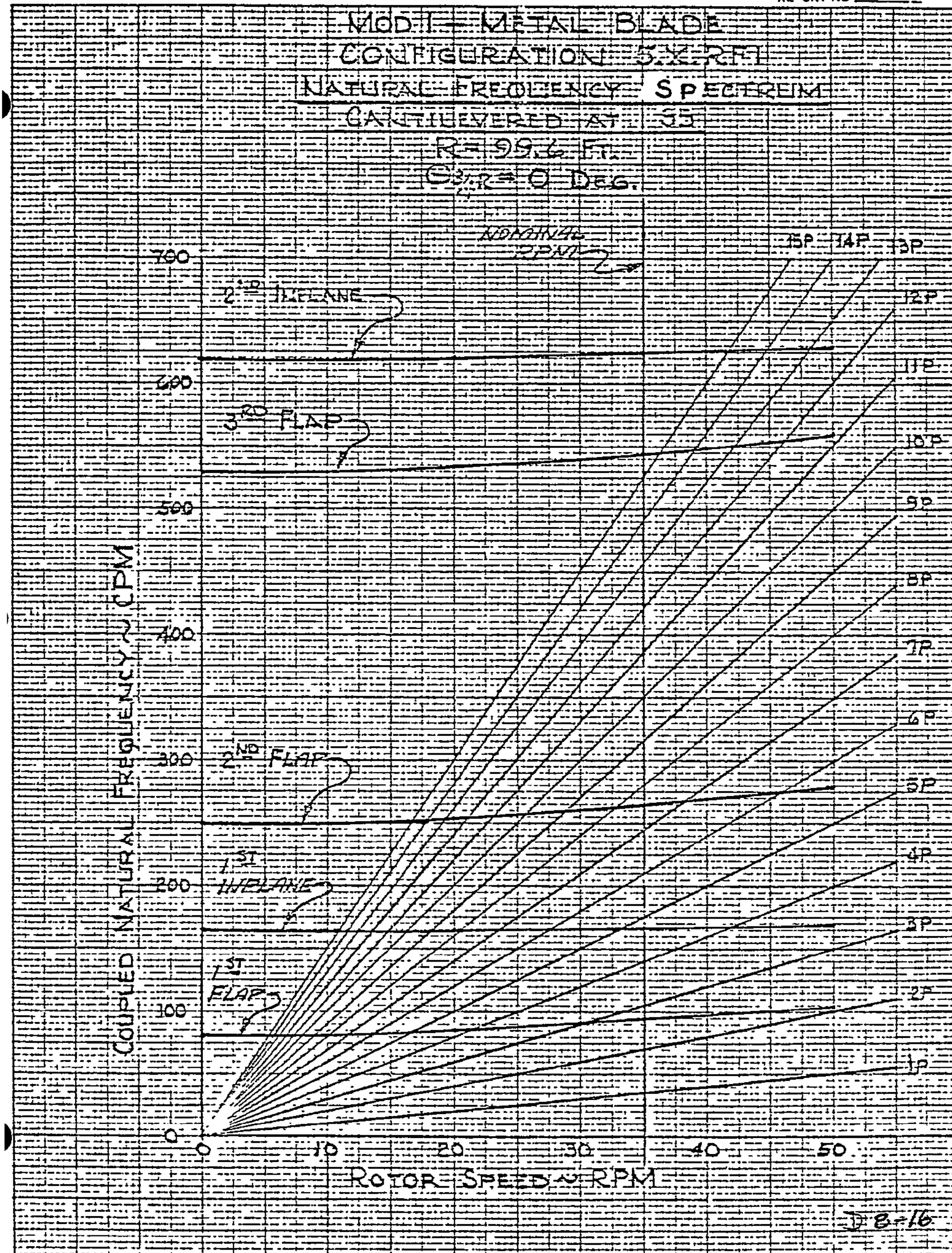


MAX. DISPLACEMENT TOWARD TOWER = 74.1 INCHES



STATIC UNDEFLECTED POSITION

$Z = (1135.2 - 16) \sin 12° = 245.2$ INCHES


MOD I – METAL BLADE
CONFIGURATION 5.X.RFI
NATURAL FREQUENCY SPECTRUM
CANTILEVERED AT .55
R = 99.6 FT.
$\Theta_{3/4R}$ = 0 DEG.
NOMINAL RPM
2ND INPLANE
3RD FLAP
2ND FLAP
1ST INPLANE
1ST FLAP
15P
14P
13P
12P
11P
10P
9P
8P
7P
6P
5P
4P
3P
2P
1P
0
100
200
300
400
500
600
700
COUPLED NATURAL FREQUENCY ~ CPM
0
10
20
30
40
50
ROTOR SPEED ~ RPM

D 8-16

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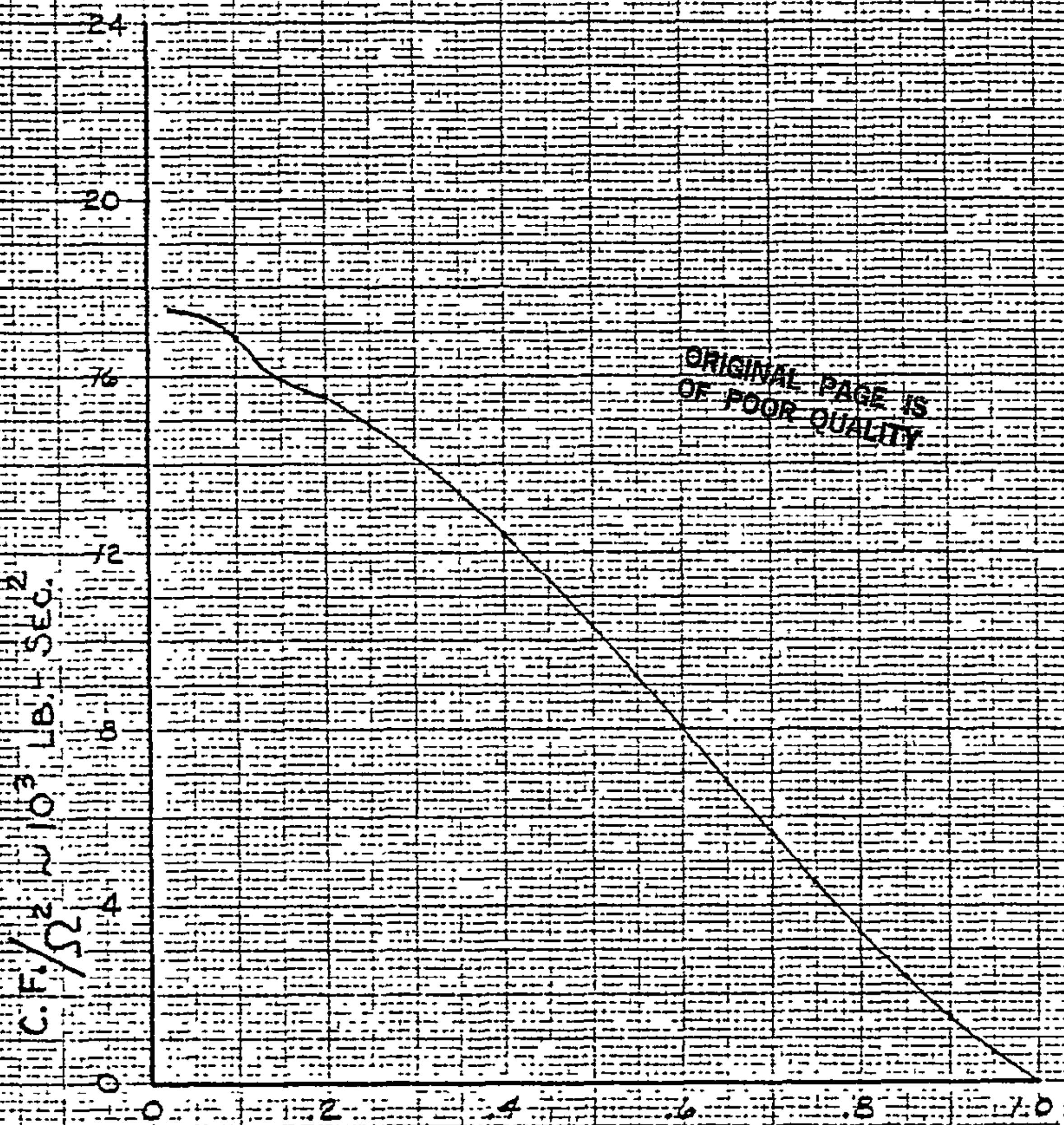
MOD I – METAL BLADE
CONFIGURATION 5.X.RF.1
CENTRIFUGAL FORCE PER Ω^2 DISTRIBUTION

R = 99.6 FT.
NOMINAL RPM = 35 (Ω = 3.6652 RAD/SEC.)

LIMIT



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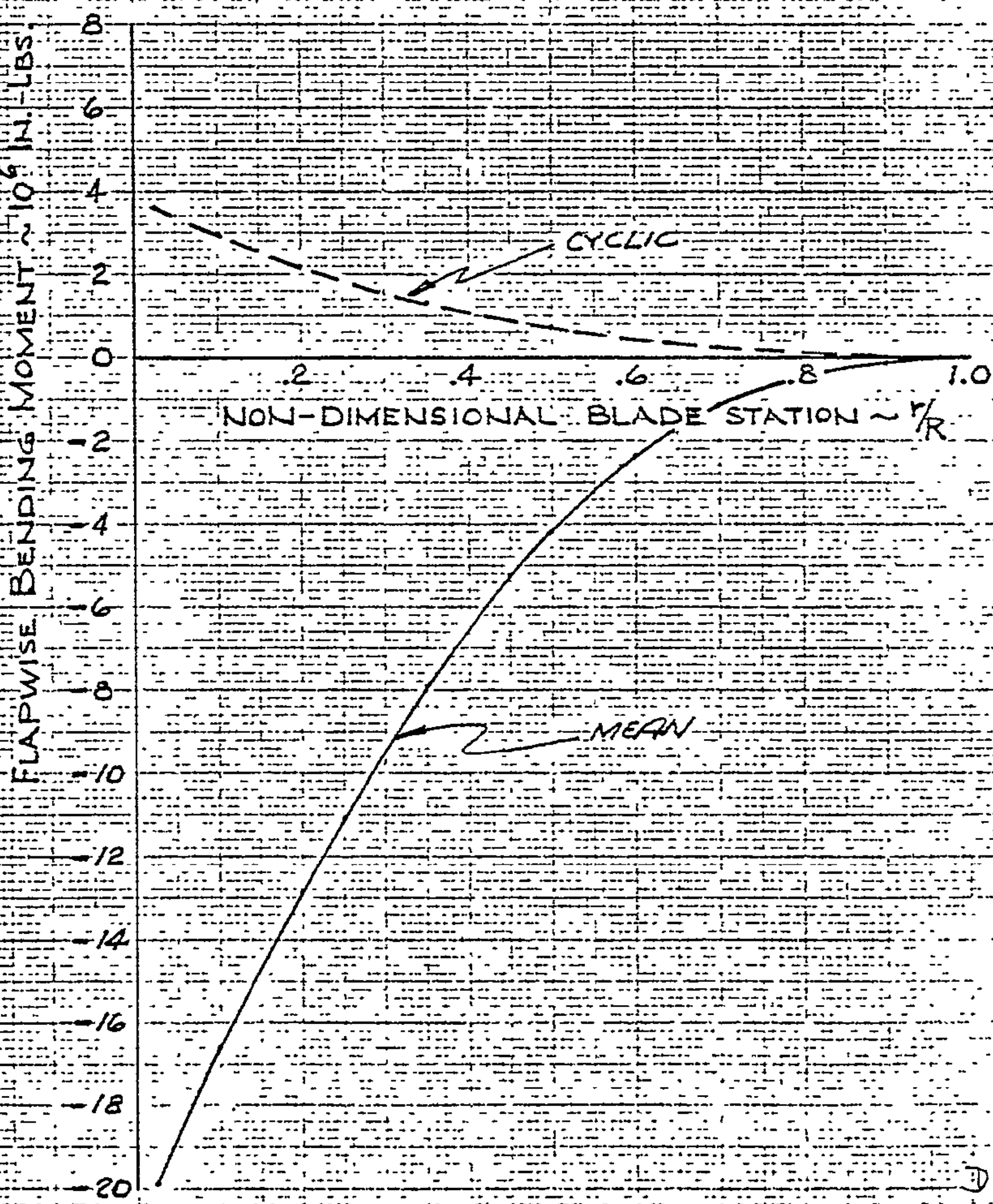
D-8-17

MOD 1 – METAL BLADE
CONFIGURATION 5.X.RF1
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 1A

LIMIT



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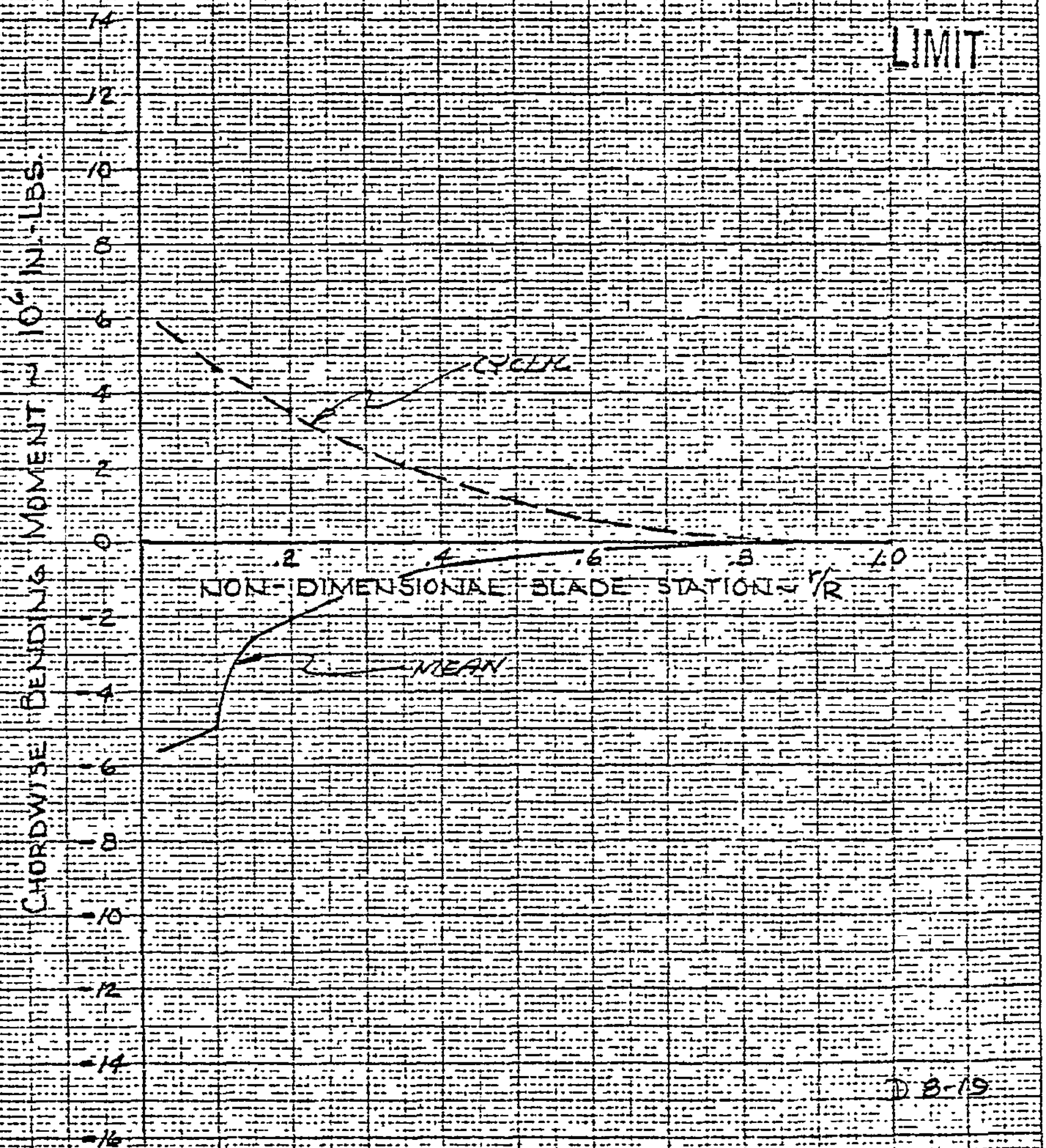
PAGE
MODEL
REPORT NO.

MOD-1 METAL BLADE
CONFIGURATION 5 X RF1
CHORDWISE BENDING MOMENT DISTRIBUTION

CASE 1A

LIMIT



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MODEL

REPORT NO.

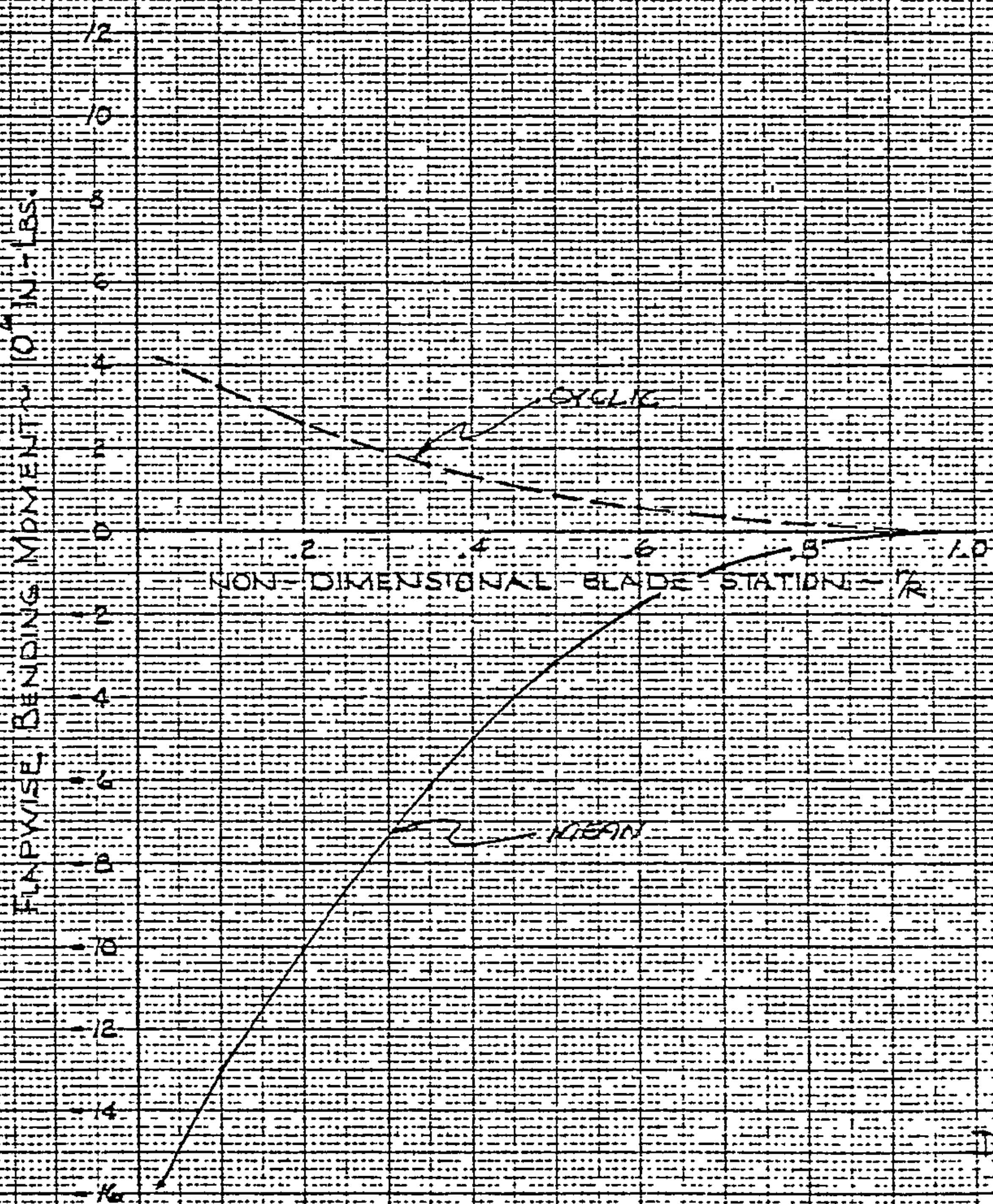
MOD I - METAL BLADE

CONFIGURATION 5.X.RF1

FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 1B

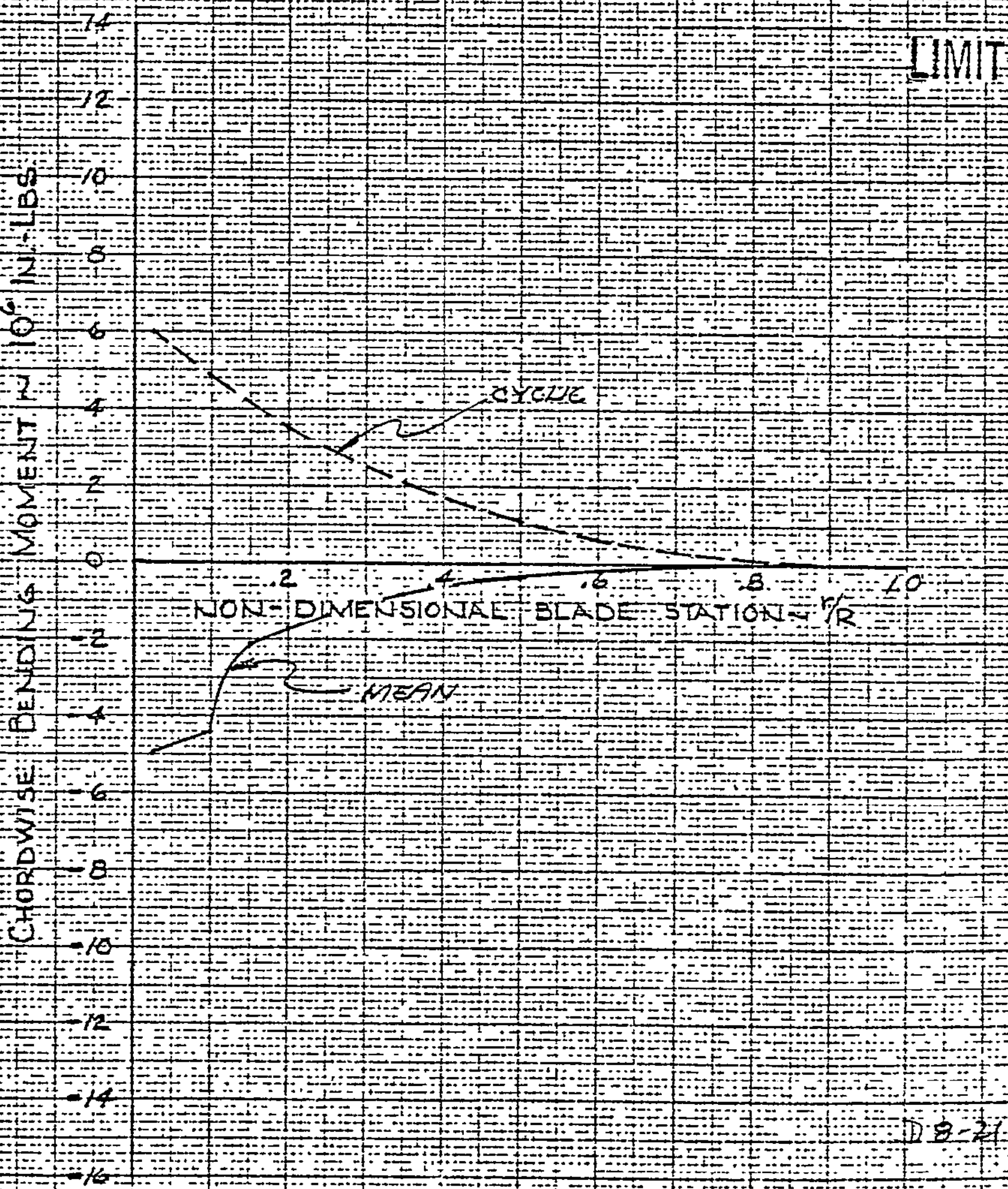
LIMIT



D 8-20

MOD 1 - METAL BLADE
CONFIGURATION 5.X RFI
CHORDWISE BENDING MOMENT DISTRIBUTION

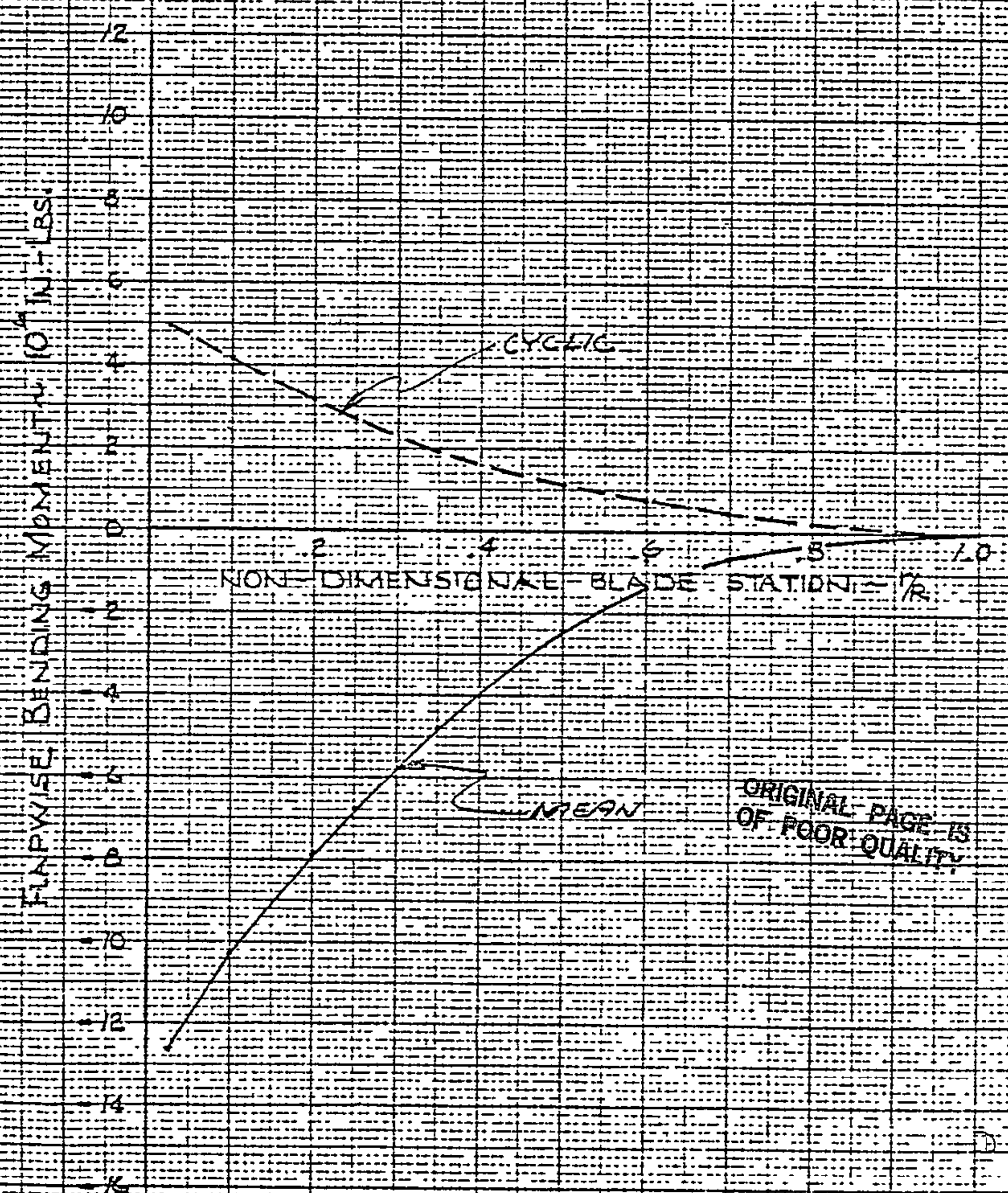
CASE 1B



MOD I - METAL BLADE
CONFIGURATION 5 X RF1
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 1C

LIMIT

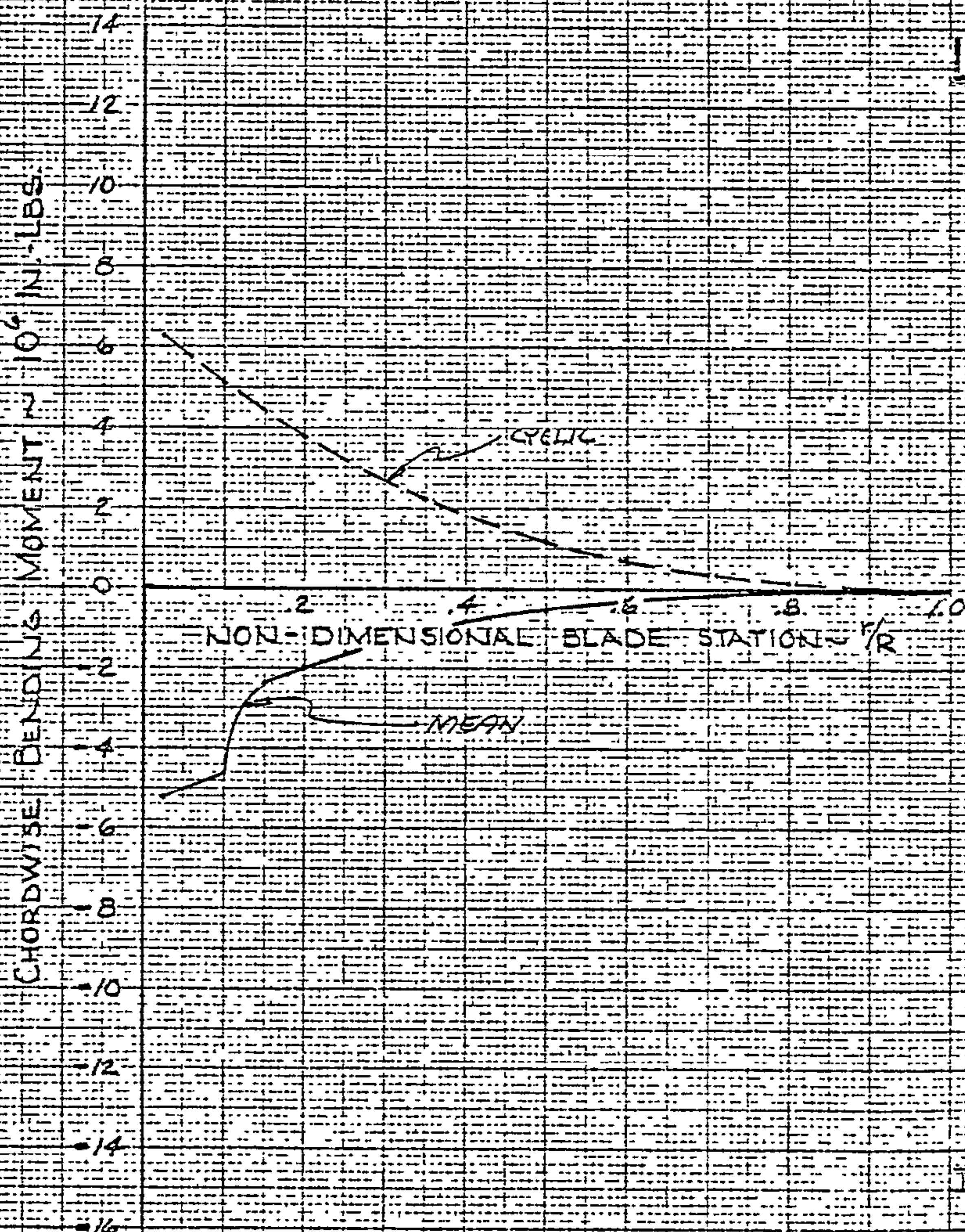


D 8-22

MOD I – METAL BLADE
CONFIGURATION 5 X RFI
CHORDWISE BENDING MOMENT DISTRIBUTION

CASE 1C

LIMIT



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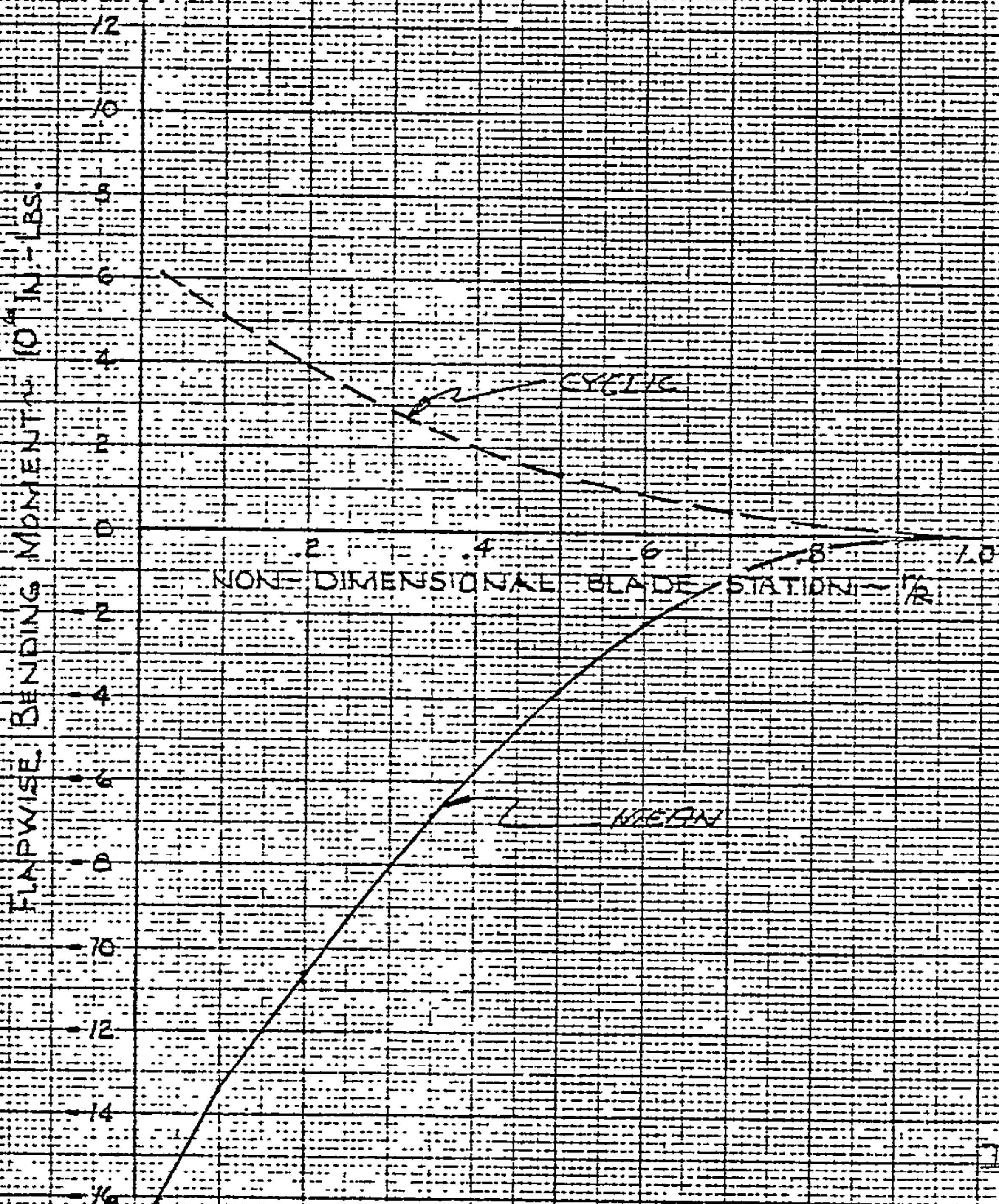
MODEL
REPORT NO.

MOD I - METAL BLADE
CONFIGURATION 5.X.RF.1
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 1D

LIMIT

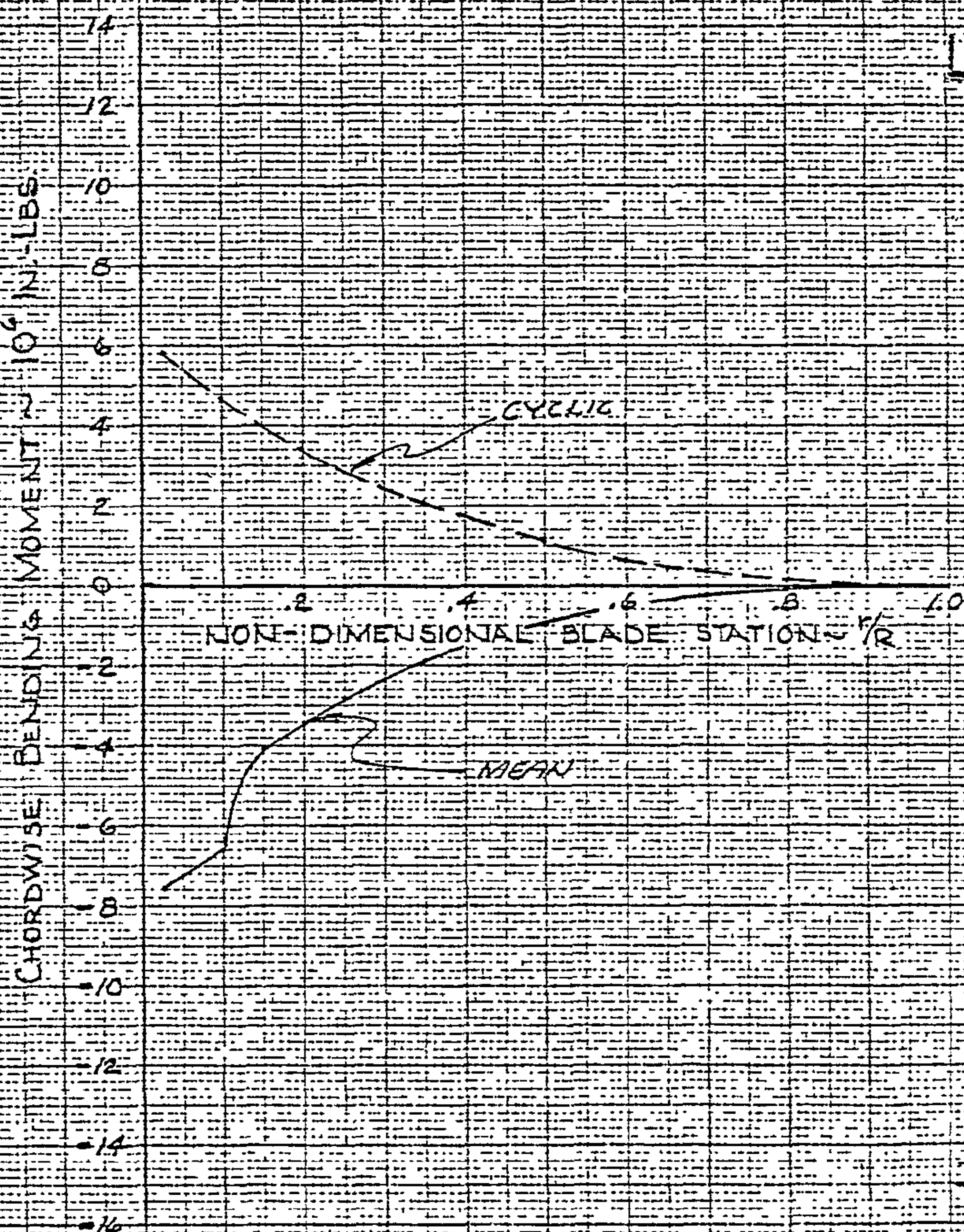


D 8-24

MOD 1 - METAL BLADE
CONFIGURATION 5.X.RF1
CHORDWISE BENDING MOMENT DISTRIBUTION

CASE 1D

LIMIT



D 8-25

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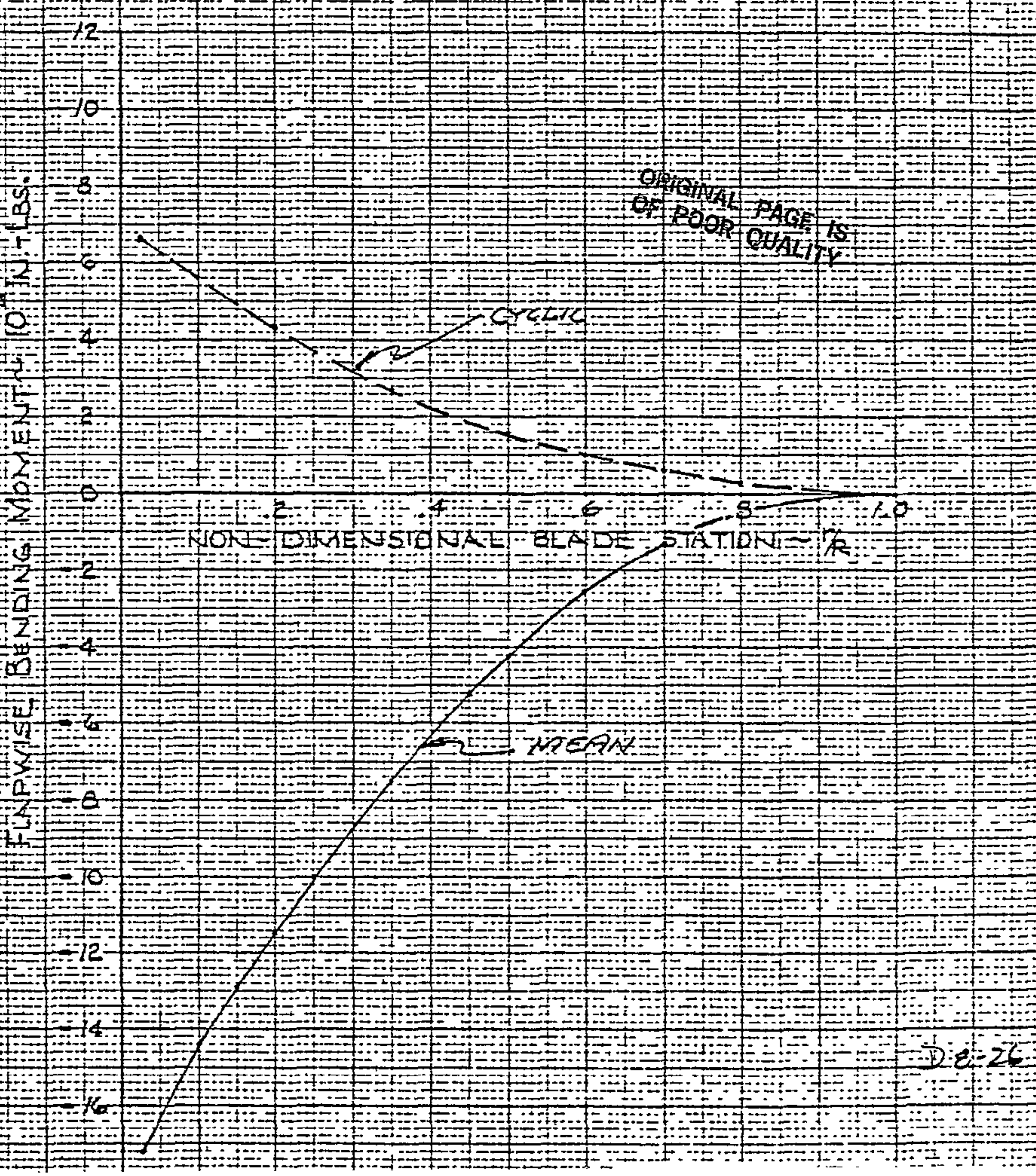
MODEL

REPORT NO

MOD I - METAL BLADE
CONFIGURATION 5.X.RF1
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 1E

LIMIT



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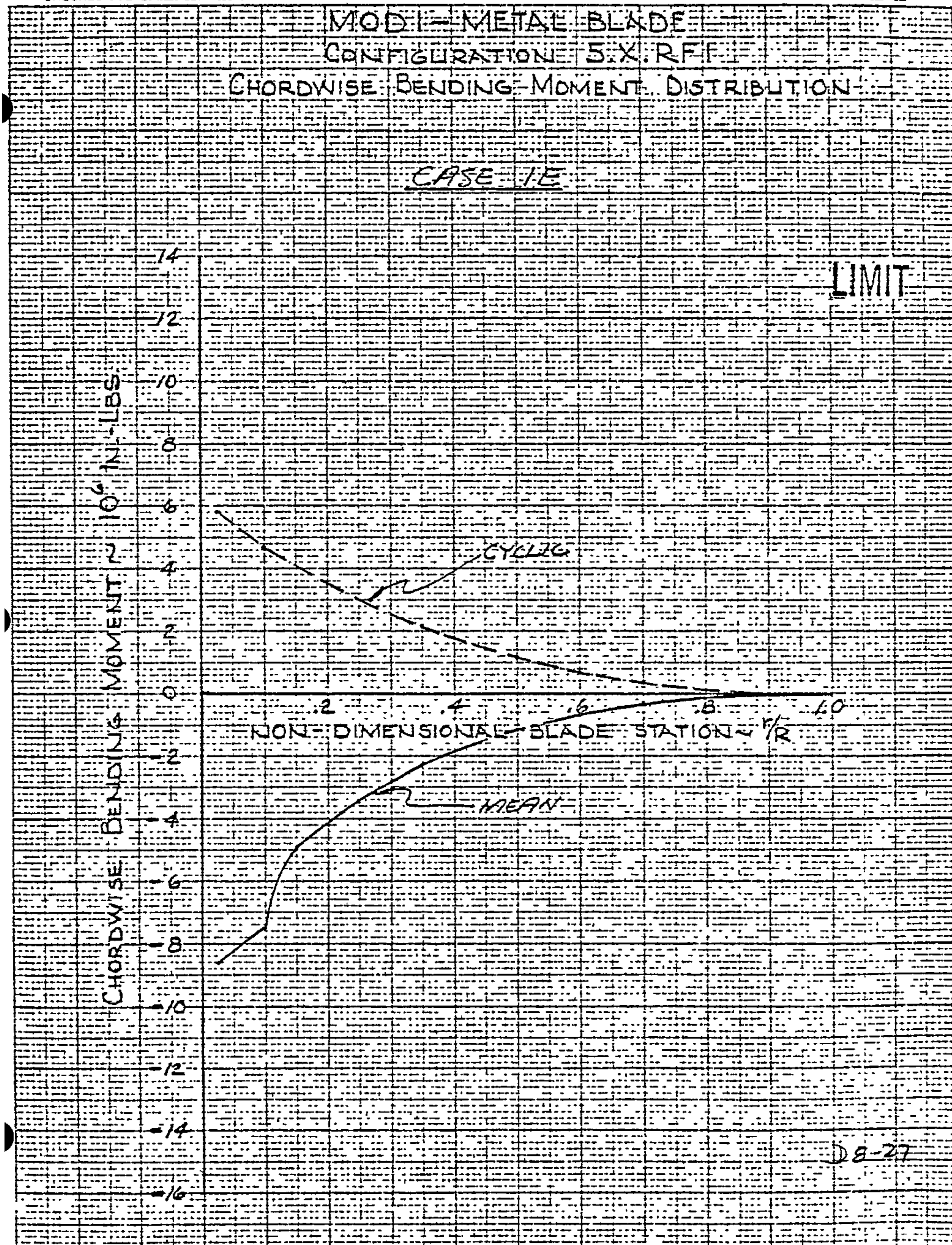
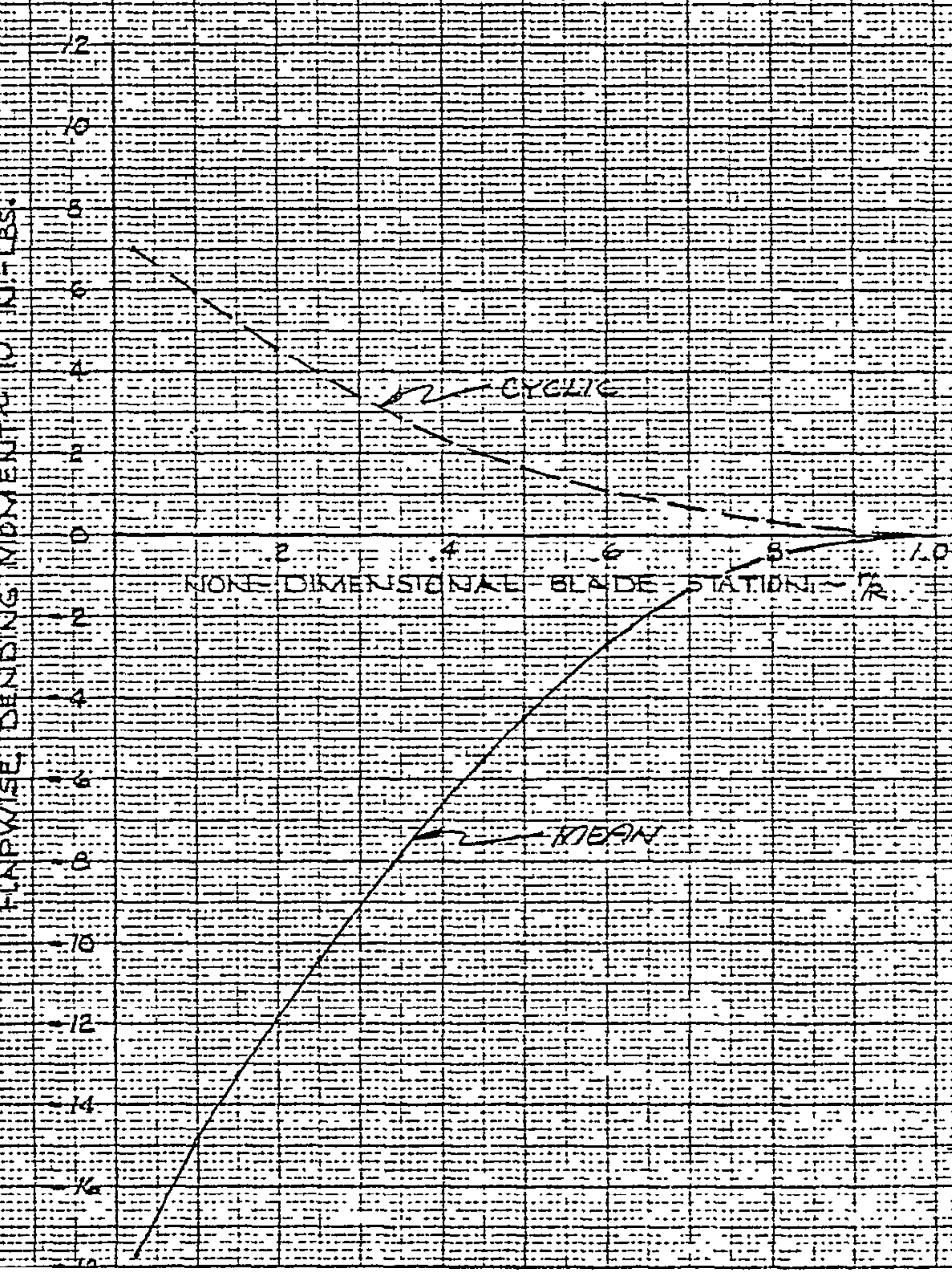


MOD 1 - METAL BLADE
CONFIGURATION 5.X.RF1
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE IF

LIMIT



D 8-28

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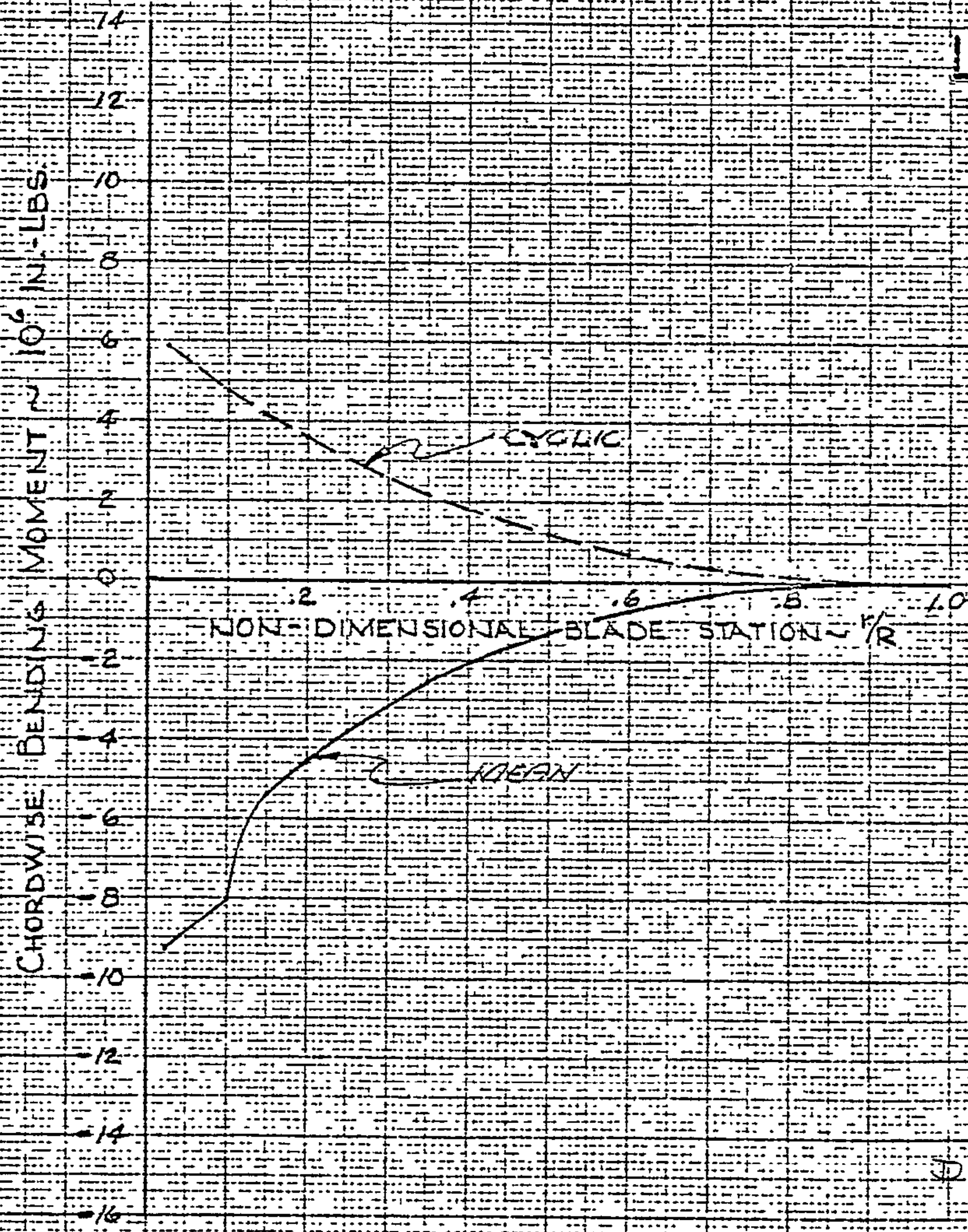
MODEL
REPORT NO.

MOD I - METAL BLADE
CONFIGURATION 5 X RF1
CHORDWISE BENDING MOMENT DISTRIBUTION

CASE 1F

LIMIT

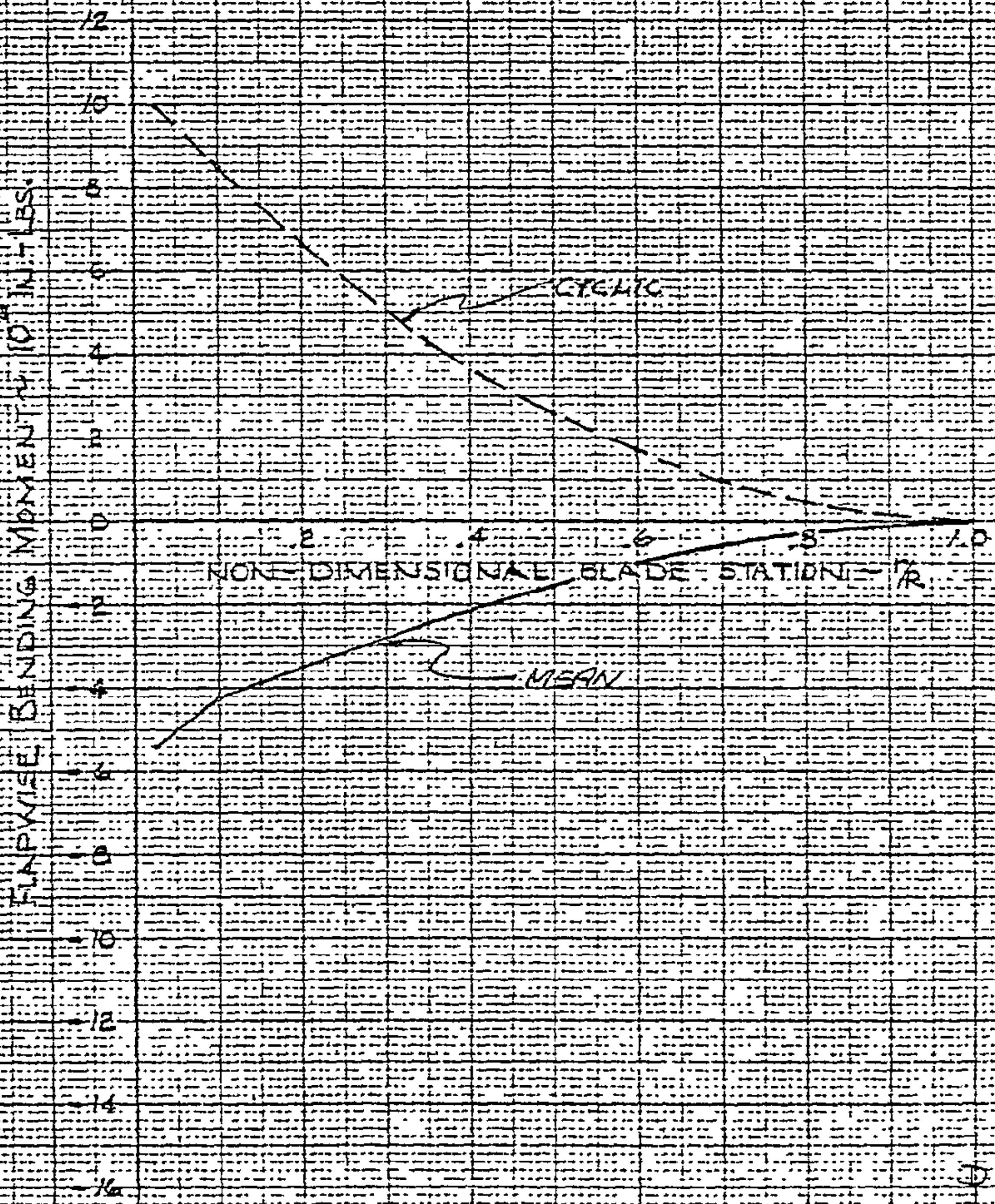


D 8-29

MOD I - METAL BLADE
CONFIGURATION 5.X.RF1
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 2D

LIMIT

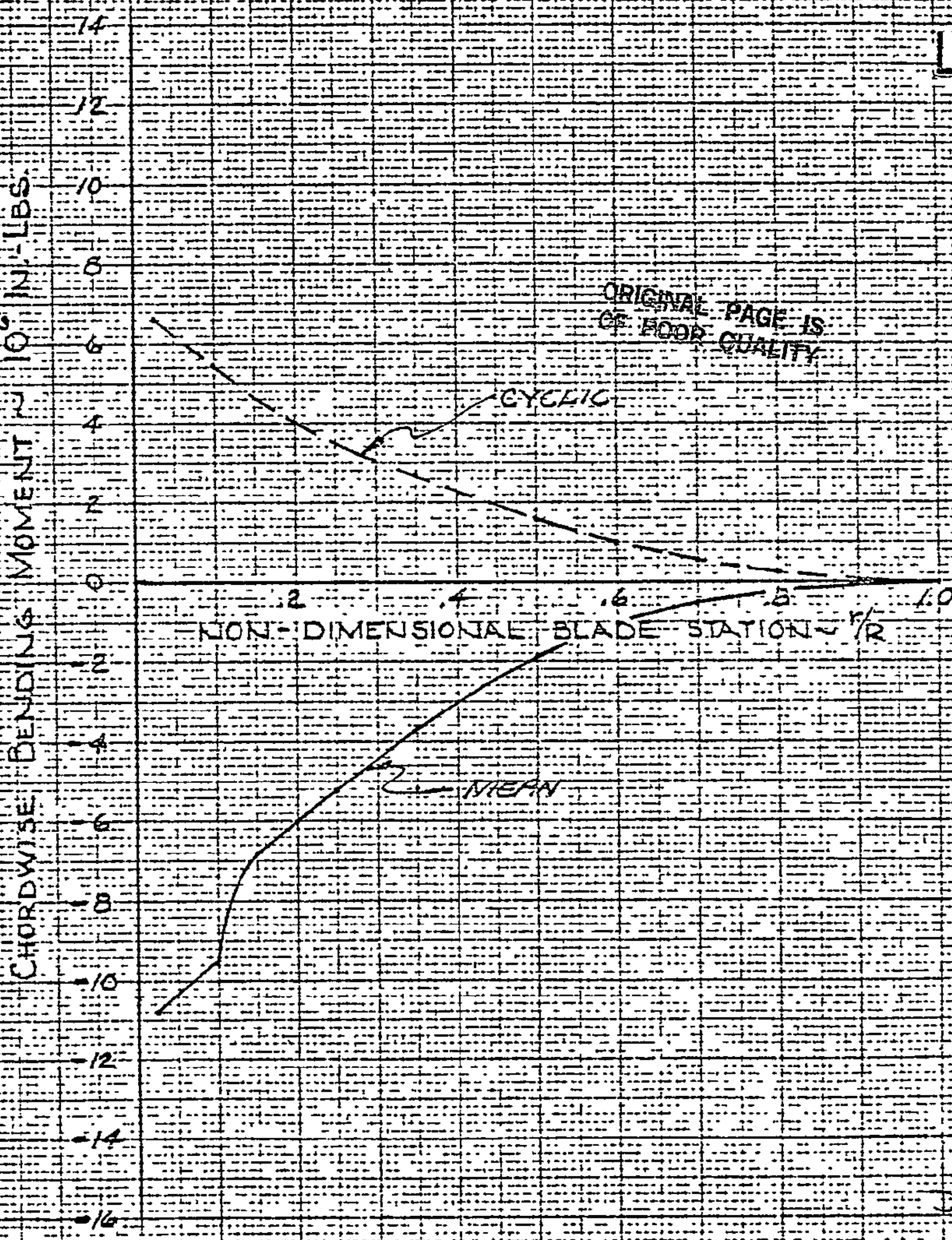


D8-30

MOD I - METAL BLADE
CONFIGURATION 5.X.RF1
CHORDWISE BENDING MOMENT DISTRIBUTION

CASE 2D

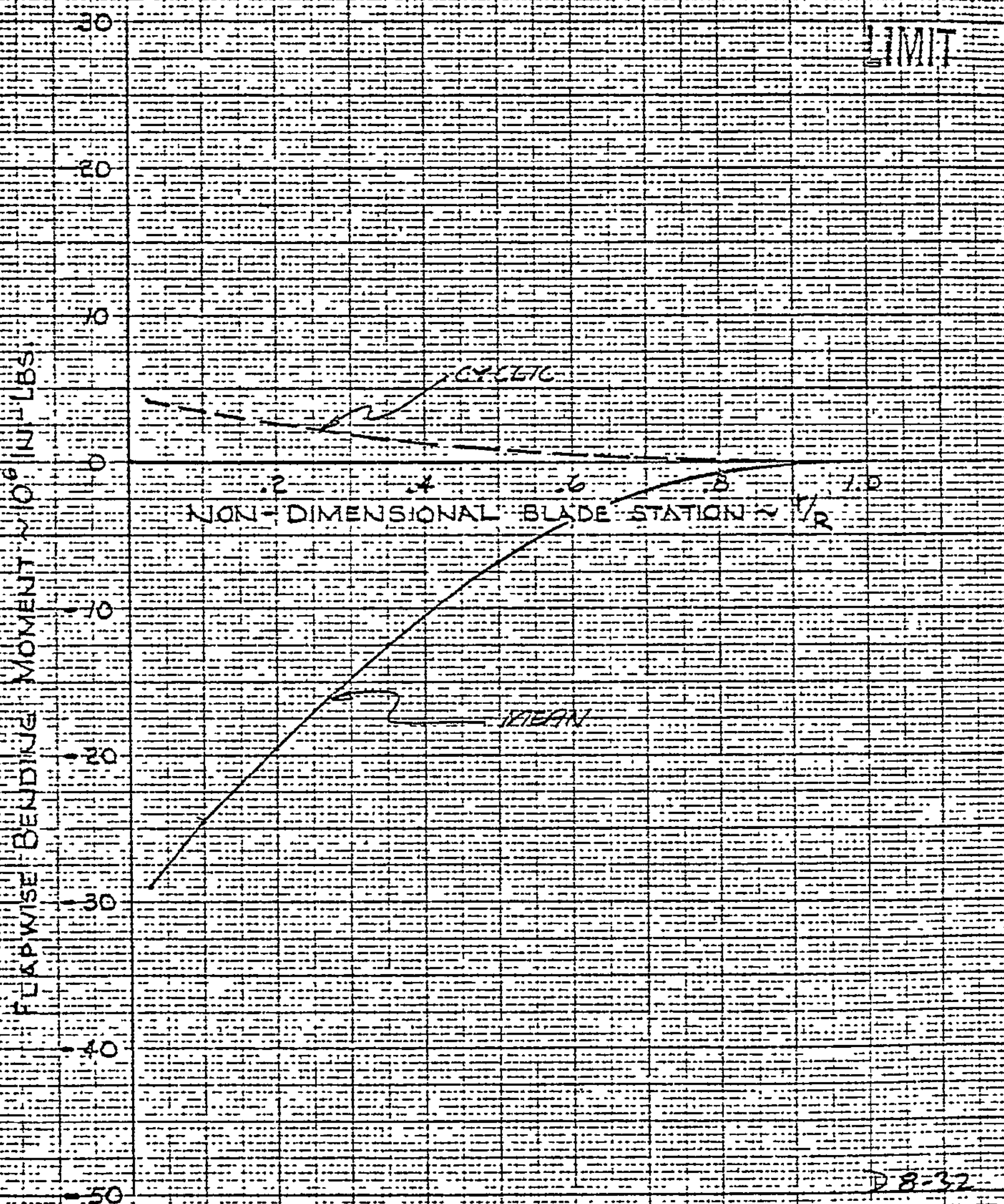
LIMIT



MOD I – METAL BLADE
CONFIGURATION 5.X.RF
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 2E

LIMIT



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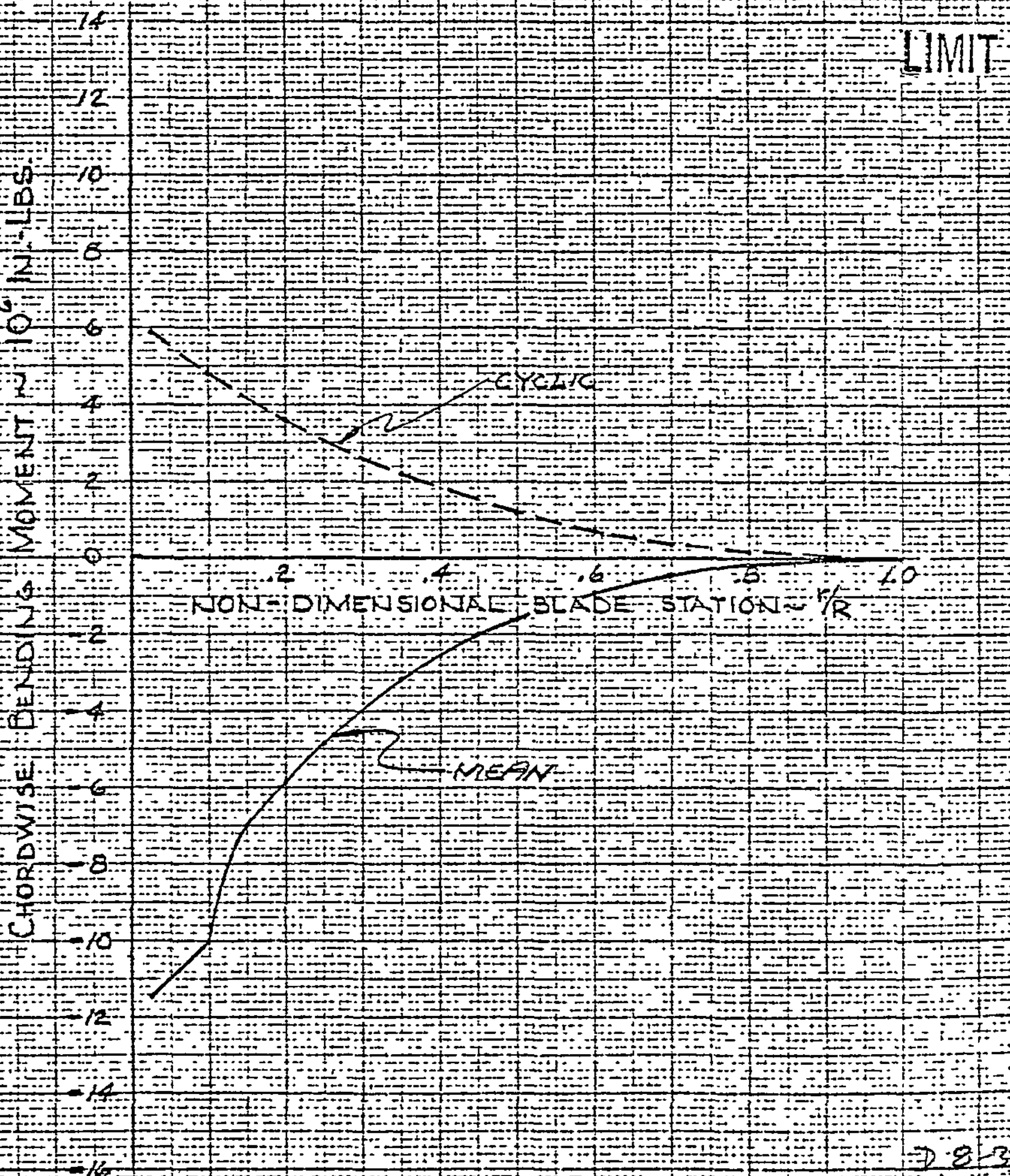
MODEL
REPORT NO.

MOD I - METAL BLADE
CONFIGURATION 5.X.RF1
CHORDWISE BENDING MOMENT DISTRIBUTION

CASE 2E

LIMIT



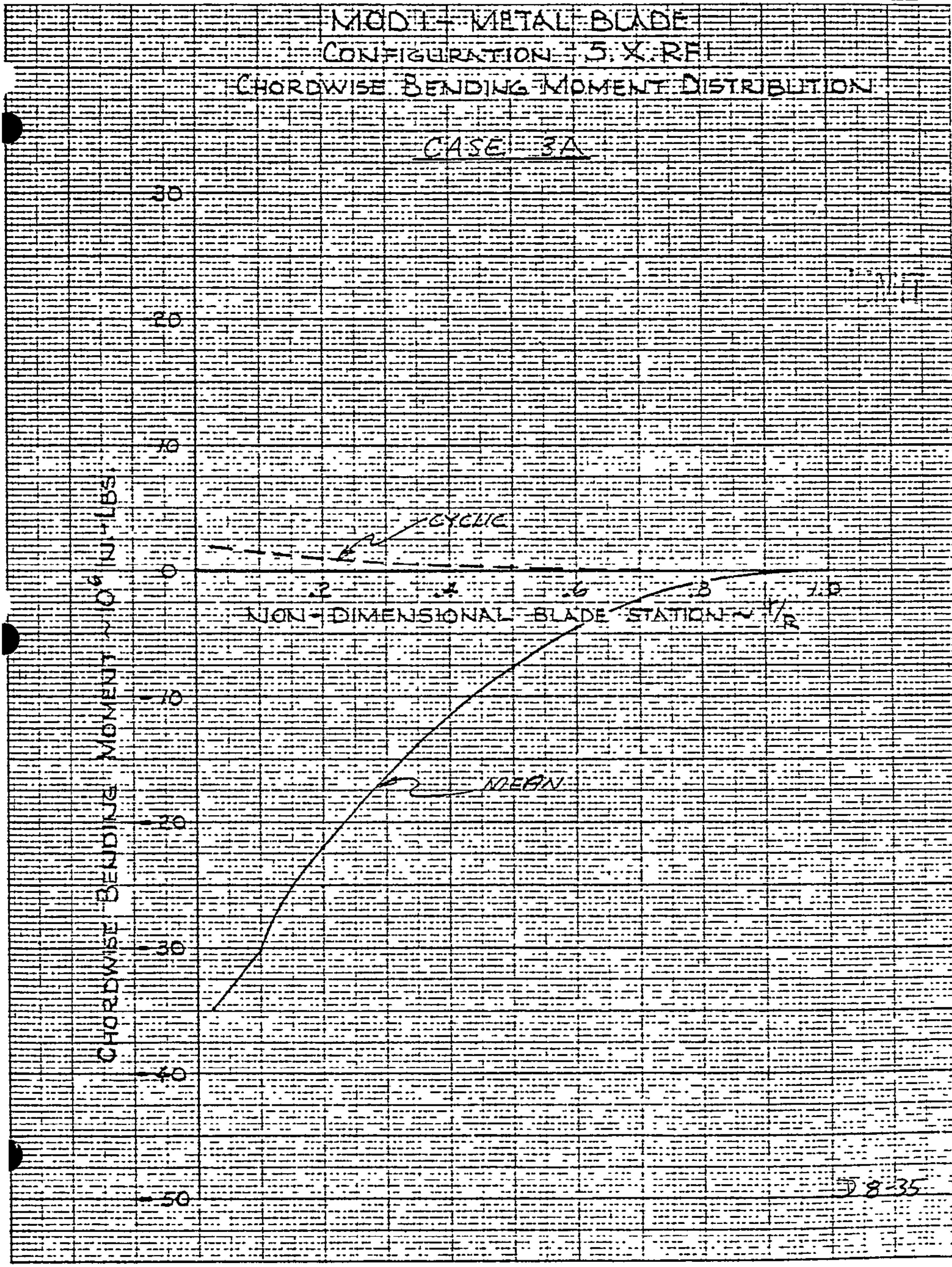
D 8-33

MOD I - METAL BLADE
CONFIGURATION 5.X.RF.I
FLAPWISE BENDING MOMENT DISTRIBUTION

CASE 3A

LIMIT

D 8-34


MOD I - METAL BLADE
CONFIGURATION 5.X.RFI
CHORDWISE BENDING MOMENT DISTRIBUTION
CASE 3A
30
20
10
0
-10
-20
-30
-40
-50
CYCLIC
MEAN
.2
.4
.6
.8
1.0
NON-DIMENSIONAL BLADE STATION ~ r/R
CHORDWISE BENDING MOMENT ~ 10^6 IN-LBS

D 8-35

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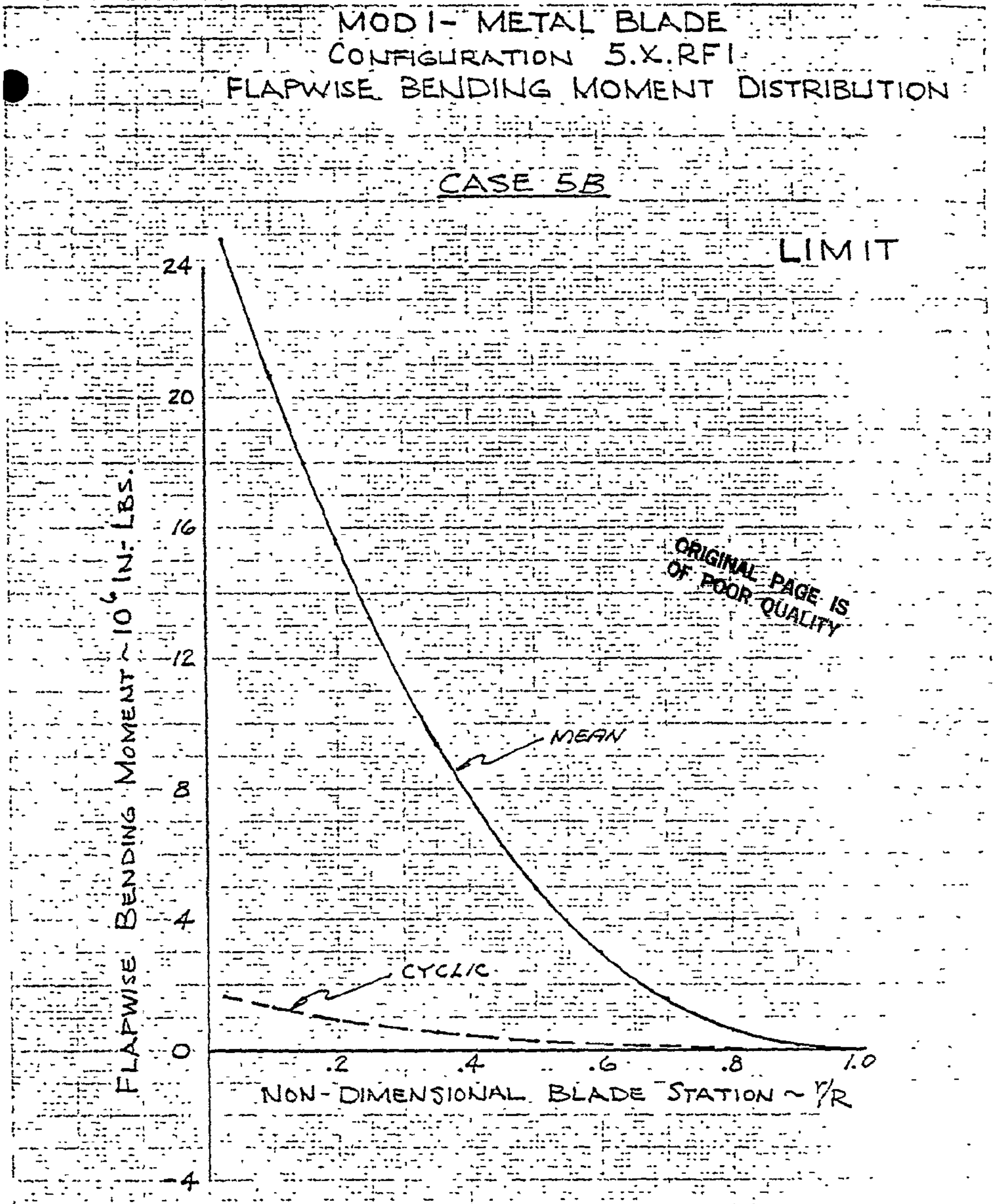
MODEL
REPORT NO



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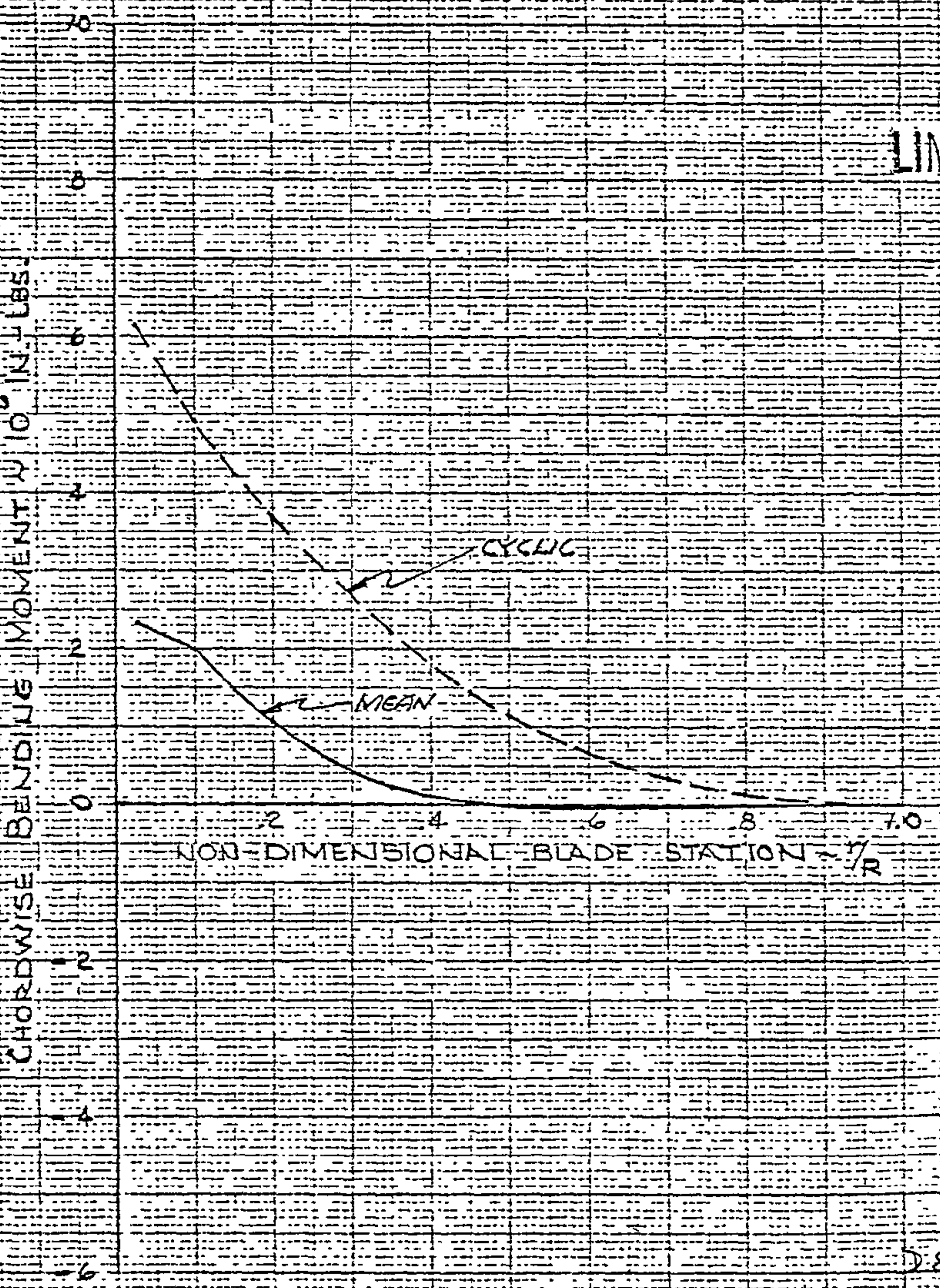
D 8-36

MOD I - METAL BLADE
CONFIGURATION 5.X.RFI
CHORDWISE BENDING MOMENT DISTRIBUTION

CASE 5B

LIMIT



APPENDIX E

PRELIMINARY BLADE DESIGN REVIEW
(COMPOSITE BLADE)

Abstract

This appendix is a report on composite blade development performed by Hamilton-Standard, consisting of copies of the slides that were used in a presentation at NASA-Lewis Research Center. This report was originally published as G. E. Space Division document No. 77SDS 4217, dated March 8, 1977.

PROGRAM SUMMARY
1500 kW WIND TURBINE GENERATOR
PRELIMINARY BLADE DESIGN REVIEW

INTRODUCTION

The Preliminary Blade Design herein described was prepared by the Advanced Energy Programs organization of the General Electric Company's Space Division under Contract NAS 3-20058",1500 kW Wind Turbine Generator Program". This program is being directed by the NASA Lewis Research Center's Wind Power office for the Energy Research and Development Administration, and is an integral part of the Federal Wind Energy Program.

The objective of this contract is to design, fabricate, assemble, install and checkout a 1500 kW wind turbine that generates electricity and delivers it into a utility network at costs competitive with alternative energy sources. Also, this wind turbine is to be designed for safe reliable operation over a period of thirty years, and it is to be compatible with utility interface requirements and general utility operations and maintenance practices. The contractor is to select the design, fabrication, assembly and installation options available throughout the contract that will best meet the above objectives, including the utilization of a management plan and work plan that will allow him to closely follow the schedule as agreed upon.

This project consists of the following nine tasks:

Task I: Requires all program management functions necessary to fulfill the objective of this contract.

Task II: Requires the design analyses and the preliminary design of the wind turbine system.

Task III: Requires the detail design of the wind turbine system, cost estimates of additional units, and the design of tooling and shipping containers.

Task IV: Requires definition of the instrumentation, data acquisition, and testing requirements for this project.

Task V: Requires the procurement and fabrication of the complete wind turbine system.

Task VI: Requires the site preparation necessary for interfacing the wind turbine with the user's existing system.

Task VII: Requires the assembly, installation, and operational checkout of the wind turbine at the site.

Task VIII: Requires the procurement of a second 1500 kW wind turbine system.

Task IX: Requires all reporting for this contract.

As the Prime Contractor, General Electric has total responsibility for each of these tasks The Hamilton Standard Division of United Technologies, under subcontract to GE, is designing and fabricating the rotor system which includes the blades, hub and pitch change mechanism.

REQUIREMENTS/CHARACTERISTICS SUMMARY

	SYSTEM REQUIREMENTS	BLADE REQUIREMENTS
POWER	1500 KW	1670 KW
RATED WIND SPEED	22 MPH	
CUT-IN/CUT-OUT WIND SPEED	11/50 MPH	
MAXIMUM WIND SPEED (SURVIVAL)	150 MPH	
ROTORS PER TOWER	1	
LOCATION OF ROTOR	DOWNWIND	
BLADES PER ROTOR	2	
CONE ANGLE	OPTIONAL	$>10^{\circ}$
ROTOR INCLINATION	$<15^{\circ}$	0°
ROTOR SPEED CONTROL	BLADE PITCH	
ROTOR SPEED	OPTIONAL	35 RPM
BLADE DIAMETER	$>$200 FEET	
AIRFOIL SECTION	L/D $>$ 50	
BLADE TWIST	OPTIONAL	
BLADE TIP CLEARNACE	50 FEET	
YAW RATE	$< 2^{\circ}$/SEC	0.25°/SEC
SYSTEM LIFE	30 YEARS	

ROTOR REQUIREMENTS REFLECT NASA
SPECIFICATION WITH ADDITIONAL CONSTRAINTS

BLADE DESIGN REQUIREMENTS

- ENVIRONMENTAL PROTECTION (SALT-SPARY, SAND, DUST)
- BALANCE PROVISIONS
- CONDUCTIVE PATH FOR LIGHTNING STRIKES
- STRAIN GAGE PROVISIONS
- TWIST AND TAPER TO CONSIDER COST, STRUCTURAL EFFICIENCY, FABRICATION, DURABILITY
- WEIGHT WITHIN 1 PERCENT

STABILITY REQUIREMENTS

- CHORDWISE CG POSITION TO PRECLUDE DIVERGENCE

- TORSIONAL FREQUENCY PLACEMENT TO PRECLUDE FLAP-PITCH OR FLAP-LAG-PITCH INSTABILITY

- ESTABLISHED CRITERIA AND EXPERIMENTAL DATA TO EVALUATE STALL FLUTTER AND PANEL FLUTTER

NASA STABILITY REQUIREMENTS
IMPOSED TO ASSURE BLADE STABILITY

BLADE TUNING REQUIREMENTS

NASA SPECIFICATION

- FUNDAMENTAL FLAP: $> 2.15P$
- FUNDAMENTAL CHORD: $> 4.15P$
- FUNDAMENTALS NOT COINCIDENT WITH EVEN ORDERS

GE SPECIFICATION

- FUNDAMENTAL FLAP: $2.15 < f < 2.7P$
- FUNDAMENTAL CHORD: $4.4P < f < 4.7P$
- HIGHER BLADE FREQUENCIES DIFFER FROM n ORDER BY 0.3 ($n \leq 6$)
- INCLUDE HUB IMPEDENCE IN BLADE FREQUENCY CALCULATION

BLADE TUNING NARROWED TO
PRECLUDE ADVERSE DYNAMIC COUPLING

BLADE LOAD CONDITIONS

CASE 1	STEADY WIND AT 22 MPH, 35 RPM, 1670 KW POWER, (10^8 CYCLES)
CASE 2	GUST 22 MPH TO 60 MPH IN 0.25 SECOND, 25 PERCENT OVERSPEED, (10^5 CYCLES)
CASE 3	EMERGENCY FEATHER IN 11 SECONDS, 22 MPH (PROPORTIONAL LIMIT)
CASE 4	GUST 22 MPH TO 0 MPH (10^5 CYCLES)
CASE 5	HURRICANE 120 MPH WIND, LOCKED AND FEATHERED (PROPORTIONAL LIMIT)
CASE 6	INFLOW OF 20^0 AT 50 MPH WITH 2^0/SEC YAW RATE (10^5 CYCLES)
CASE 7	STEADY 50 MPH WITH 50 PERCENT TOWER SHADOW (10^5 CYCLES) • GE IMPOSED 35% TOWER SHADOW

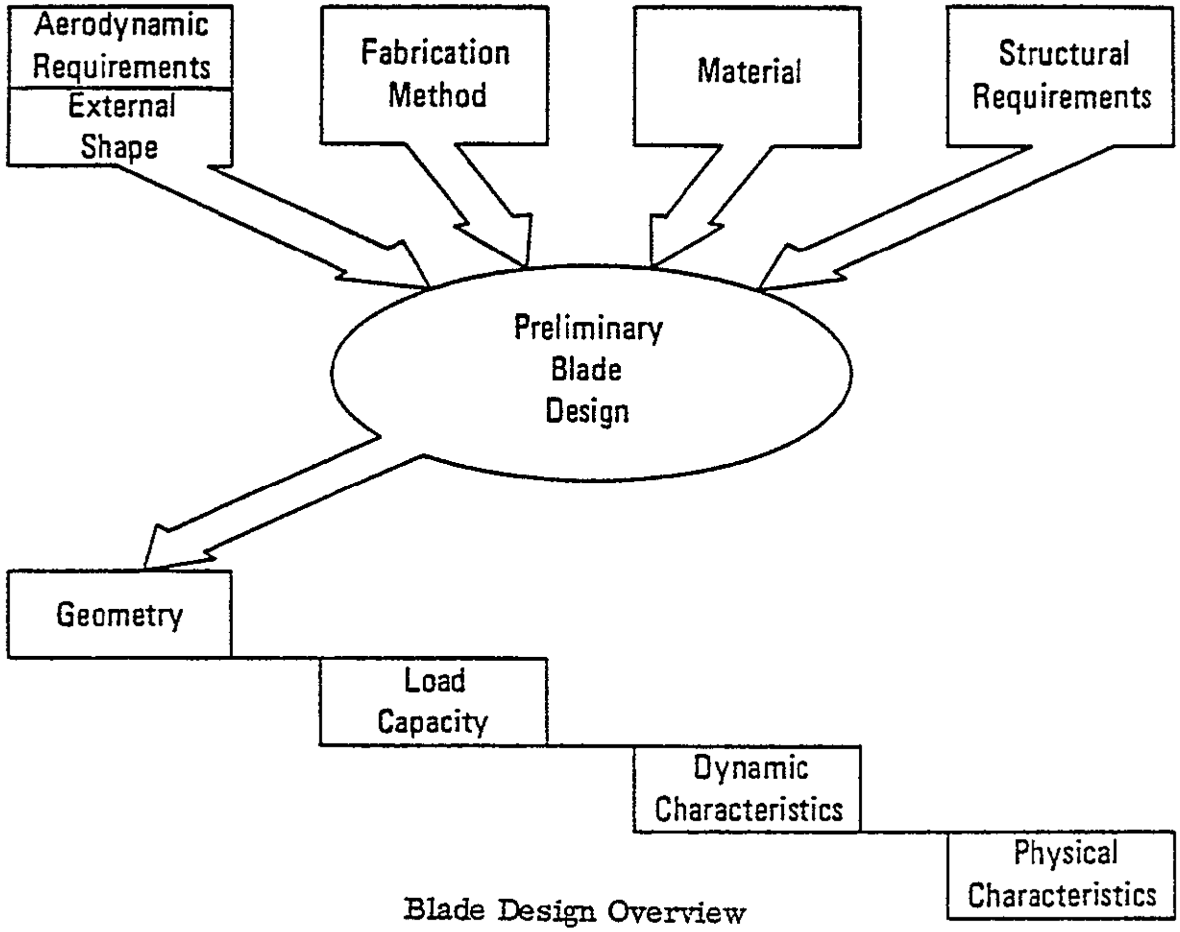
BLADE REQUIREMENT CHANGES UNDER CONSIDERATION

- CORRELATION FACTOR ON BLADE LOADS

- MODIFIED GUST - CASE 2

  2(a) INFREQUENT GUST (PROPORTIONAL LIMIT) 22 MPH TO 40 MPH, 7-SECOND PERIOD

  2(b) FREQUENT GUST (10^5 CYCLES) 22 MPH TO 35 MPH, 15-SECOND PERIOD



Blade Design Overview

General Requirements

Structural

- Load cases from Exhibit B, Para. 2.1.2
- Supplementary load cases
- Stability cases from Exhibit A, Para. 2c
- Critical speed placement from Exhibit B, Para. 2.1.3
- Stall flutter from Exhibit A, Para. 2f
- Divergence from Exhibit A, Para. 2e

Environmental

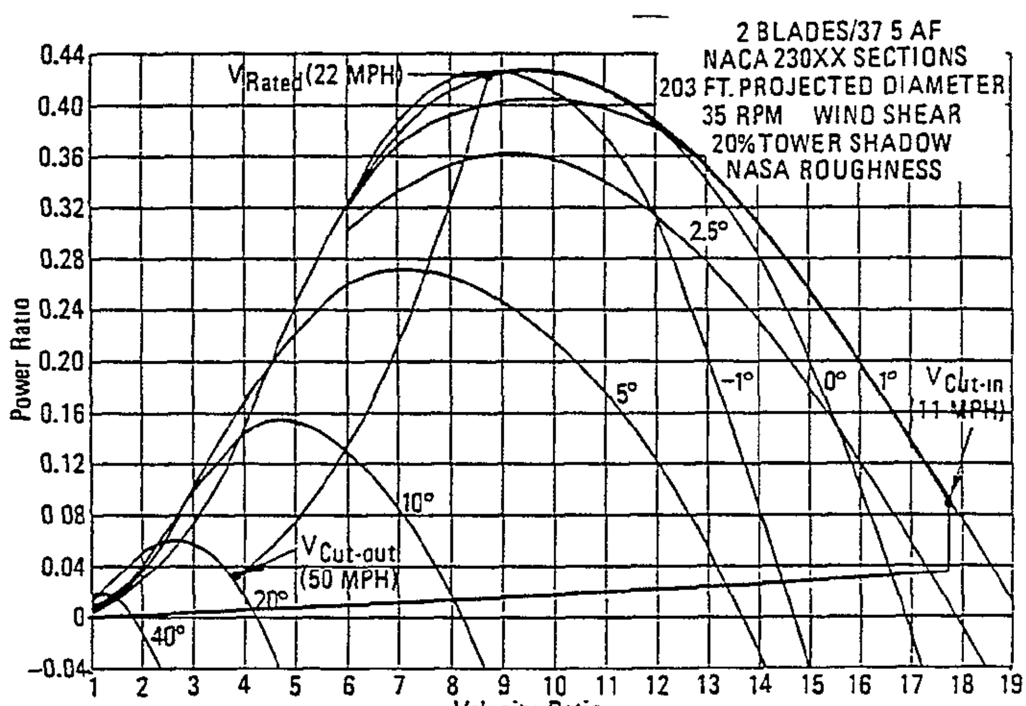
- Life
- Lightning

Geometric

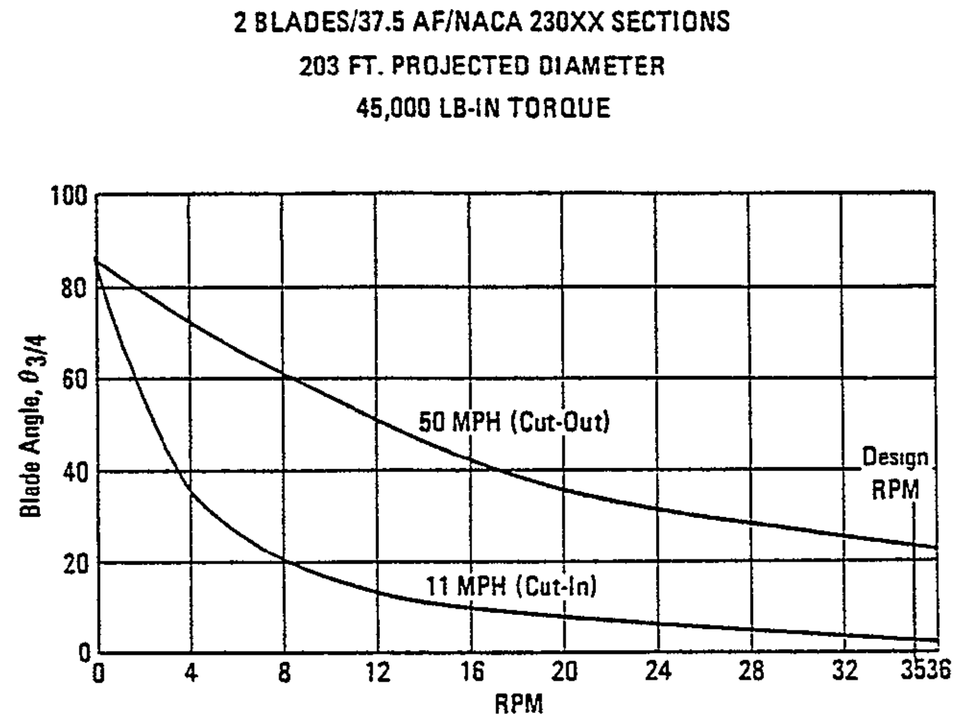
- Shape
- Weight
- Balance

Specifications Mod-1 Wind Turbine

- Output power – 1670 KW (2240 HP)
- Rated velocity – 22 MPH at 30′
- Cut-in velocity – 11 MPH at 30′
- Ground clearance – 50′
- Diameter > 200′
- Tip speed < 400′/sec
- Wind shear & tower shadow



Predicted Performance Mod-1 Wind Turbine



Mod-1 Wind Turbine Start-up Blade Angle Schedule

Mode Cases

Case 1 – A wind velocity of 22 mph occurs 30 feet above ground level The rotor produces 1670kw of power (no losses) at operating rotor speed (endurance limit).

Case 2 – With the rotor blades set to operate for Case 1, the wind velocity increases to 60 mph in 0.25 seconds. No change in blade pitch angle occurs. The rotor speed increases to 25 percent overspeed (10^5 cycles).

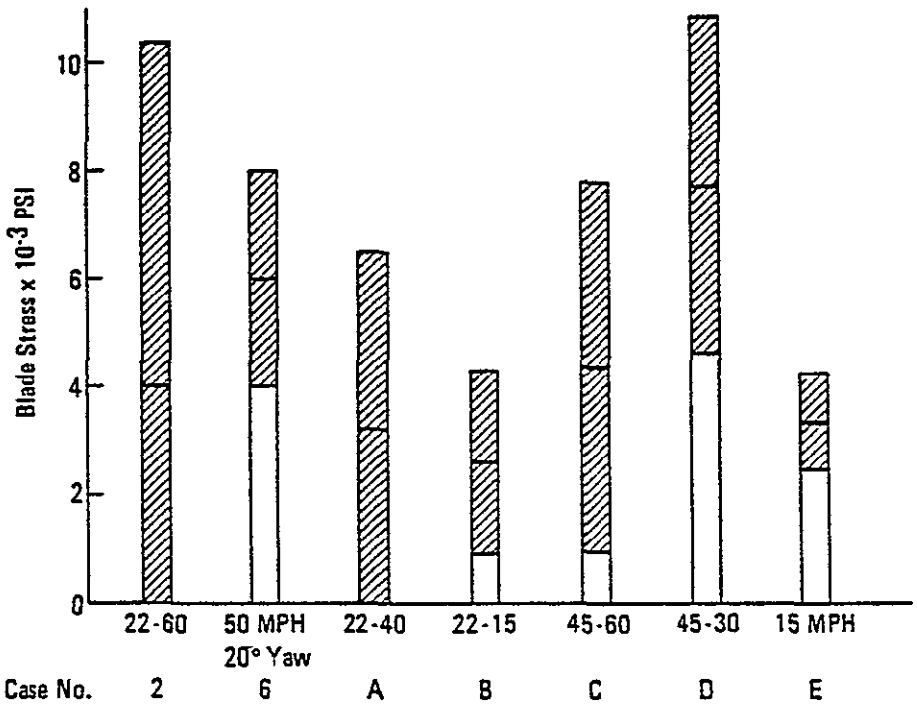
Case 3 – With the rotor blade pitch angle set to operate for Case 1, the wind velocity is at 22 mph and the rotor speed at operating rpm the blade pitch angle is changed to the feathered position in 11 seconds (proportional limit).

Case 4 – With the rotor blade pitch angle set to produce 1670 kw of power (no losses), the wind velocity decreases from 22 mph to 0 mph in 0 25 seconds (10^5 cycles)

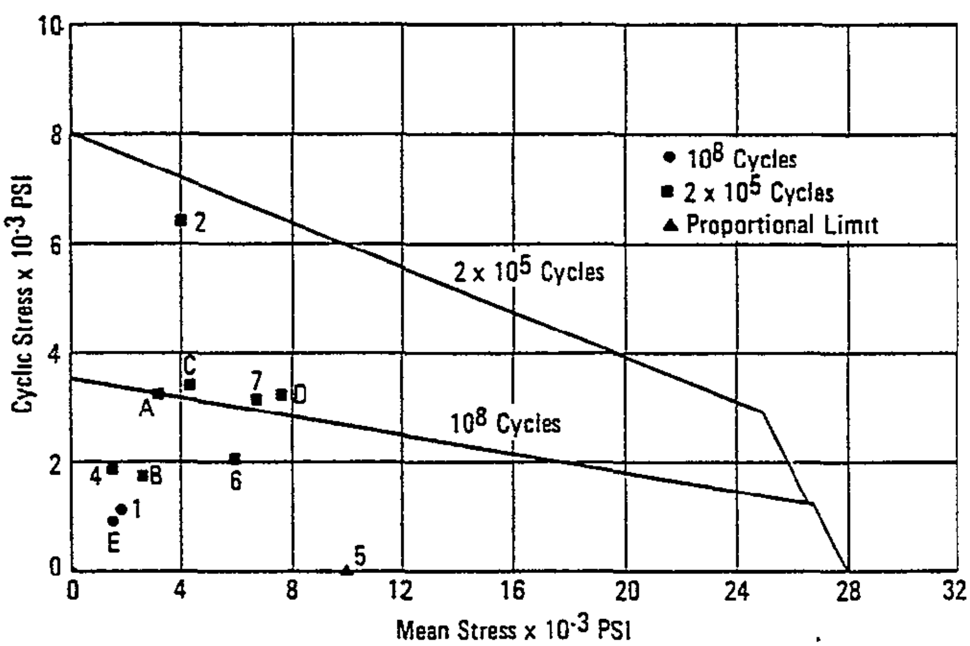
Case 5 – With the blades set and locked in a horizontal feathered position, a maximum wind velocity of 120 mph occurs at 30 ft. above ground level in any direction while the blade yaw angle remains fixed (proportional limit)

Case 6 – With the rotor yawed to the wind 20° and operating at design rpm rotor speed at a wind velocity of 50 mph, the nacelle is yawed at its maximum rate of .25°/sec in the direction producing the maximum shaft bending moments (10^5 cycles)

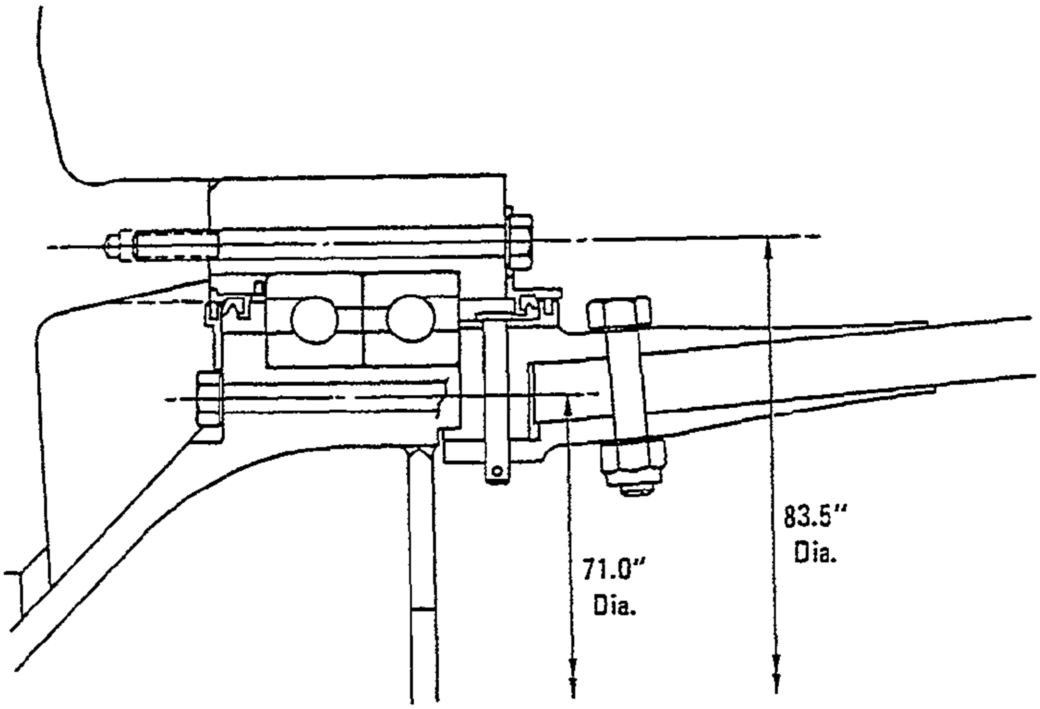
Case 7 – With the rotor operating at design rpm rotor speed and no power on the generator, a tower shadow of 50 percent (velocity retardation) occurs behind the tower (10^5 cycles)



Supplementary Load Cases



Blade Goodman Diagram



Blade Retention Bearing

Bearing Design Requirements

30 yrs. life for following conditions

Condition I	22 MPH steady state 5×10^9 cycles
Condition II	22 MPH gusting to 60 MPH 1×10^6 cycles
Condition III	50 MPH steady state 20° inflow angle 5.5×10^8 cycles

Blade Status

- Spar mandrel templates complete
- Shell mandrel templates started
- Retention ring material ordered
- System dynamic analysis started
- Estimated blade weight 15,100 lbs.

Blade Retention Bearing Types

- Three roller
- Duplex tapered roller
- Duplex angular contact ball

Bearing Status

Bearing type selected

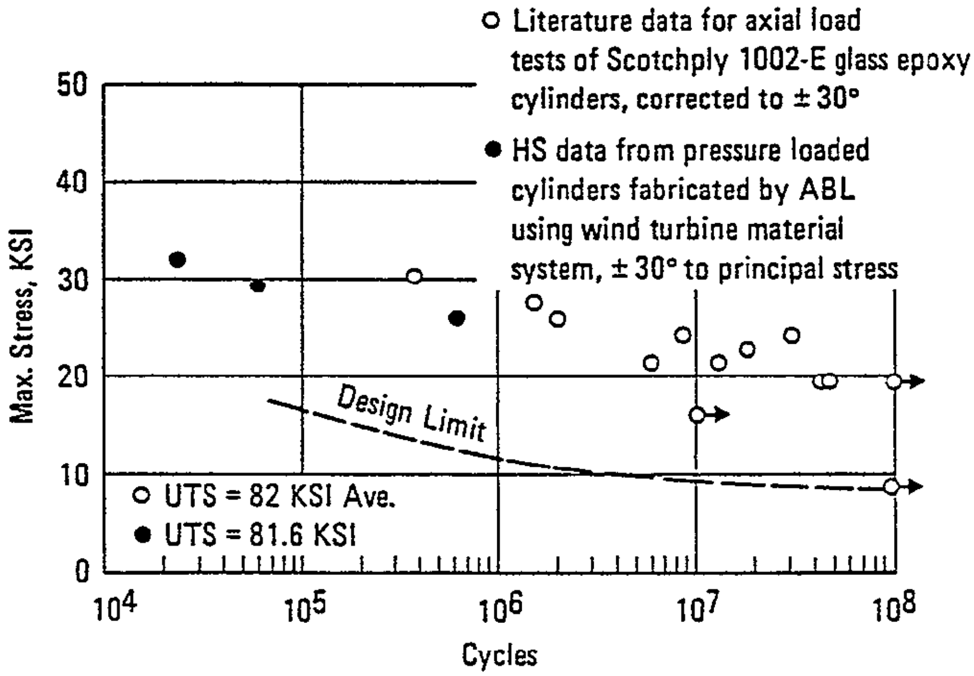
Loading defined

Bearing specification sent to vendors

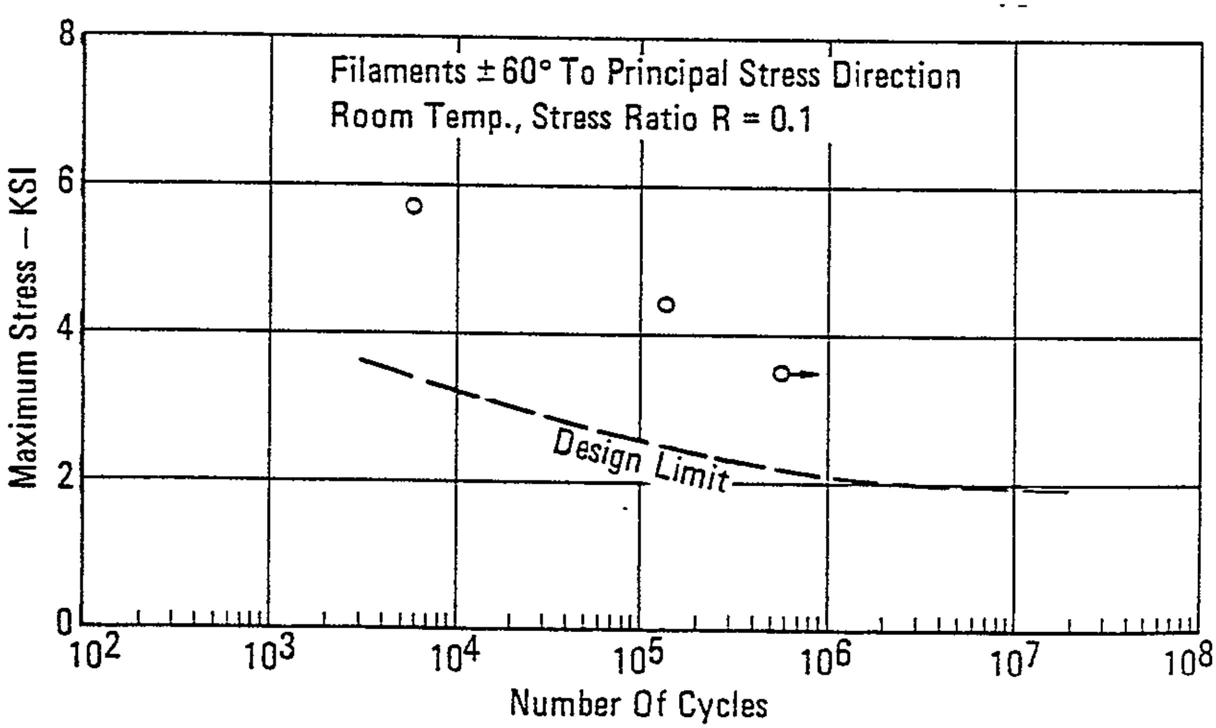
Bearing vendors have responded

Mod-1 Wind Turbine Blade Materials

Glass Roving	PPG Type 1062NT-15
Finish	Chrome-Silane
Resin	Epon 826
Hardener	Jeffamine D230
Mix Viscosity CPS @21^{0}C	450
Part Cure	RT Followed By 16 Hrs @ 80^{0}C



Test Results from Cylinders
(Room Temp., Stress Ratio R = 0.1)



Fatigue Tests Results from Cylinders Wound by ABL Using Wind Turbine Blade Materials System

General Inspection Test Coverage

- Visual

  During winding
  Final overall

- Tap test

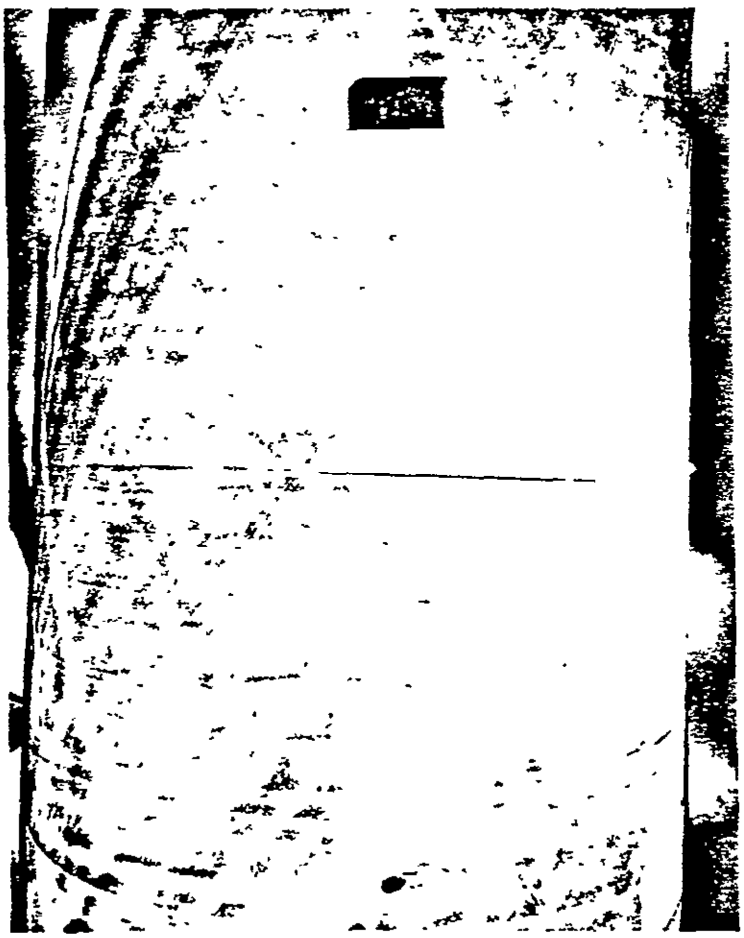
  Spar
  Shell

- Bright light

  Entire completed blade

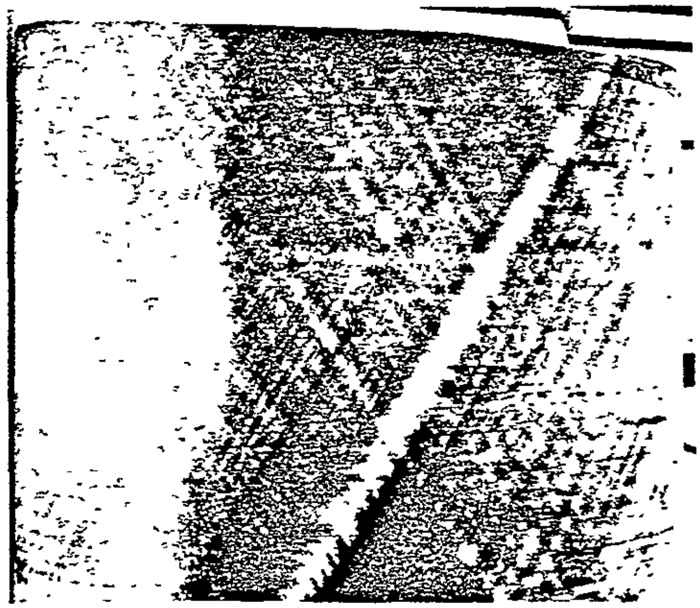
- X-ray/fluoroscope

  Recommended for recording of metal ring area



Satisfactory Laminate Bright Light Test

Unsatisfactory Laminate Bright Light Test



Mod-1 First Article Inspection

WIDTH

BLADE THICKNESS

CONTOUR

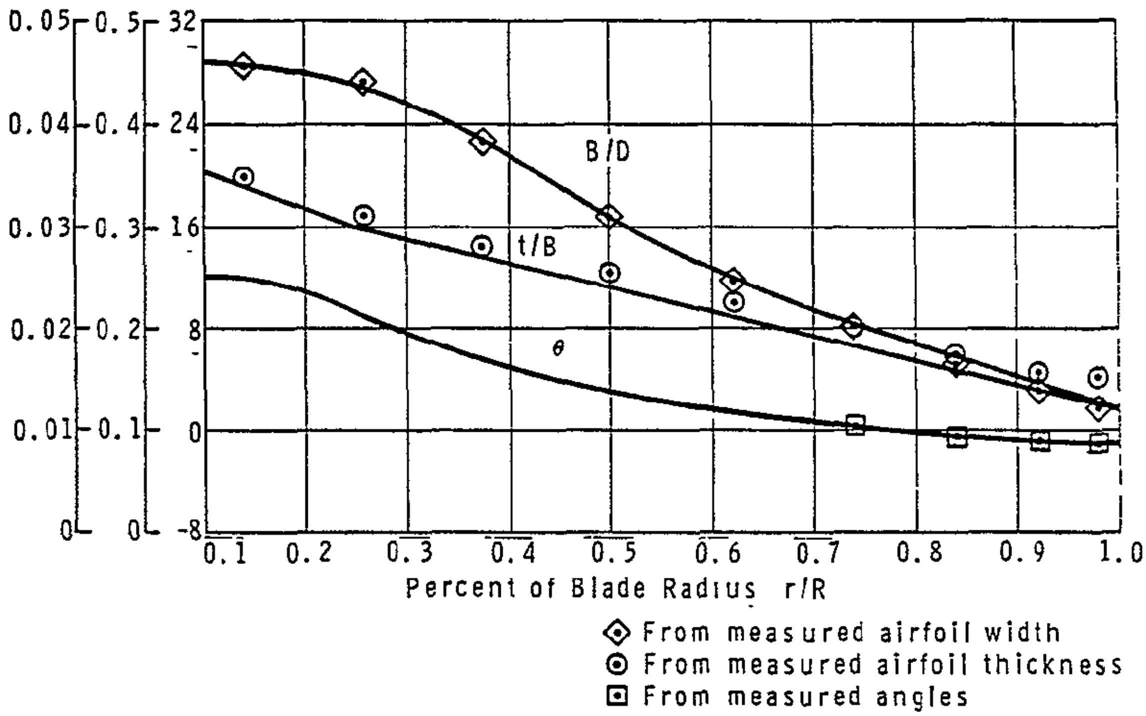
TWIST ANGLE

SHELL THICKNESS

LEADING EDGE ALIGNMENT

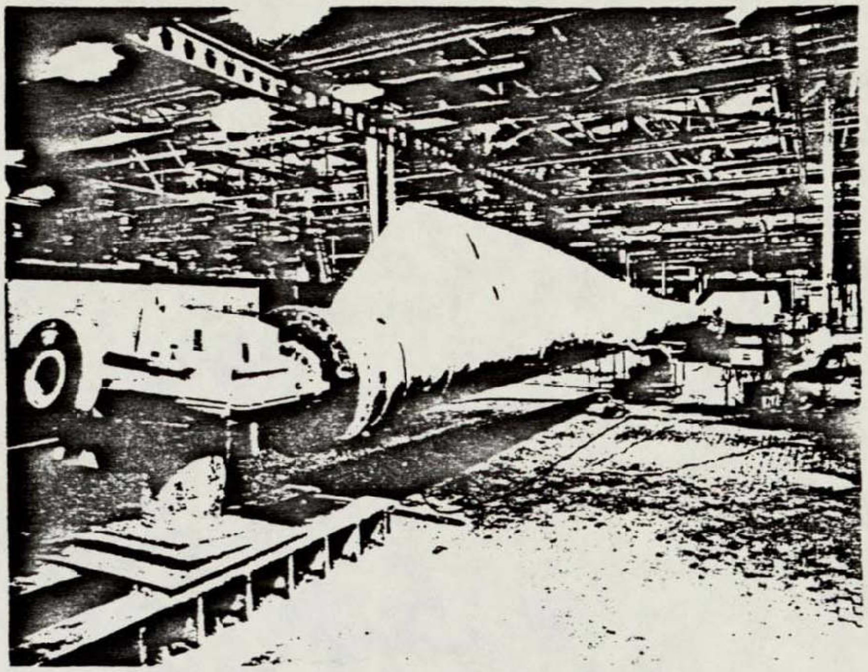
FACE ALIGNMENT

Manufacturing Prototype Blade Measured Aerodynamic Shape

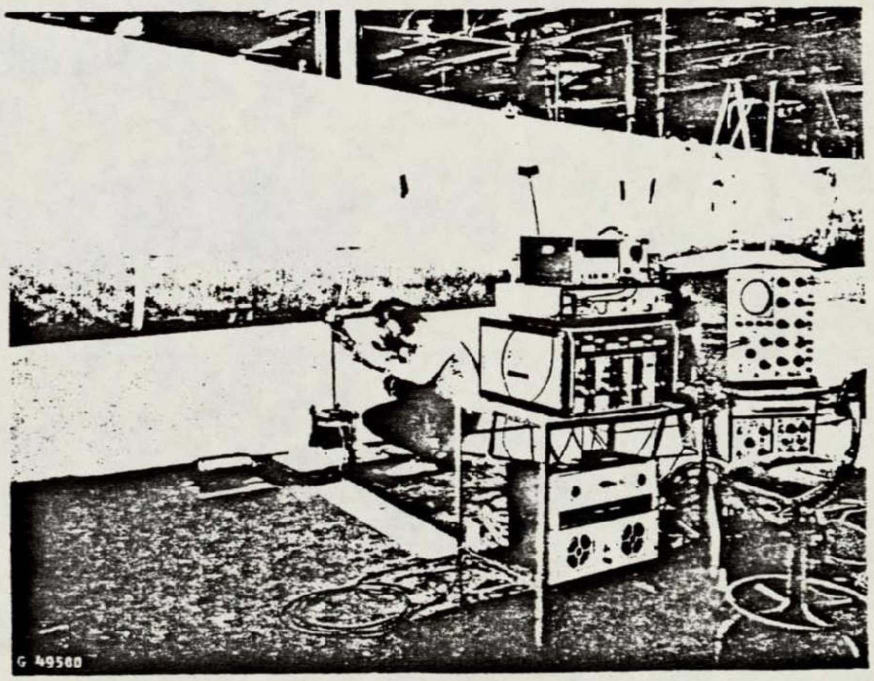


Experimental Modal Analysis

- Determines blade resonant frequencies and mode shapes
- Performed with blade cantilever mounted T.E. up
- Blade randomly excited with a shaker
- Accelerometer responses recorded at intervals along blade
- Responses computer plotted for frequency peaks
- Displacement mode shapes computer synthesized and plotted
- Test successfully performed on mfg. prototype blade



Test Arrangement



EMA Shaker Installation

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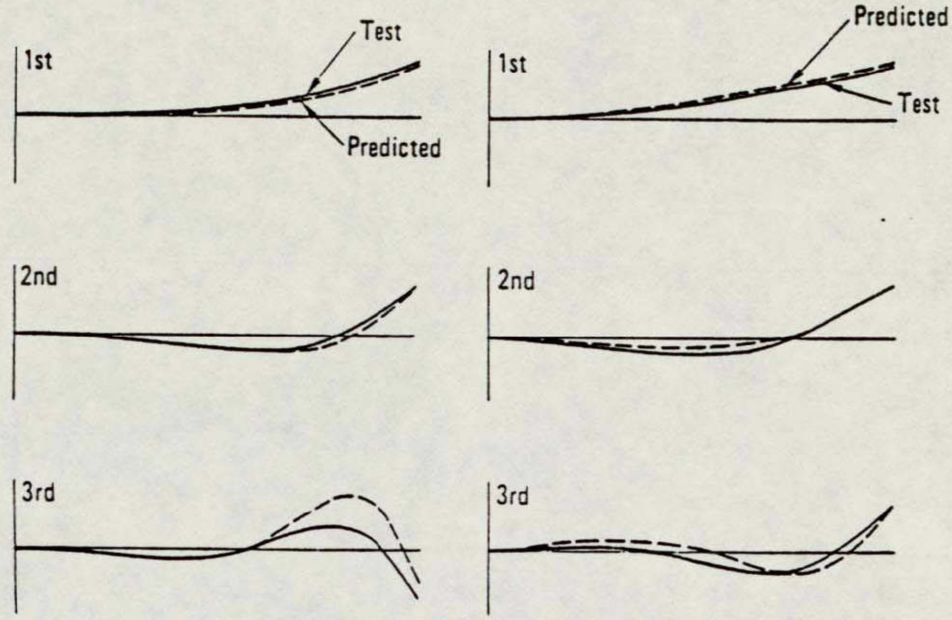
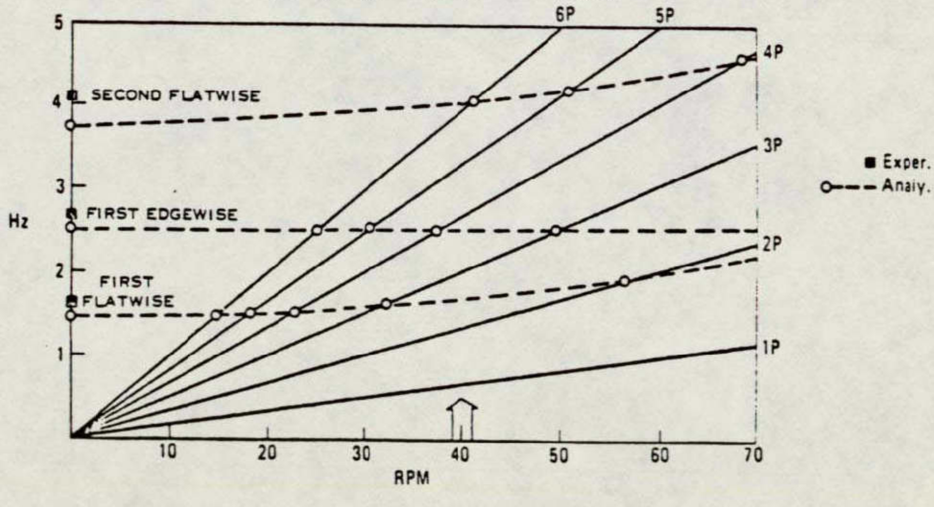
Manufacturing Prototype - Mode Shapes

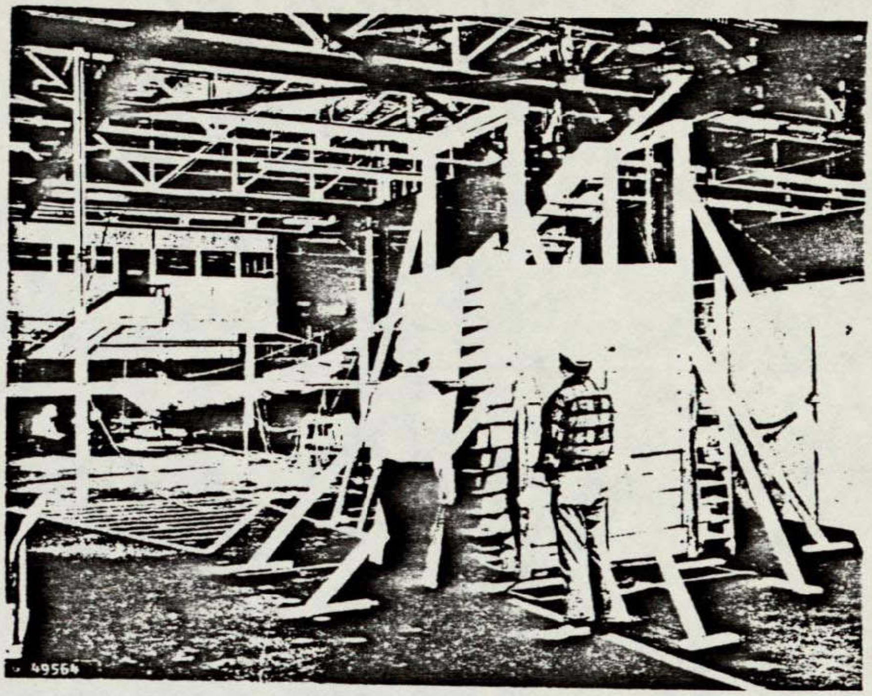


Comparison with Analytical Model
Manufacturing Prototype in Test Mount

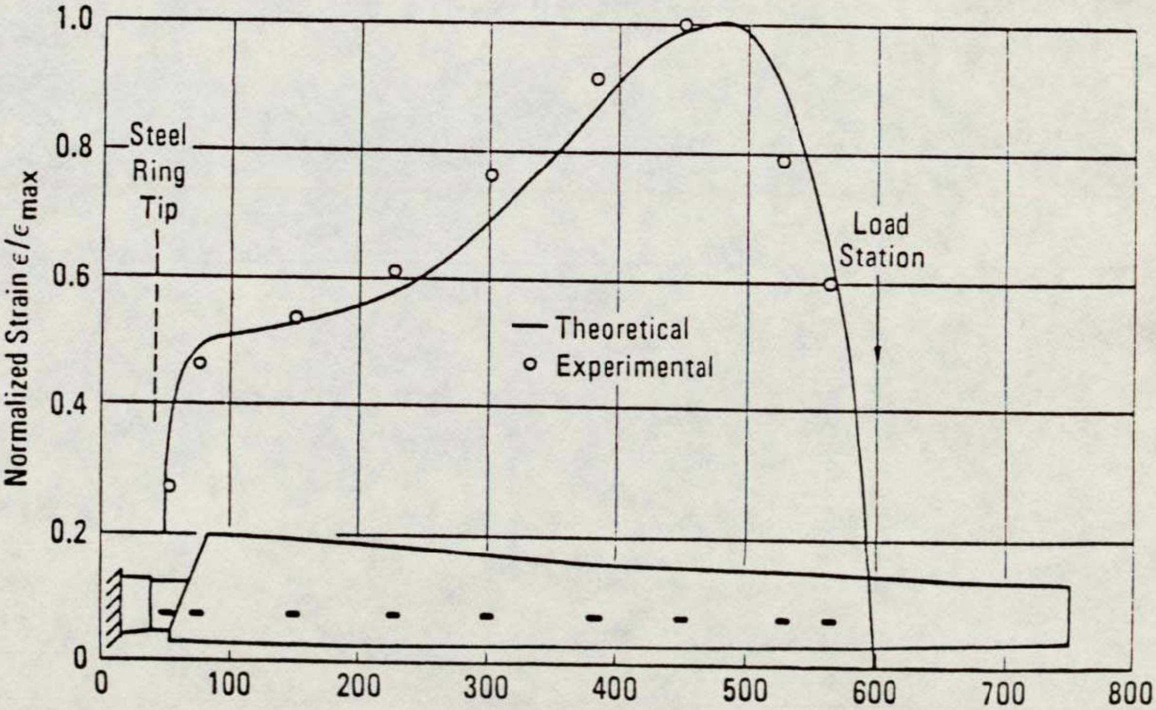
Mode	Frequency – Hz Analytical	Experimental
1st Flatwise	1.45	1.53
1st Edgewise	2.51	2.63
2nd Flatwise	3.73	4.05
3rd Flatwise	8.08	8.59
2nd Edgewise	9.86	9.36
3rd Edgewise	24.75	22.38
Static Torsional	28.59	29.26



Blade Under Static Load



Mod-0 Manufacturing Prototype Longitudinal Strain Distribution
Load at 600 in Station



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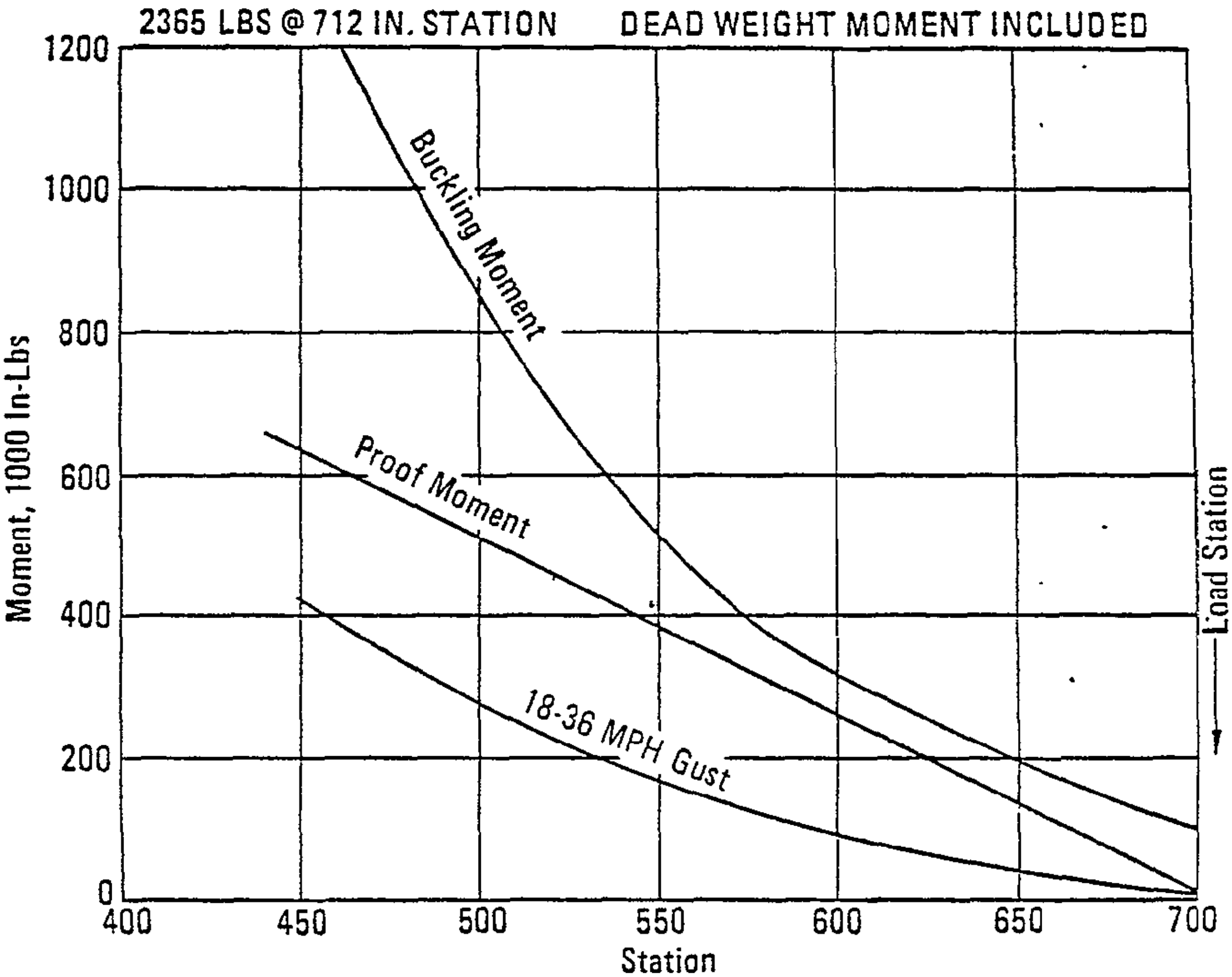
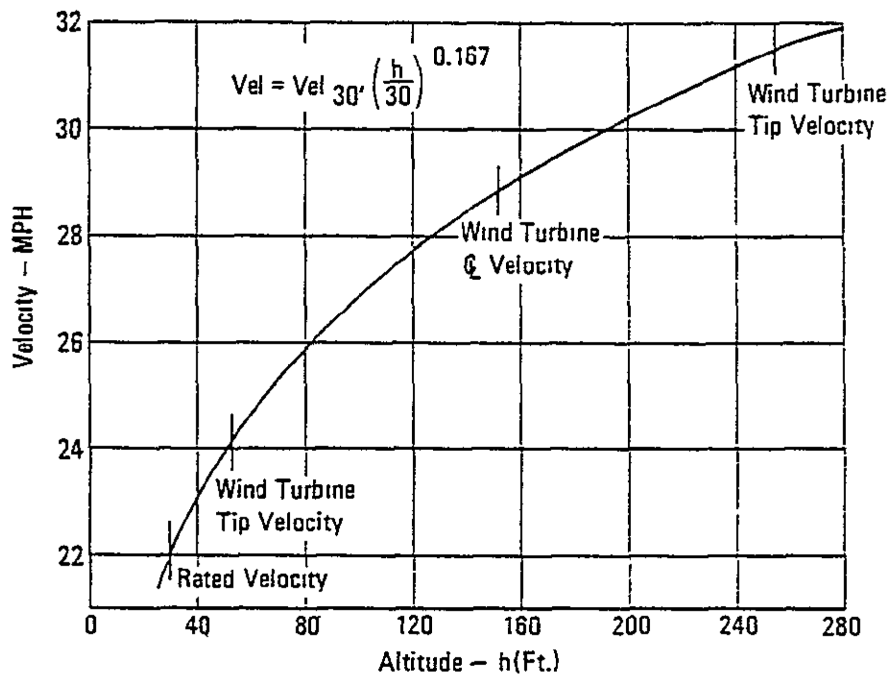
Mod-0 Manufacturing Prototype Blade Proof Moment Plot



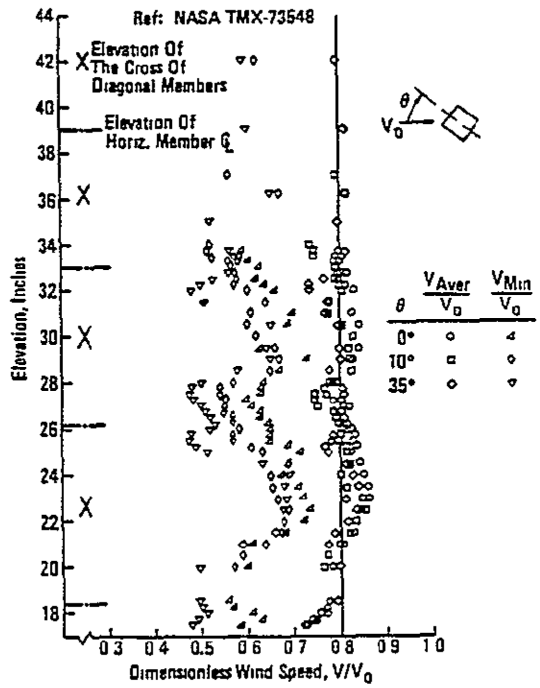
C-3

Specifications Mod-1 Wind Turbine

- Output power – 1670 KW (2240 HP)
- Rated velocity – 22 MPH at 30'
- Cut-in velocity – 11 MPH at 30'
- Ground clearance – 50'
- Diameter > 200'
- Tip speed < 400'/sec
- Wind shear & tower shadow



Wind Shear Velocity Gradient



_Vertical Distribution of the Average and Minimum Wind Speeds in the Wake of the Bare Mod-0 Tower Model

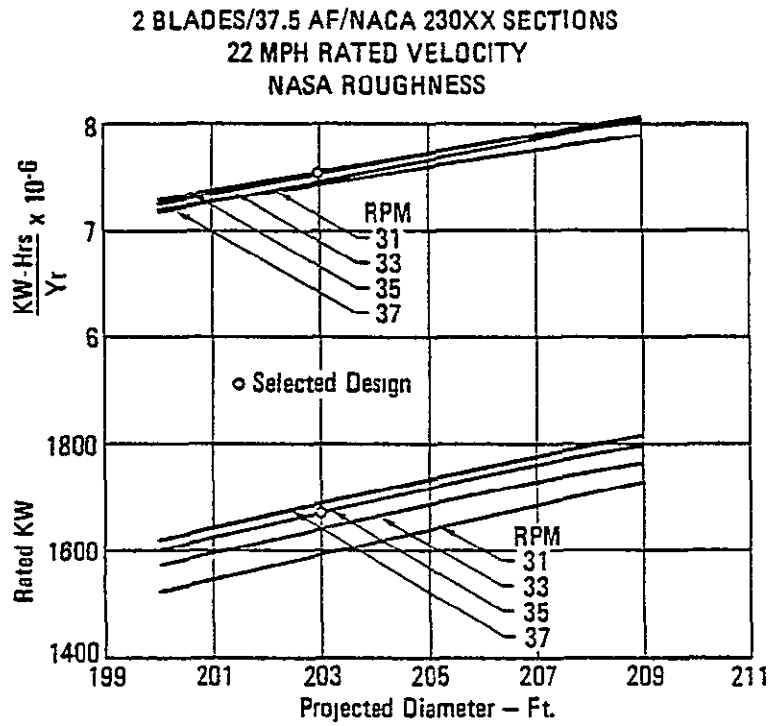
Aerodynamic Design Objective:

To meet performance specifications with maximum yearly power output consistant with structural constraints.

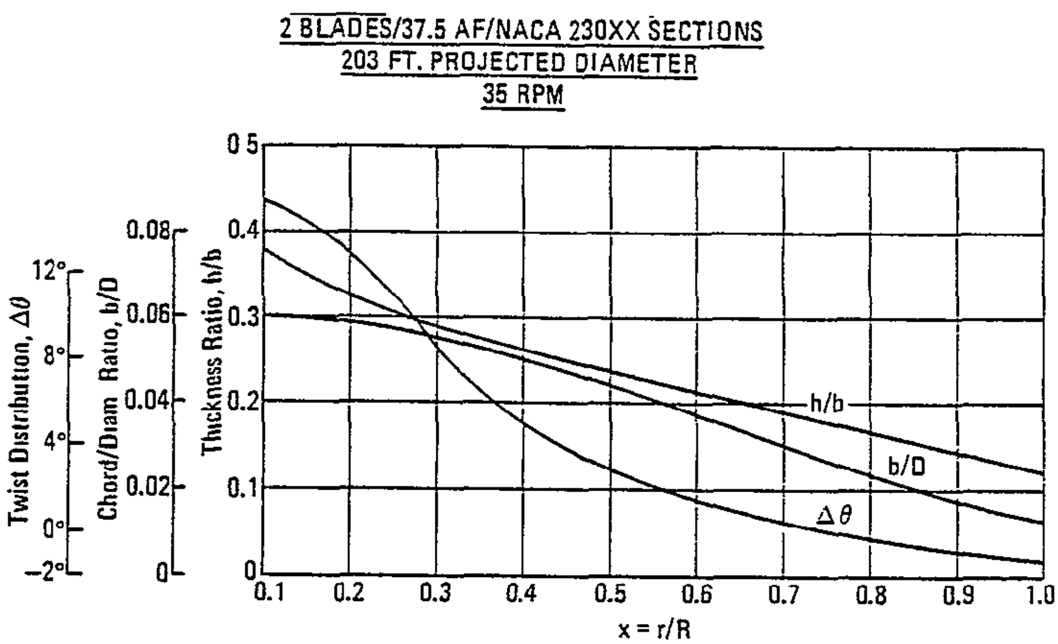
Parameters Investigated

- Twist distribution
- Planform
- Activity factor
- RPM
- Diameter

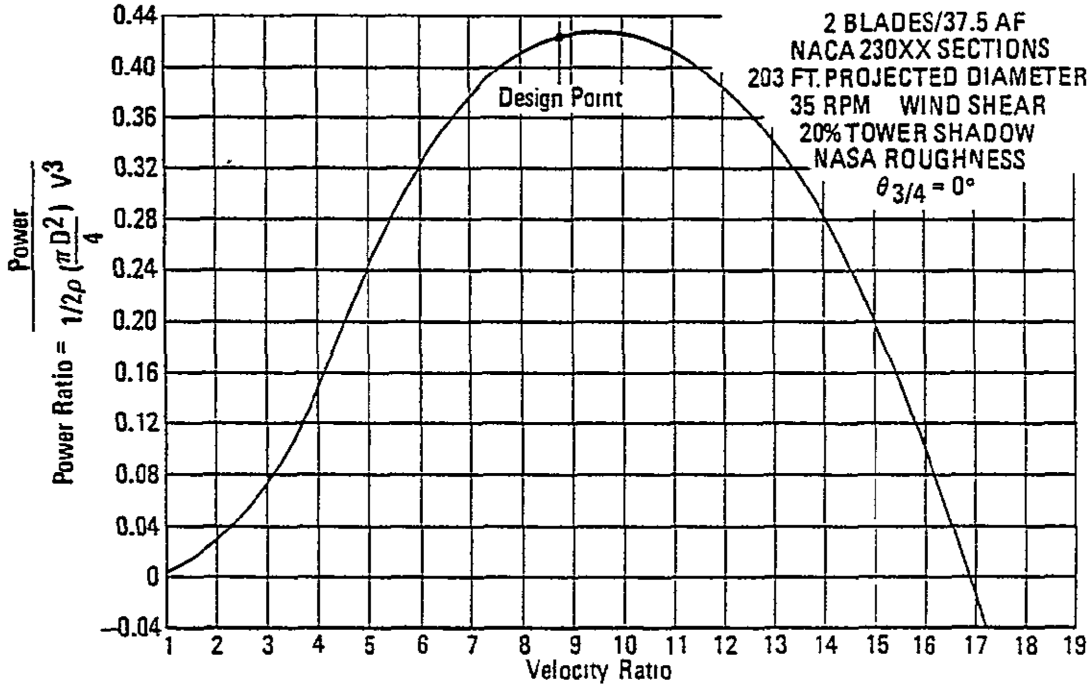
Optimization Study Parameters



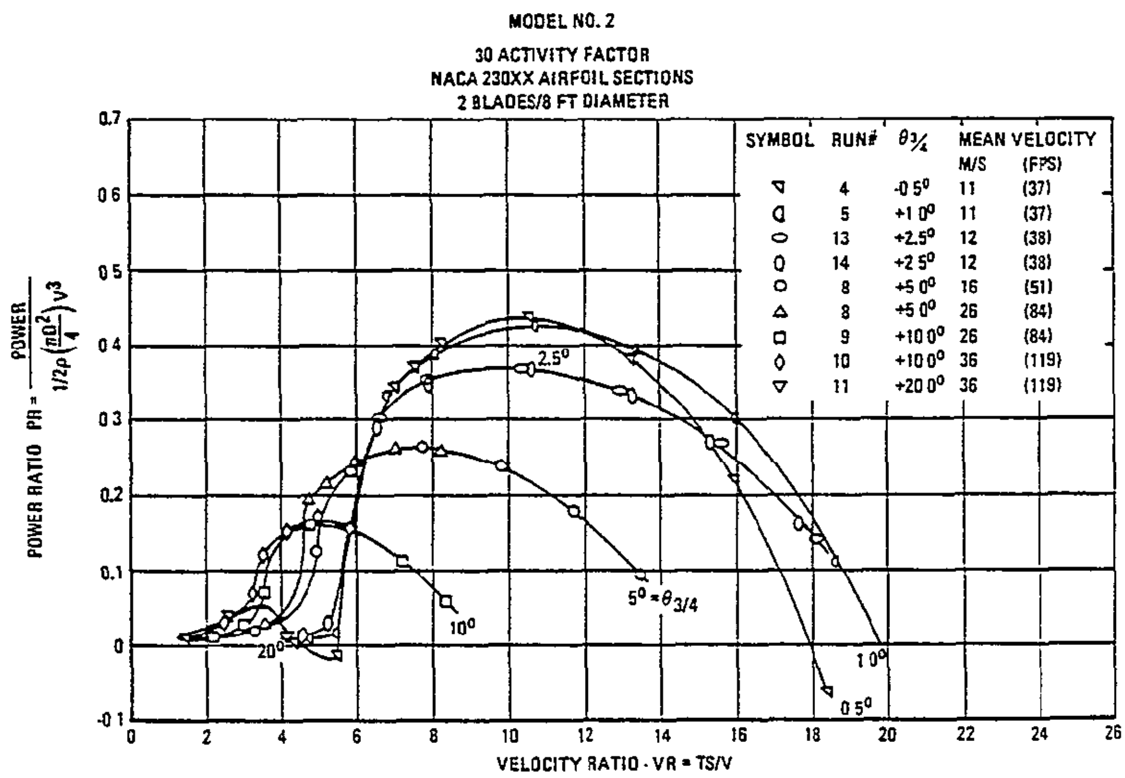
Parametric Study of Diameter and RPM



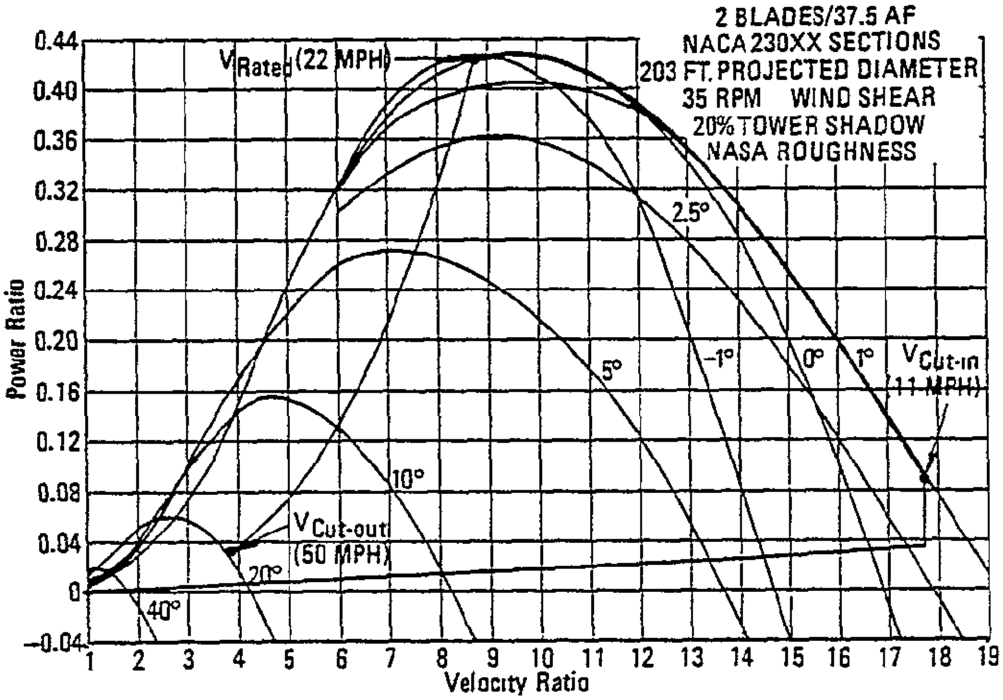
Blade Characteristics Mod-1 Wind Turbine



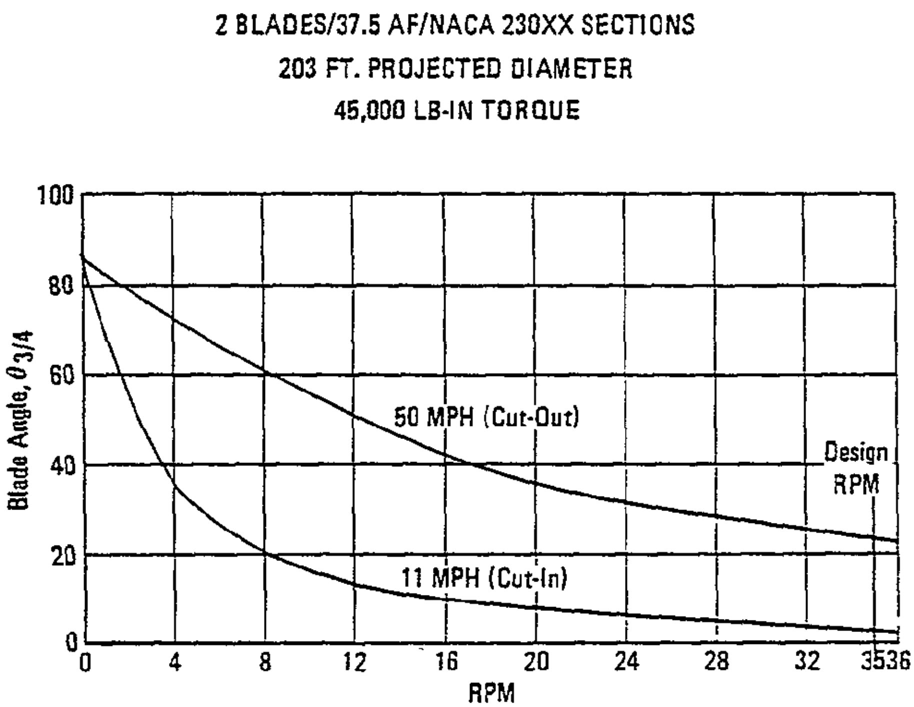
Predicted Performance Mod-1 Wind Turbine



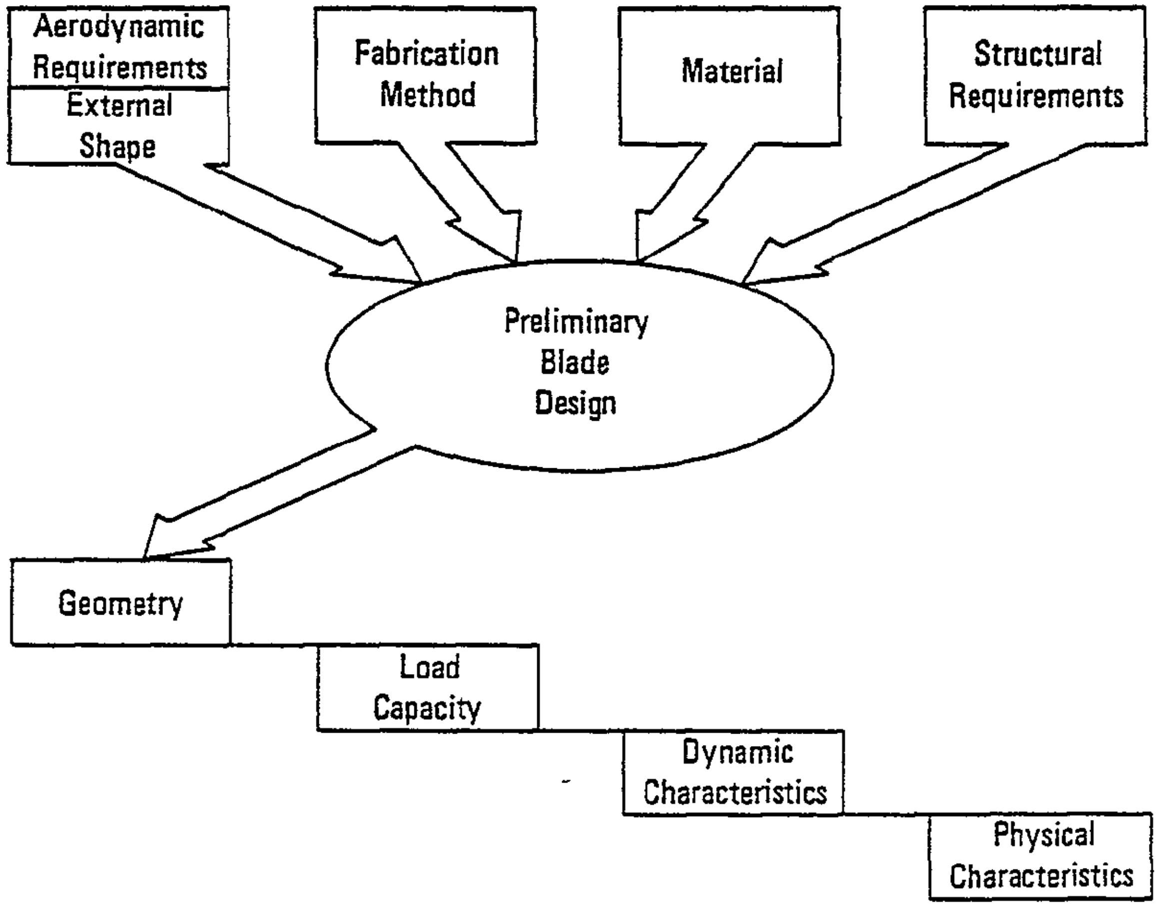
Effect of Blade Angle on Measured Power Ratio



Predicted Performance Mod-1 Wind Turbine



Mod-1 Wind Turbine Start-up Blade Angle Schedule



Blade Design Overview

General Requirements

Structural

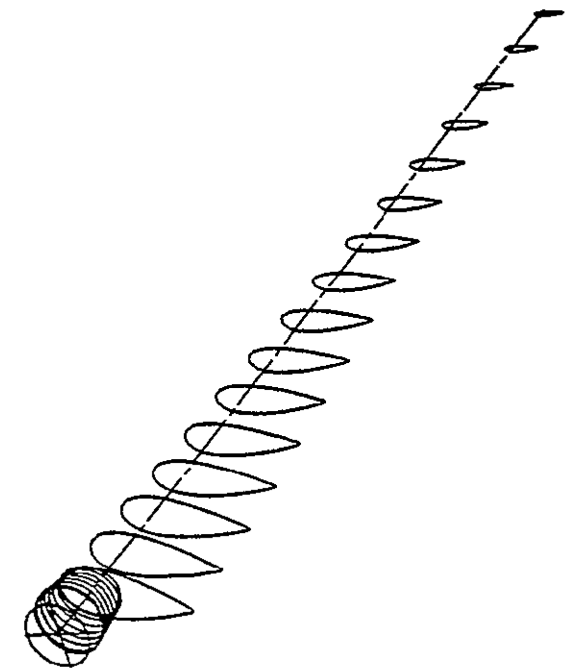
- Load cases from Exhibit B, Para. 2.1.2
- Supplementary load cases
- Stability cases from Exhibit A, Para. 2c
- Critical speed placement from Exhibit B, Para. 2.1.3
- Stall flutter from Exhibit A, Para. 2f
- Divergence from Exhibit A, Para. 2e

Environmental

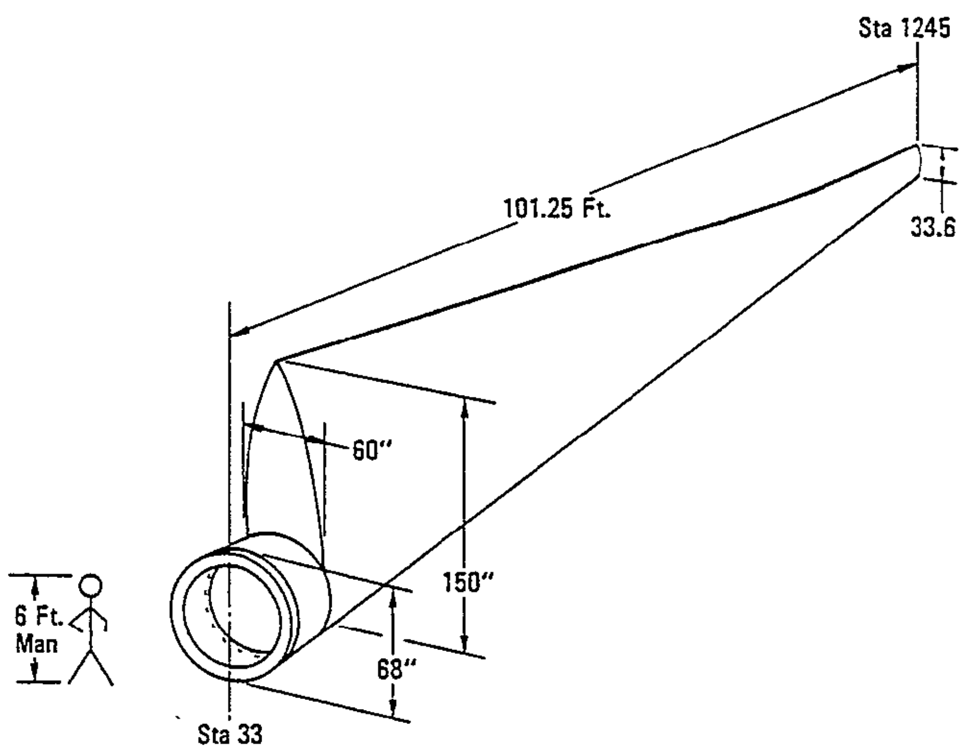
- Life
- Lightning

Geometric

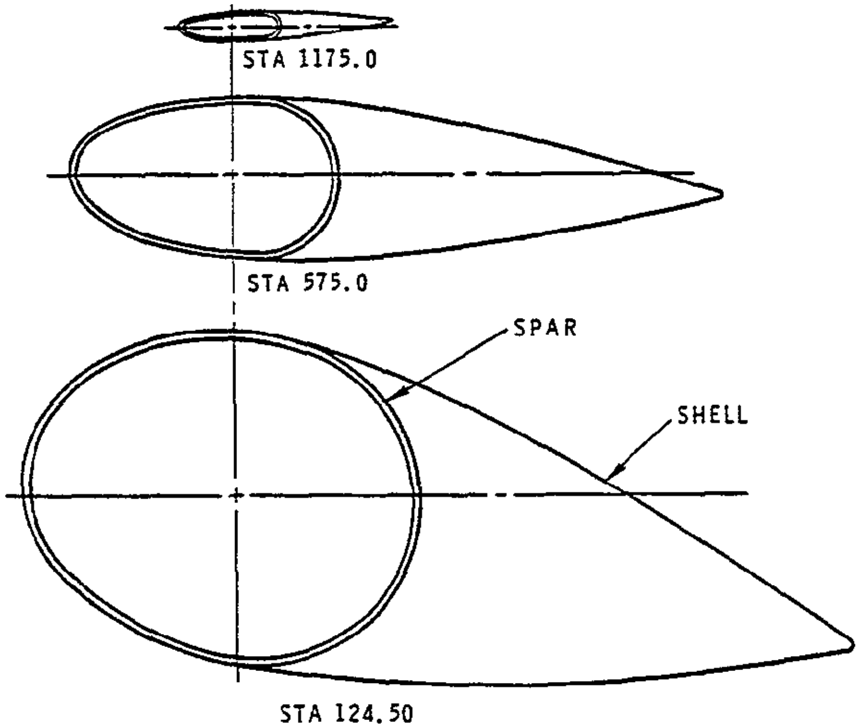
- Shape
- Weight
- Balance



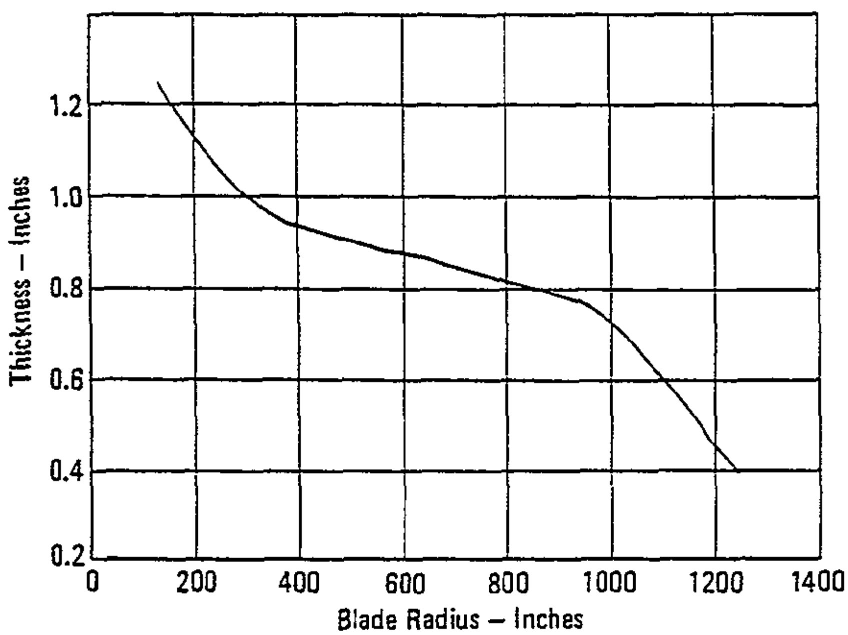
Blade Isometric View



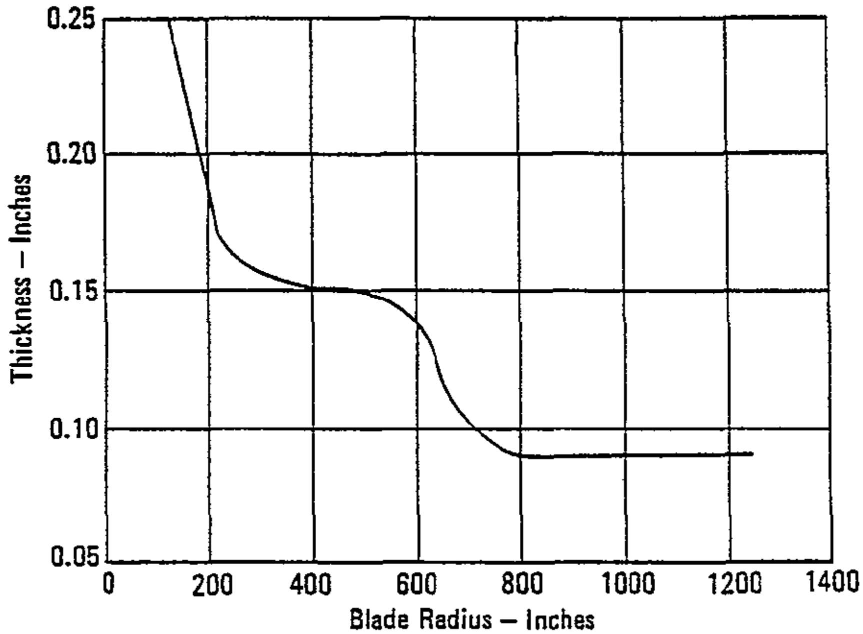
External Shape



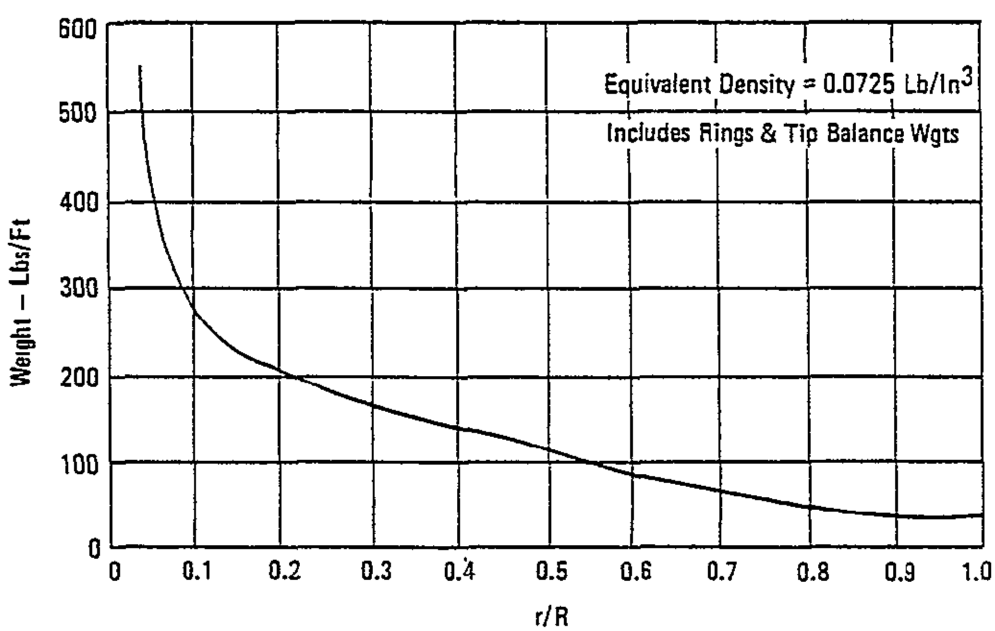
Blade Cross Sections



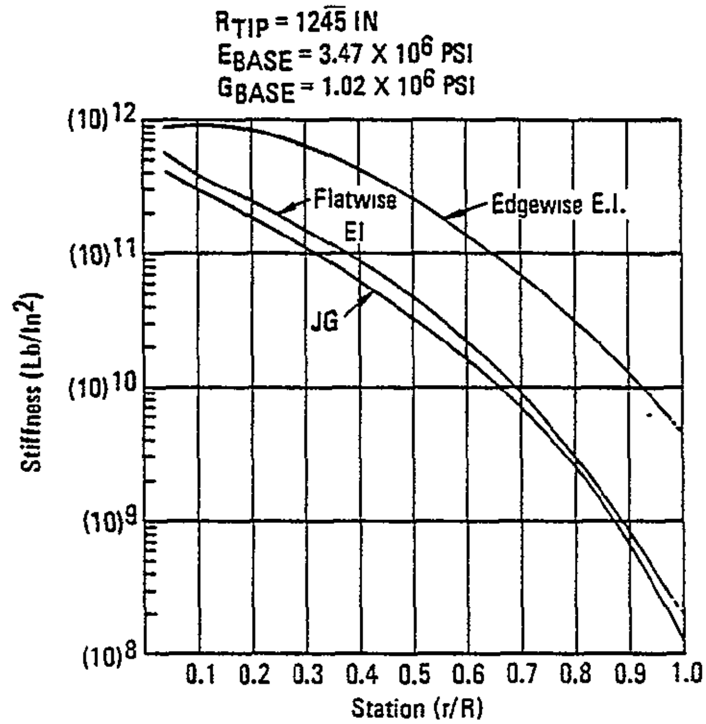
Blade Spar Wall



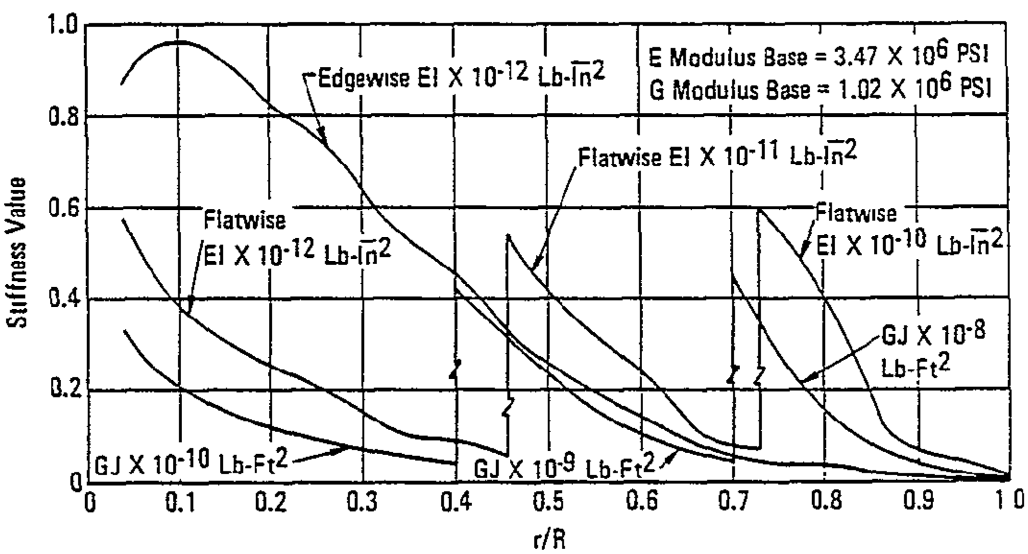
Blade Shell Wall



Weight Distribution



1500 kW Wind Turbine Generator Stiffness Distribution



Stiffness Distribution

Design Load Cases

Case 1 – A wind velocity of 22 mph occurs 30 feet above ground level. The rotor produces 1670kw of power (no losses) at operating rotor speed (endurance limit).

Case 2 – With the rotor blades set to operate for Case 1, the wind velocity increases to 60 mph in 0.25 seconds. No change in blade pitch angle occurs. The rotor speed increases to 25 percent overspeed (10^5 cycles).

Case 3 – With the rotor blade pitch angle set to operate for Case 1, the wind velocity is at 22 mph and the rotor speed at operating rpm the blade pitch angle is changed to the feathered position in 11 seconds (proportional limit).

Case 4 – With the rotor blade pitch angle set to produce 1670 kw of power (no losses), the wind velocity decreases from 22 mph to 0 mph in 0 25 seconds (10^5 cycles).

Case 5 – With the blades set and locked in a horizontal feathered position, a maximum wind velocity of 120 mph occurs at 30 ft. above ground level in any direction while the blade yaw angle remains fixed (proportional limit).

Case 6 – With the rotor yawed to the wind 20° and operating at design rpm rotor speed at a wind velocity of 50 mph, the nacelle is yawed at its maximum rate of .25°/sec in the direction producing the maximum shaft bending moments (10^5 cycles).

Case 7 – With the rotor operating at design rpm rotor speed and no power on the generator, a tower shadow of 50 percent (velocity retardation) occurs behind the tower (10^5 cycles).

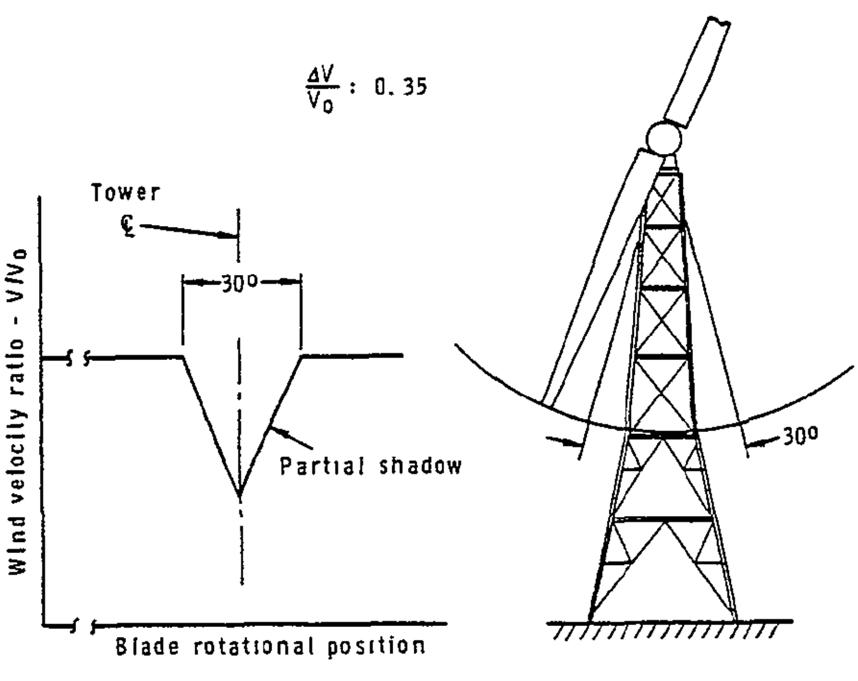
Supplementary Load Cases

Case A	22-40	Instantaneous gust, no blade angle change
Case B	22-15	
Case C	45-60	
Case D	45-30	

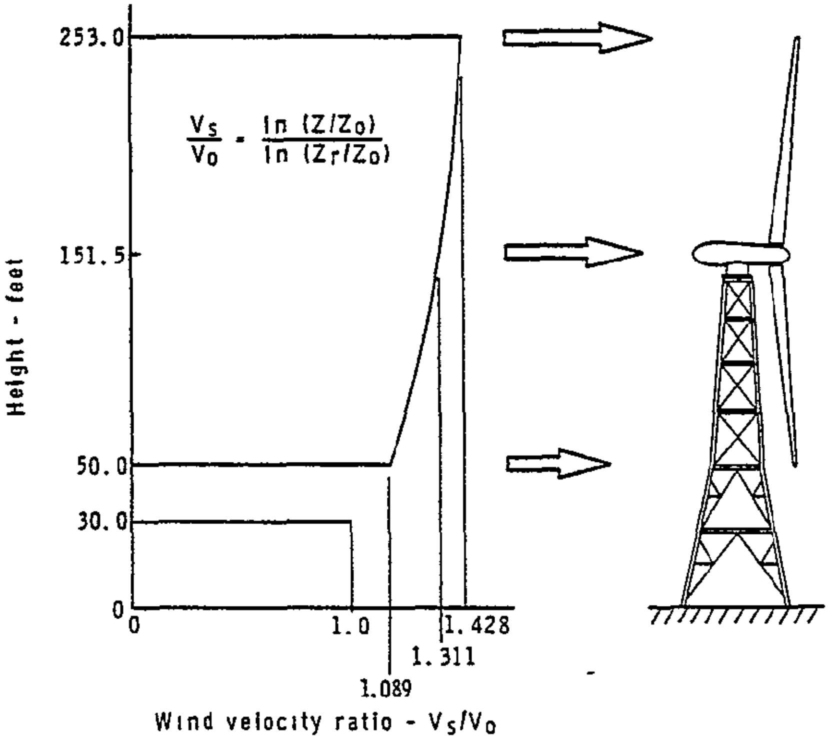
Case E 15 MPH steady state

Blade Design Criteria

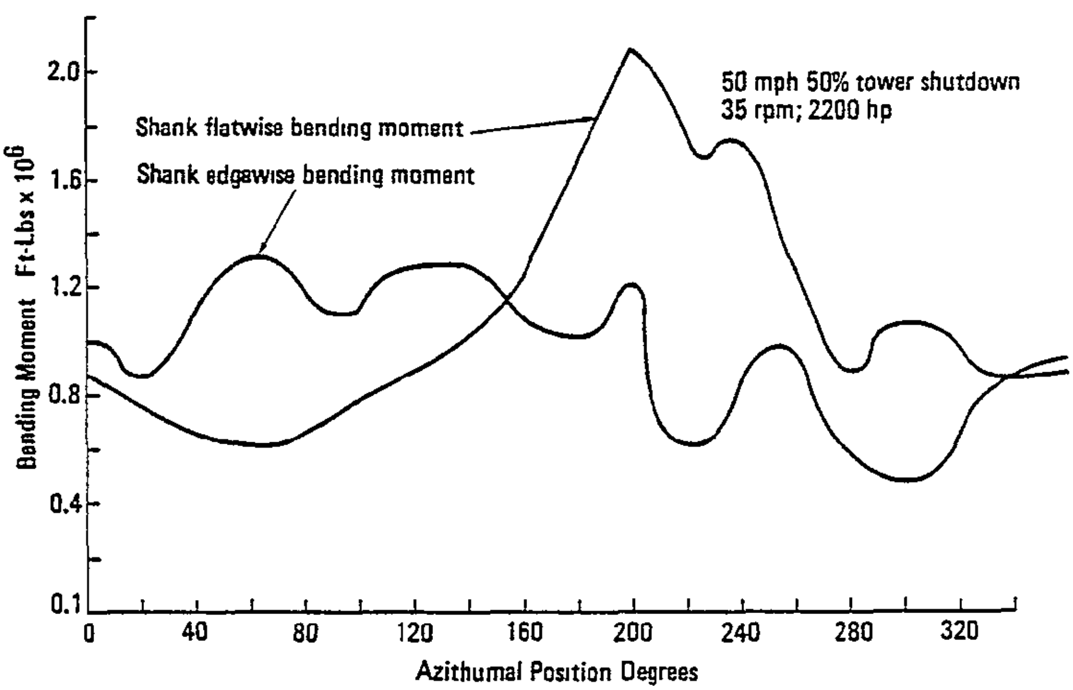
- Smooth airfoil loadings
- Wind shear profile
- Tower velocity retardation - 35%
- Gust modeled as a step change



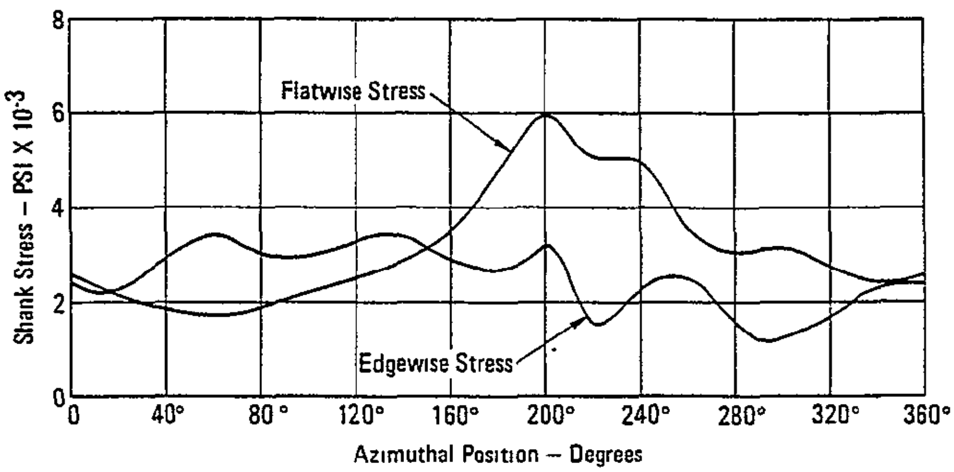
Tower Shadow



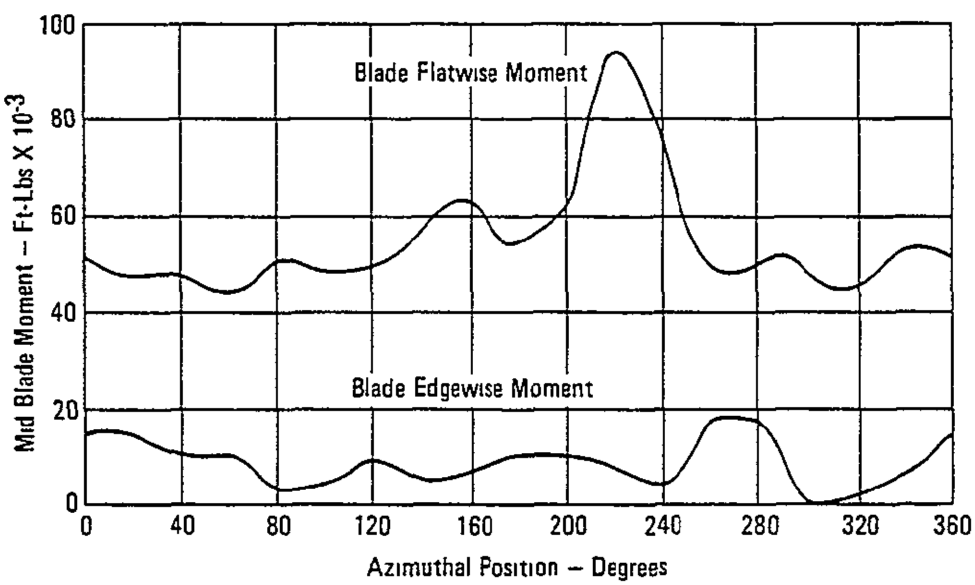
Wind Shear



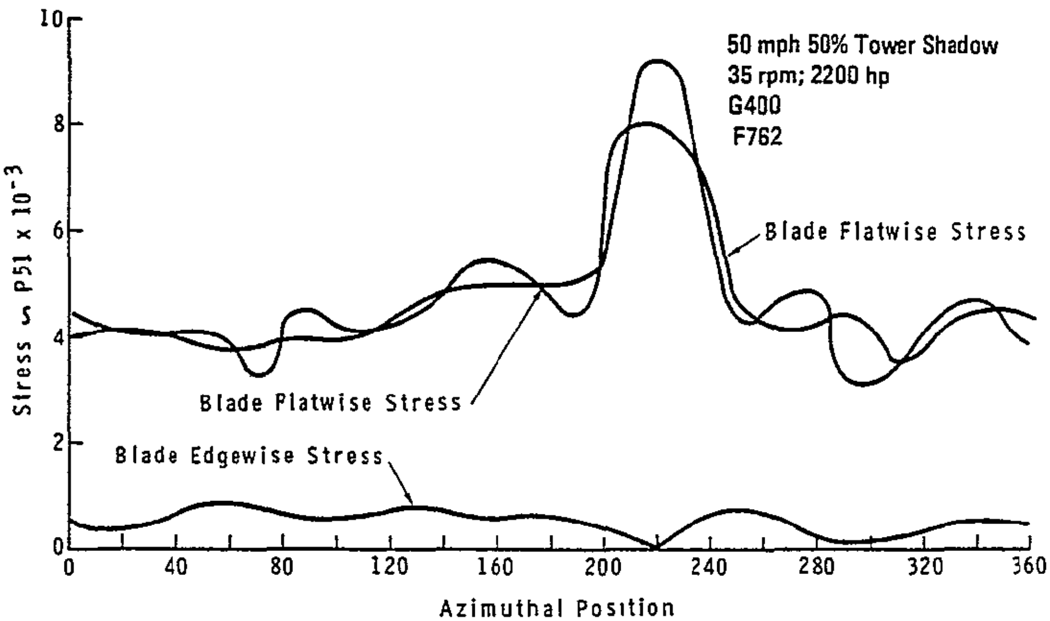
Shank Moment



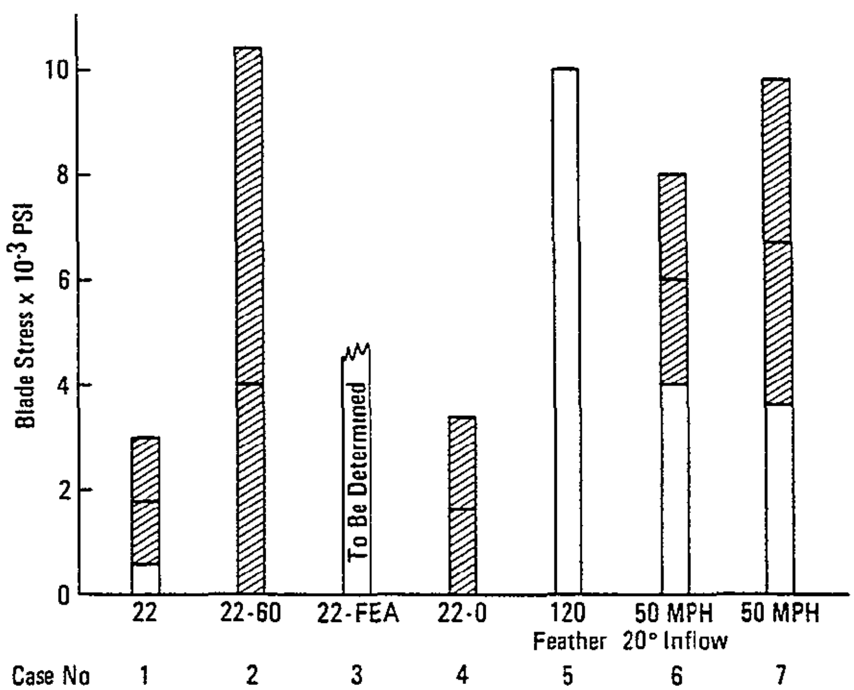
Shank Stress



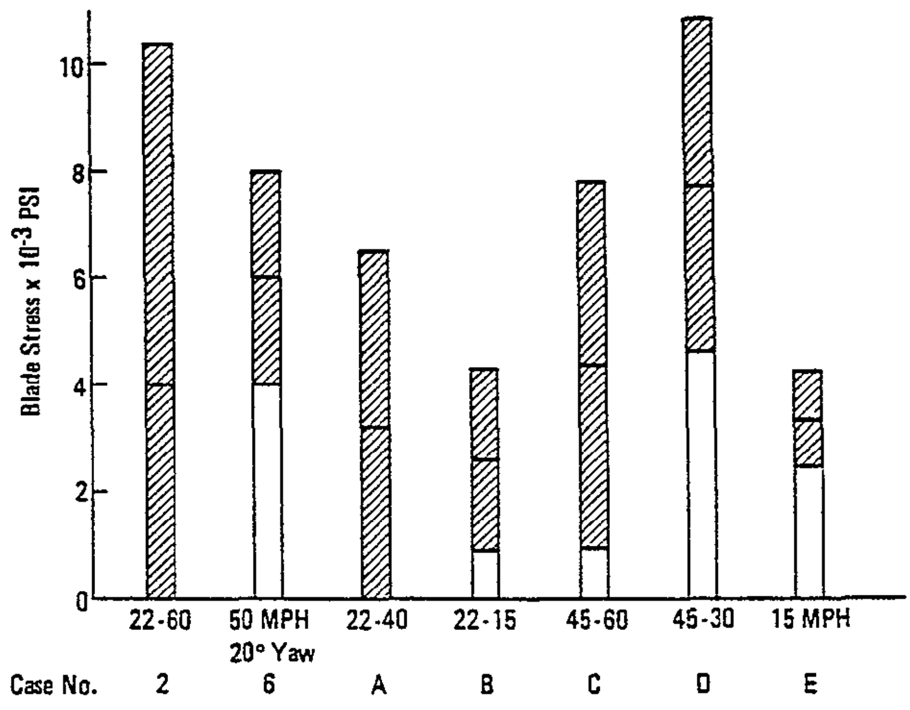
Mid-Blade Moment



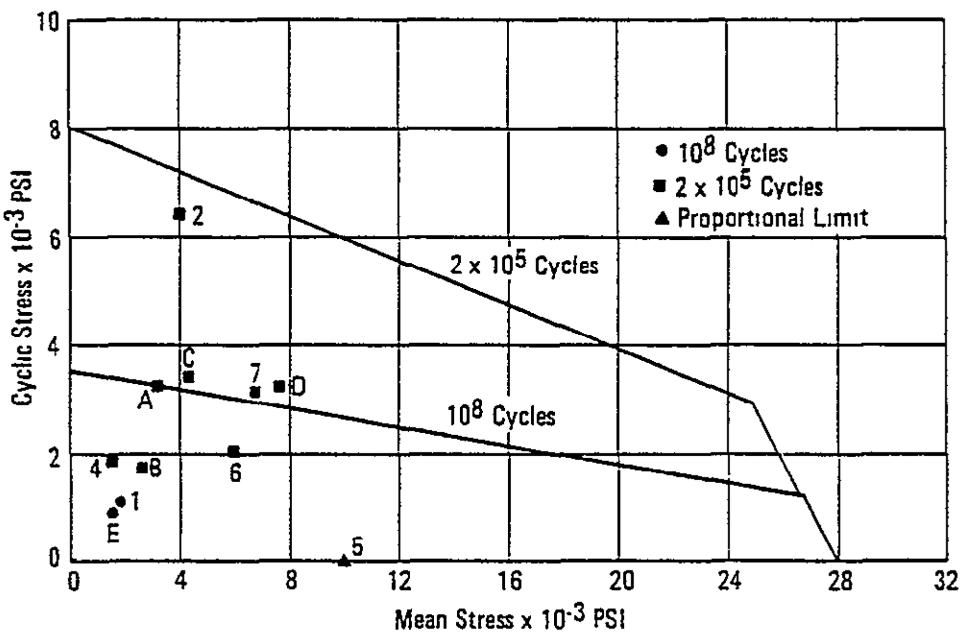
Mid-Blade Stress



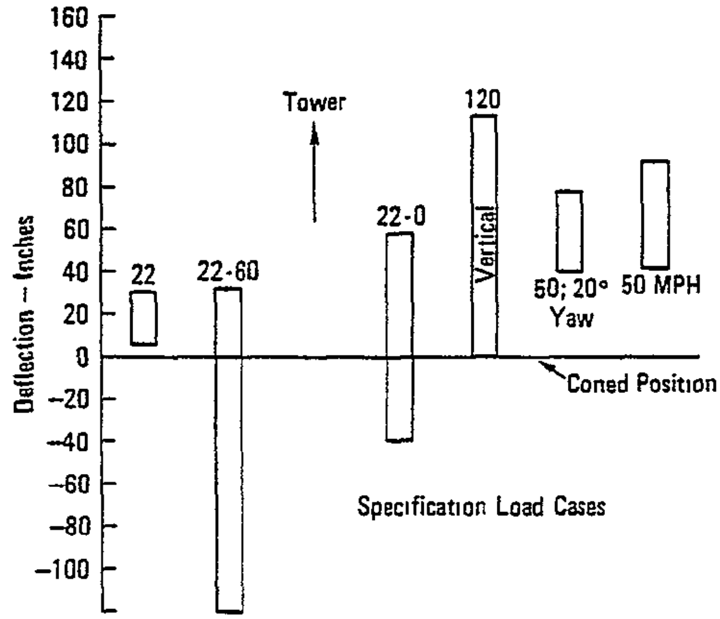
Specification Load Cases



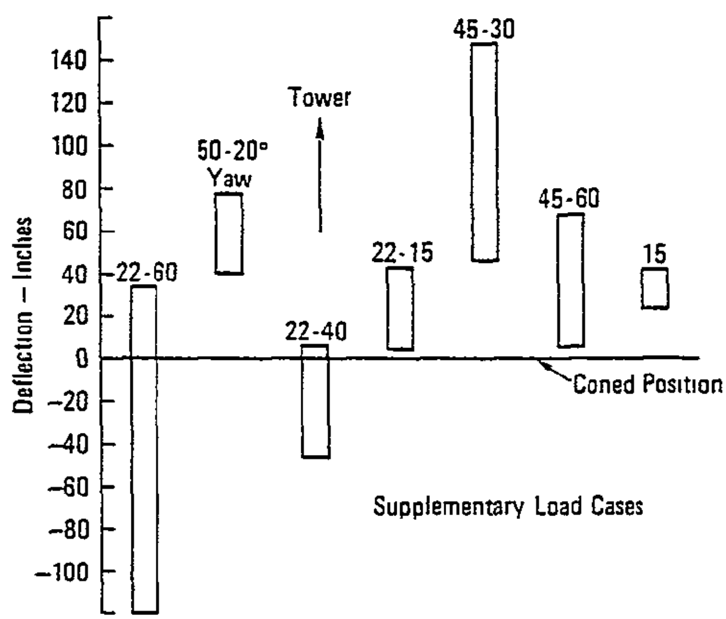
Supplementary Load Cases



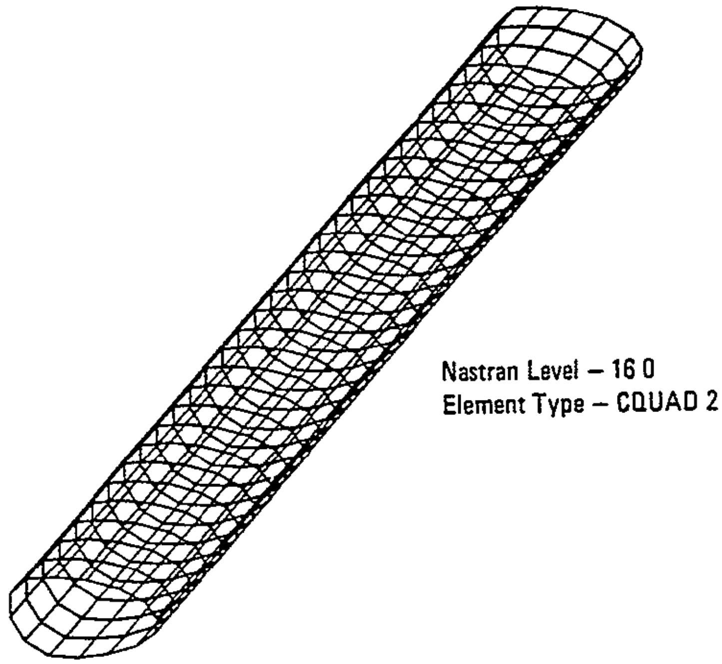
Blade Goodman Diagram



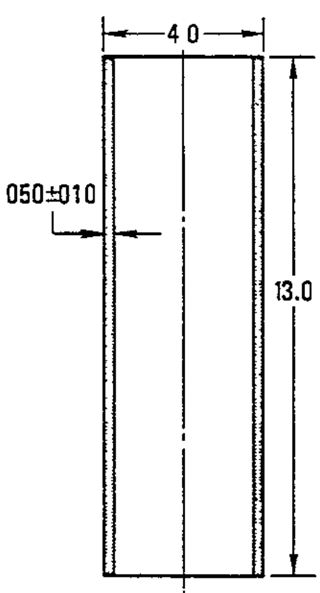
Blade Tip Deflection



Blade Tip Deflection



Nastran Buckling Model

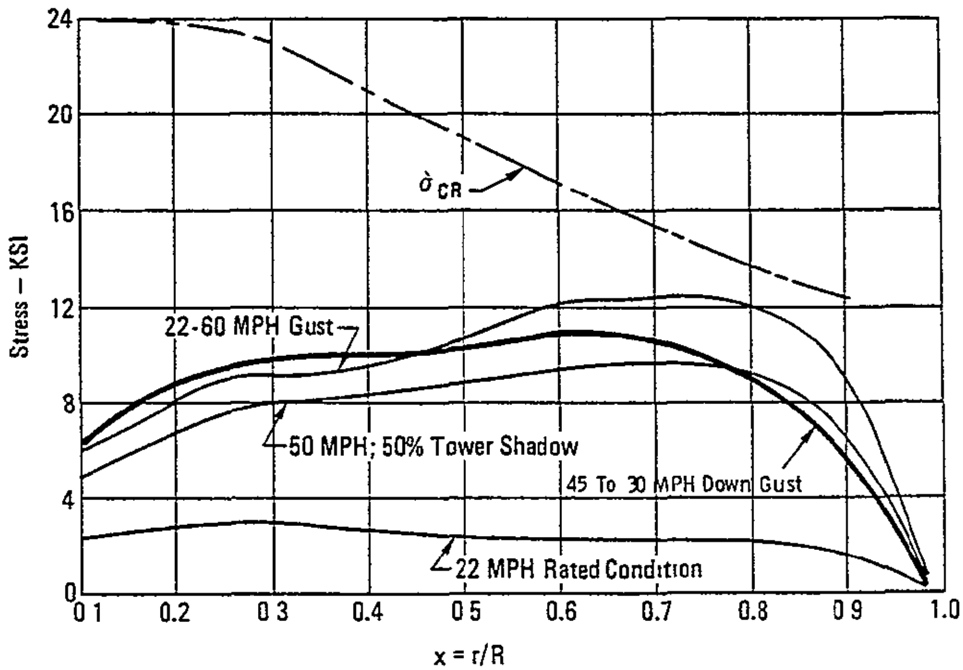


Serial No.	Layer					
	1	2	3	4	5	6
1 thru 10	40	-30	+30	-30	+30	90
11 thru 20	0	+60	-60	+60	-60	0

	Buckling	
	Stress	Load
Nastran	47754	30517
Nastran Corrected*	34956	22338
Test**	37314	24217

* Correction Factor, $\alpha = 0.732$, Structural Analysos Of Shells, pg 230
** AFFDL TR 73-7, Vol. I (Cylinder No. 10)

Cylinder Buckling Test Case



Buckling Capacity

Critical Speed Placement

	$1st_F$	$2nd_F$	$1st_E$	$1st_T$
Requirements	2.15-2.7 P	—	4.4-4.7 P	—
Current blade	2.6	6.35	4.57	19.2

Stability Conditions

Rotor Speed RPM	Wind Speed* MPH		
20	22	50	80
35	22	50	80
60	22	50	80

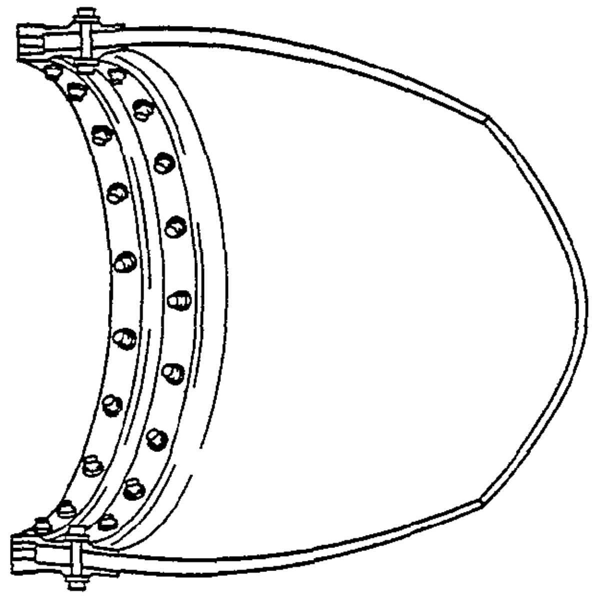
*All wind velocities @ 30 feet above ground.

72 X 10^6 In-Lb/Rad
180 X 10^6 In-Lb/Rad
216 X 10^6 In-Lb/Rad

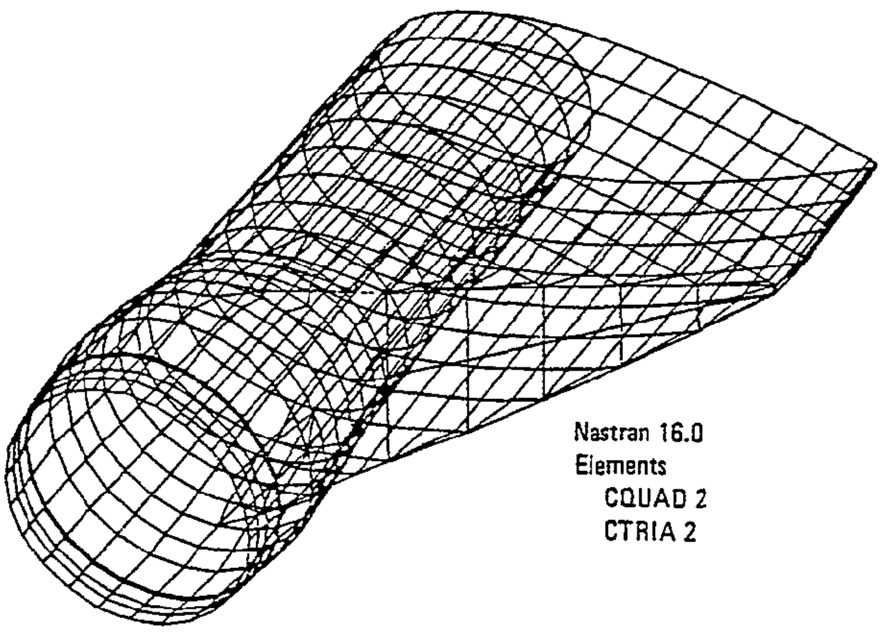
Normal Operating Range

Stall Flutter and Divergence

- Blade is free of stall flutter over full range of load and stability cases
- Blade is free of static divergence
- Blade is dynamically divergent under load case 2



Blade Retention



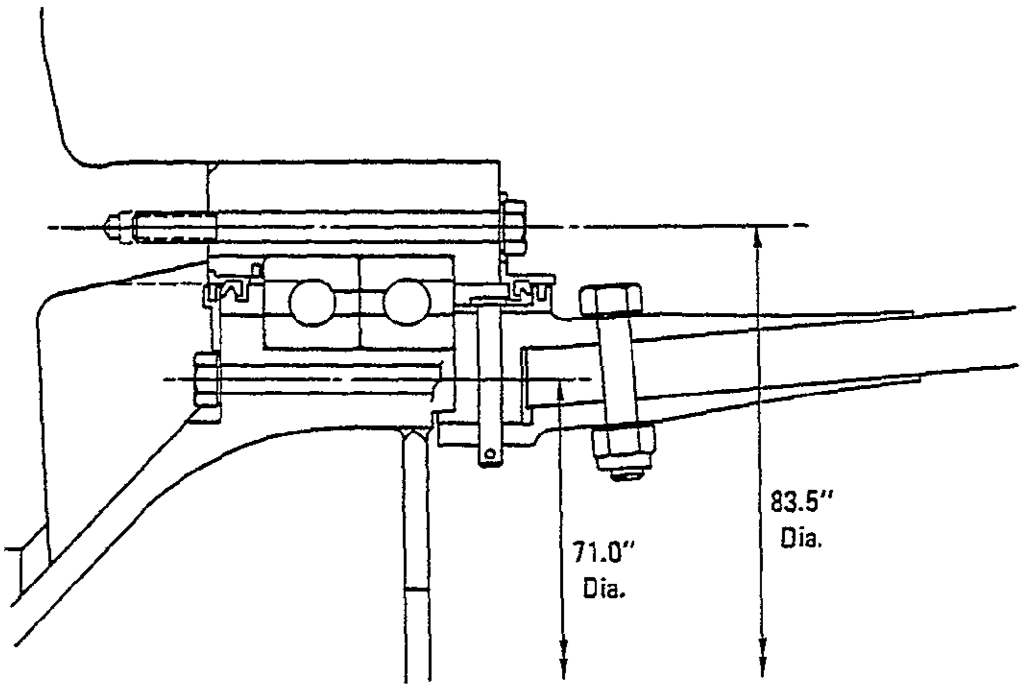
Inboard Blade Nastran Model

Blade Status

- Spar mandrel templates complete
- Shell mandrel templates started
- Retention ring material ordered
- System dynamic analysis started
- Estimated blade weight 15,100 lbs.

Blade Retention Bearing Types

- Three roller
- Duplex tapered roller
- Duplex angular contact ball



Blade Retention Bearing

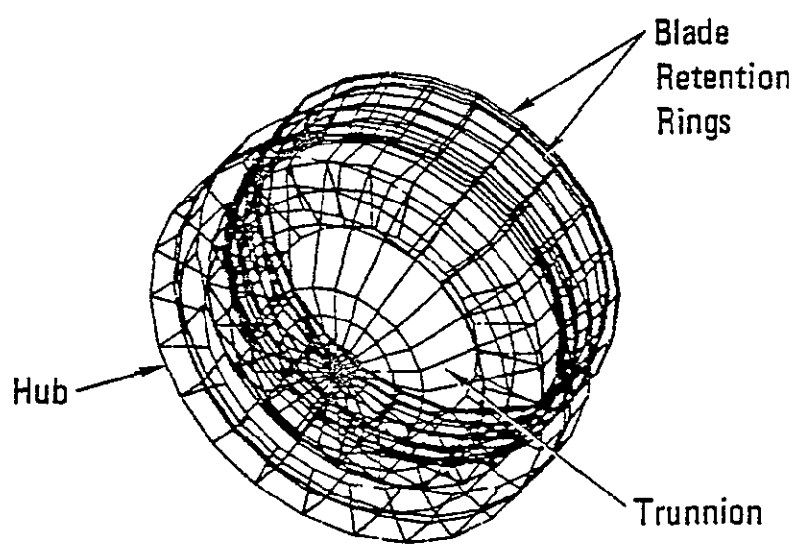
Bearing Design Requirements

30 yrs. life for following conditions

Condition I	22 MPH steady state 5×10^9 cycles
Condition II	22 MPH gusting to 60 MPH 1×10^6 cycles
Condition III	50 MPH steady state 20° inflow angle 5.5×10^8 cycles

Bearing Load Summary

	Moment In-Lbs	Centrifugal Load Lbs	Side Load Lbs
Condition I	10.9×10^6 max 1.6×10^6 min	176,000	25,540 max 9,500 min
Condition II	40×10^6 max 0 min	176,000	58,280 max 3,130 min
Condition III	29.1×10^6 max 8.3×10^6 min	176,000	50,000 max 15,300 min



Blade Retention Bearing Model

Bearing Status

Bearing type selected

Loading defined

Bearing specification sent to vendors

Bearing vendors have responded

Glass Roving Reinforcement for Wind Turbine Blade

Roving Designation – PPG's type 1062 NT-15

Filament

Material	– E-glass
Diameter	– 50-55 x 10^{-5} inch
Tensile strength	– > 225,000 PSI

Strand

Type	– ECK 37
Strand ten. strength	– > 200,000 PSI

Roving

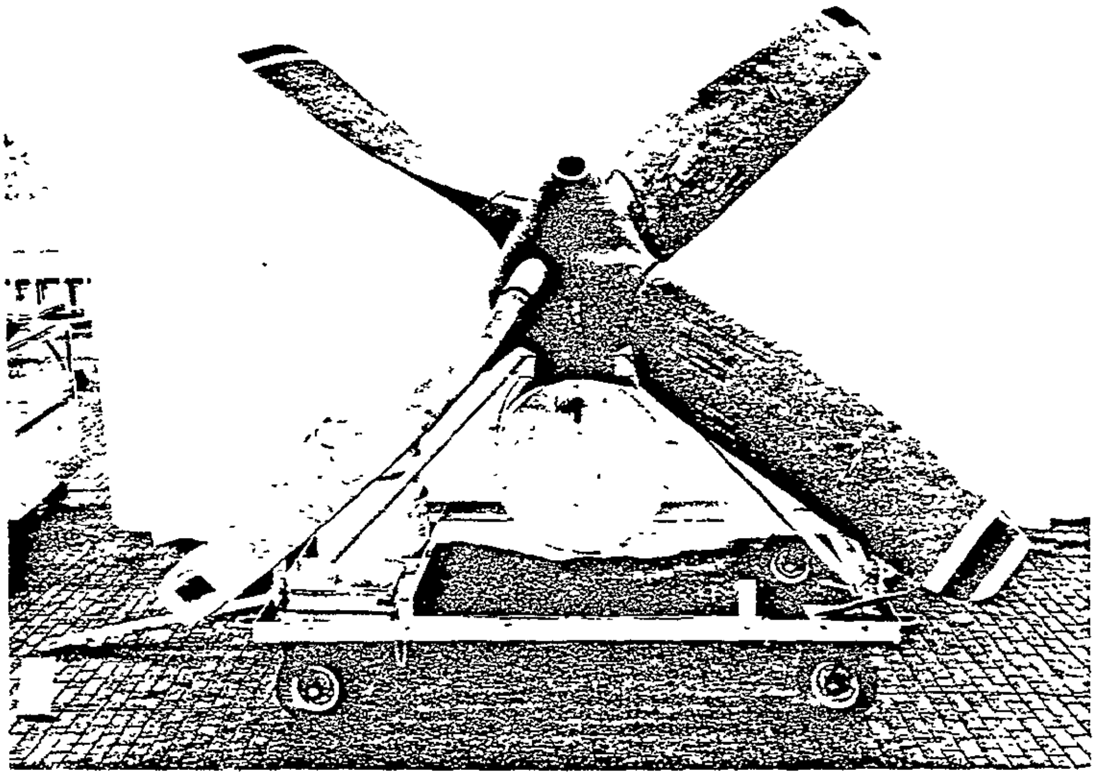
No. of strands	– 15
No. of filament ends	– (15) (408 filaments/strand) = 6,120
Nominal yield	– 247 yds./lb.

Sizing

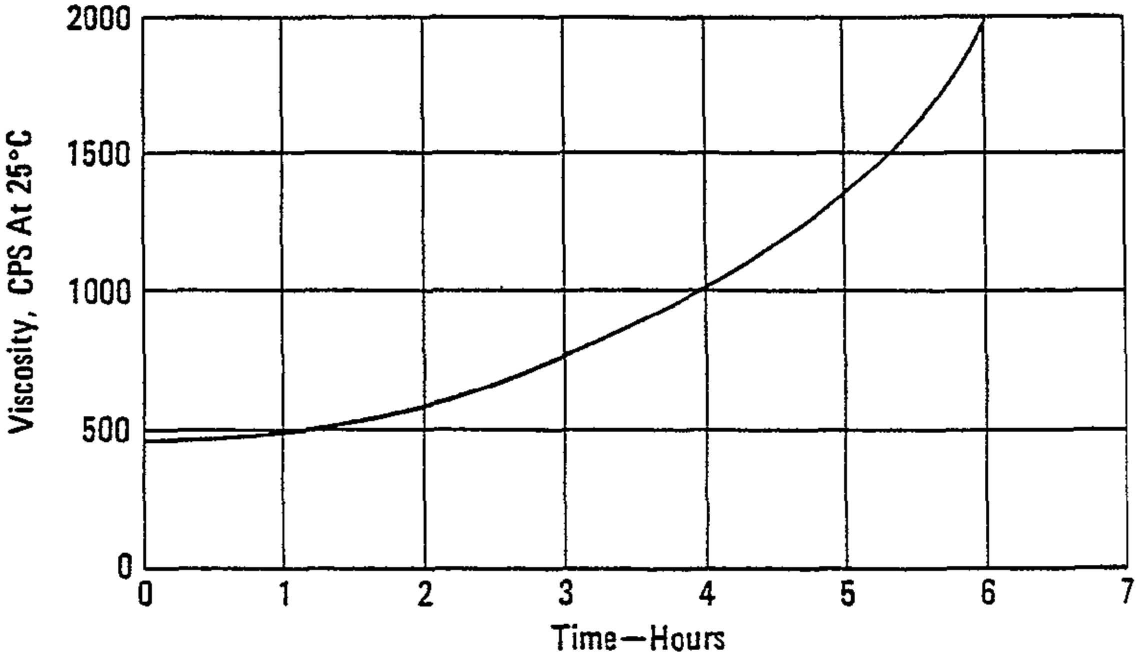
Chrome-silane

Resin Materials

Composition	Propeller Blade	Wind Turbine Blade
Epoxy resin	ERL 2256	EPON 826
Hardener	Sonite 41	Jeffamine D230
Epoxy/hardener ratio	80/20 pbw	100/30 pbw
Viscosity, CPS @ 70°F	910	450
Pot life, hrs.	7	8
Cure	1.5 hrs. @ 200°F and 2.0 hrs. @ 300°F	16 hrs. @ 176°F
HDT, °F	290	185
Mechanical Properties		
Tensile strength, PSI	9,000	10,600
Tensile modulus, PSI x 10^6	0.57	0.43
Ultimate elongation, %	2 4	7.7
Flexural strength, PSI	15,000	17,600
Flexural modulus, 10^6 PSI	0.50	0.43



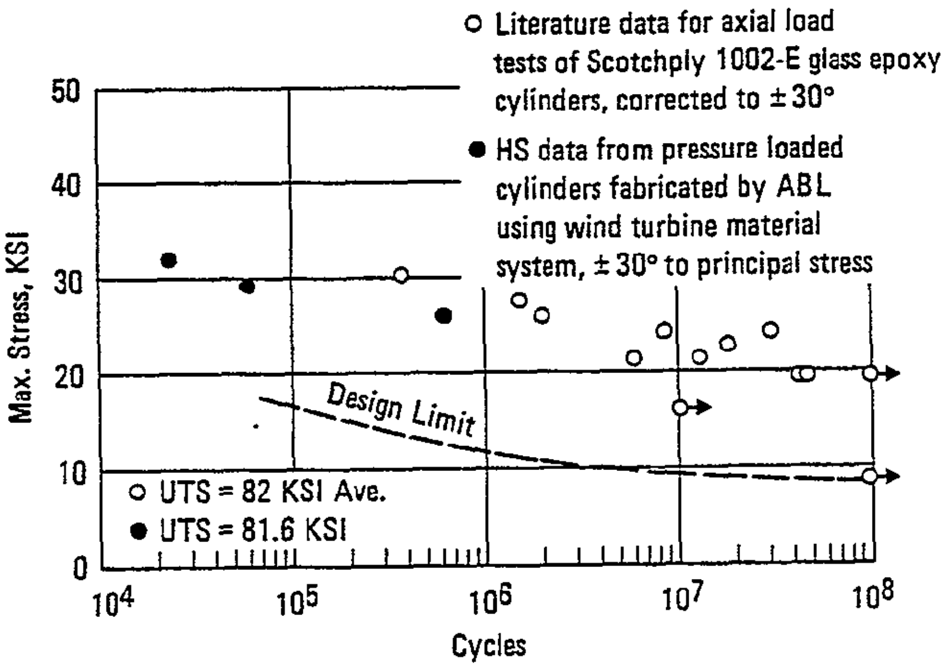
Fiberglass Propeller



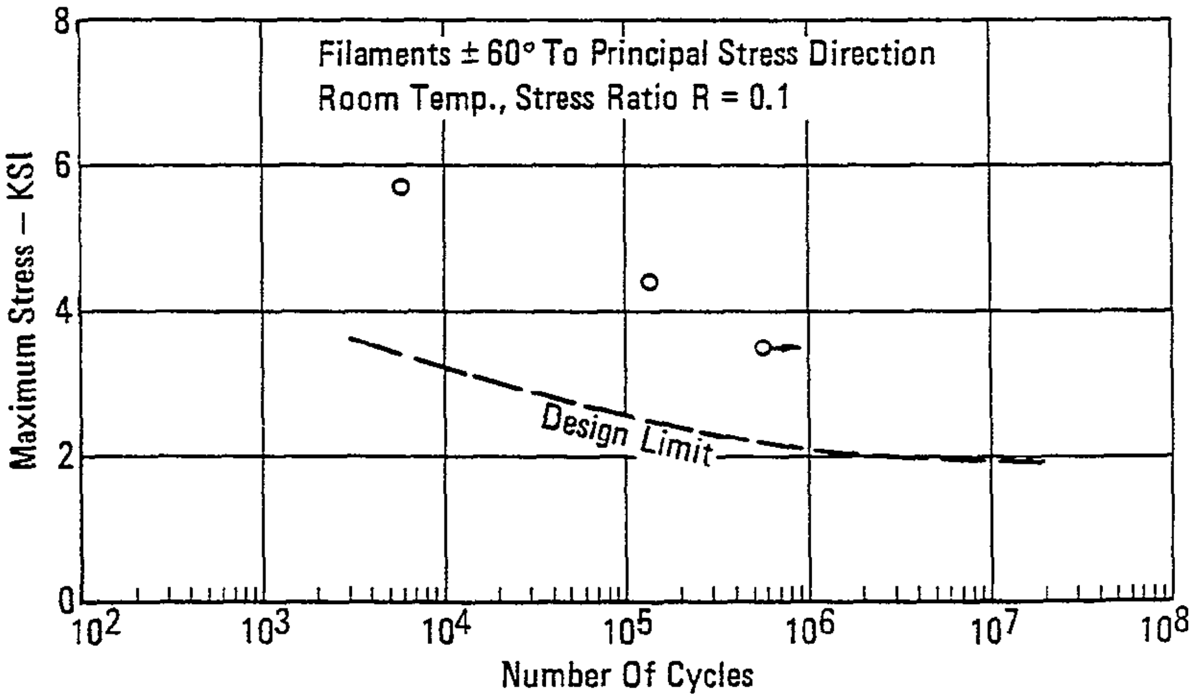
Resin Viscosity vs Time

Protective Coatings

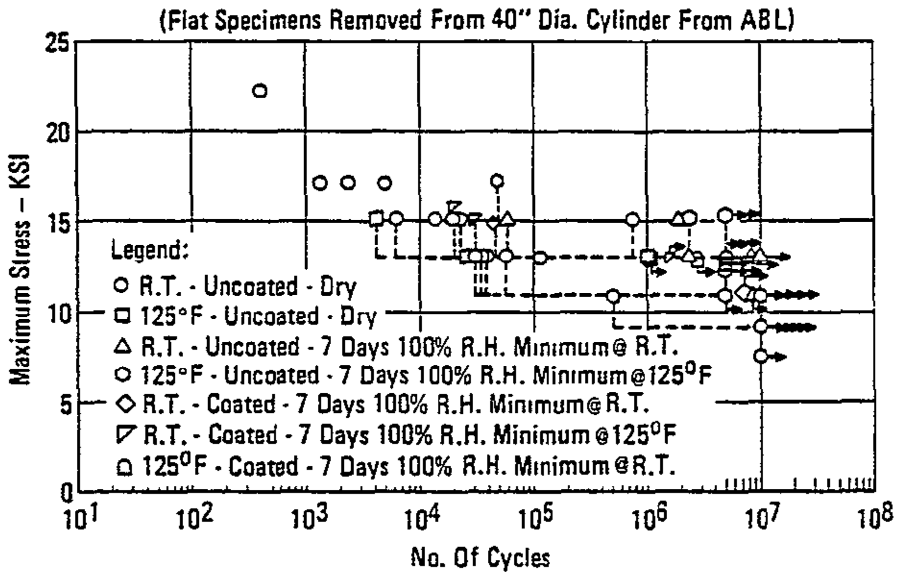
Blades and spinners		Laminar 8-B-6 with conductive carbon filler (anti static coating)
	Spinners	Thickness – 5 to 7 mils
	Blades	Thickness – 5 mils min.
Wind turbine		Laminar 8-B-6 with conductive carbon filler
		Thickness – 5 to 7 mils
		Mil-C-81773 urethane coating (insignia white)
		Thickness – 1 to 2 mils



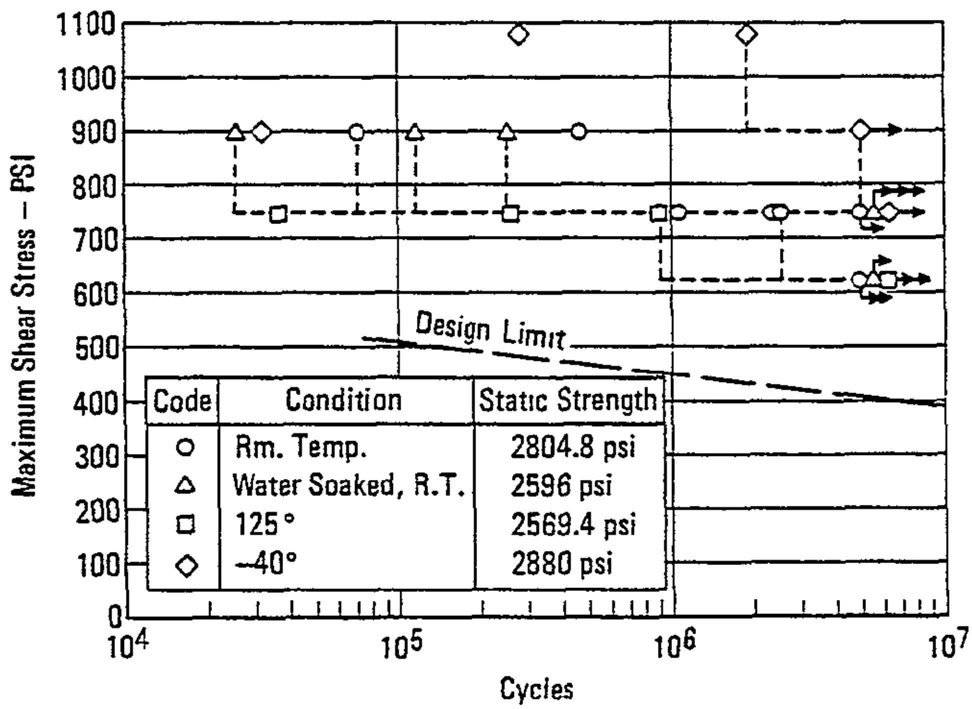
Test Results from Cylinders
(Room Temp., Stress Ratio R = 0.1)



Fatigue Tests Results from Cylinders Wound by ABL Using Wind Turbine Blade Materials System



Tension-Tension Fatigue Test Data for $\pm 30^{\circ}$ Filament Wound Composite
R = 0.1 Room Temp.



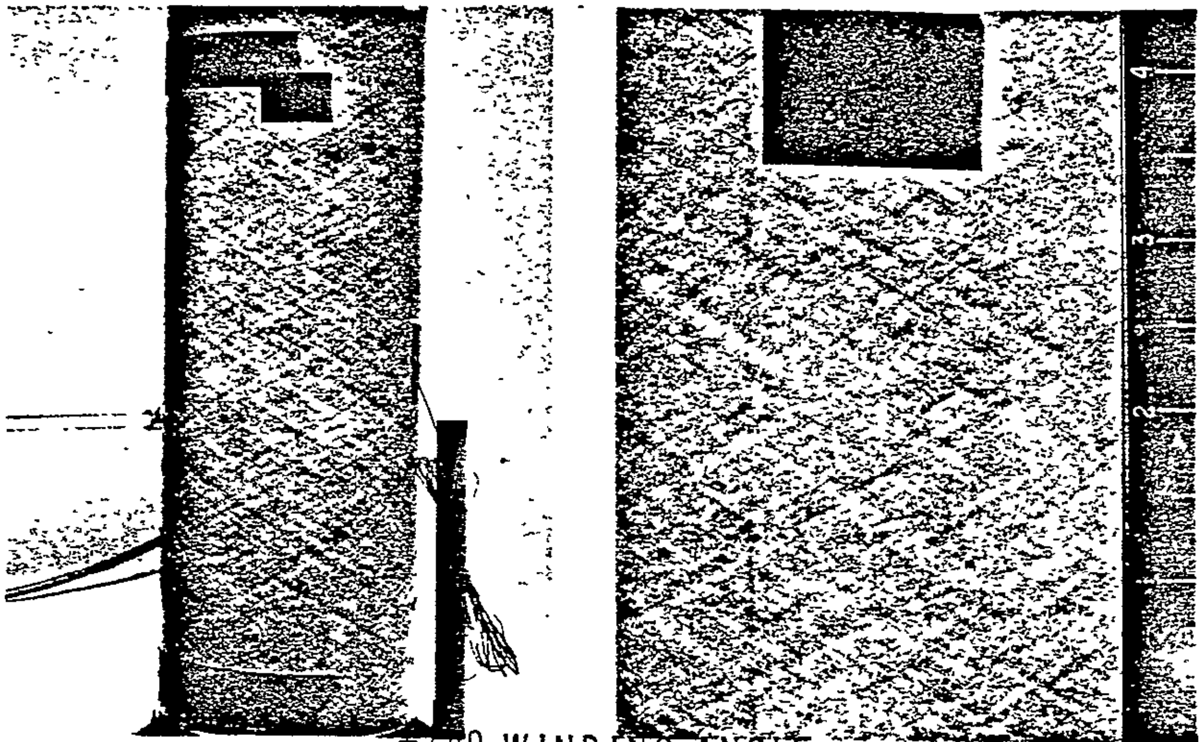
Interlaminar Shear Strength of Region of Interrupted
Manufacturing During Filament Winding

Moisture Effects Comparison

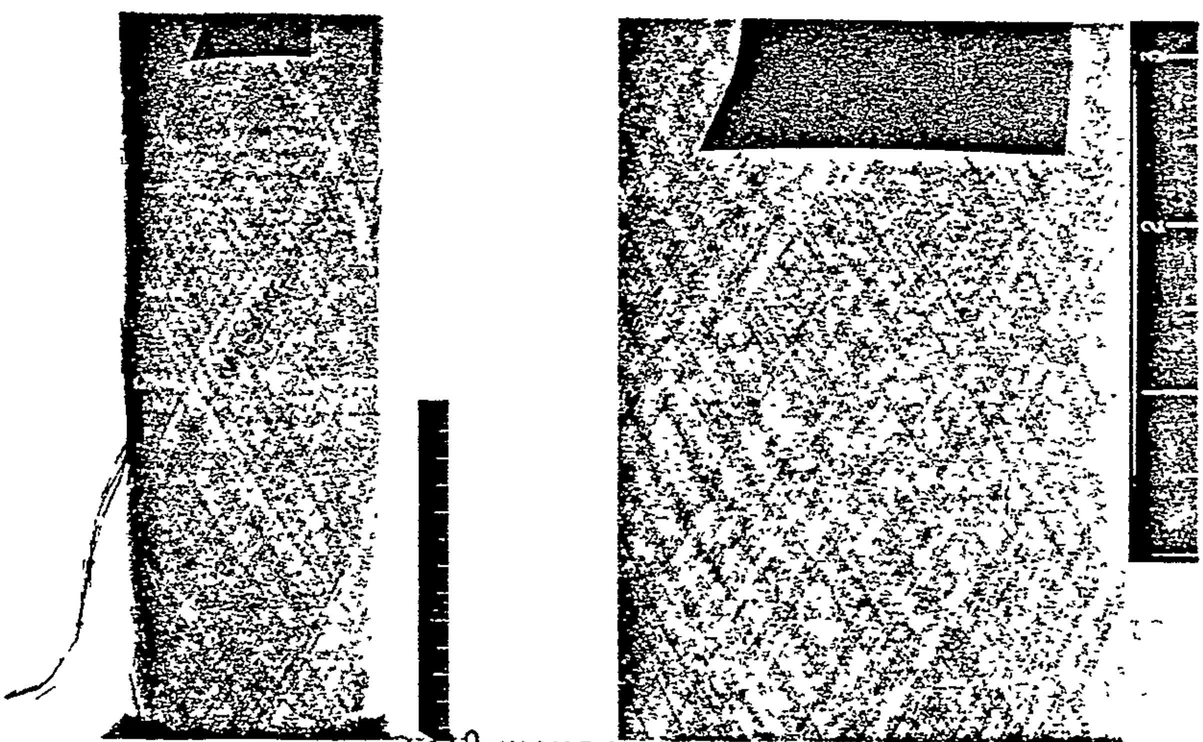
WIND TURBINE RESIN VS 300°F TEMPERATURE CURED RESIN USED IN PROPELLER BLADE SHELLS; ± 45° E-GLASS CLOTH REINFORCEMENT IN BOTH RESINS

Material	Cure	Pre-Test Treatment	% Moisture Absorption	(1) S.B.S. Ra
Epon 826/Jeffamine D-230	2 hrs @ R.T. 16 hrs @ 176°F	24 hr. distilled H_2O boil	0.79	0.61
APCO 434/Sonite-41	1½ hrs @ 200°F 2 hrs @ 300°F	24 hr. distilled H_2O boil	0.67	0.69

(1) Short beam shear strength ratio, 24 hour water boil specimen test results divided by as-cured specimen test results. Tests conducted in room temperature air.

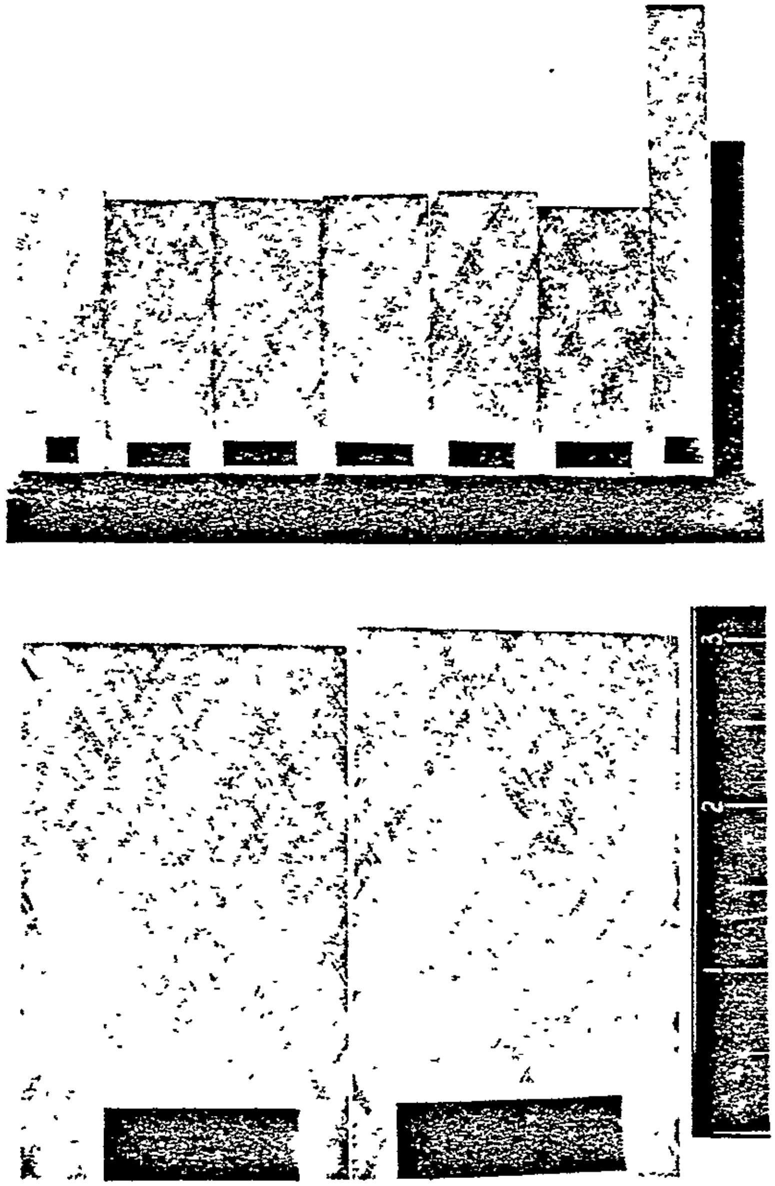


±60° WINDING ANGLE

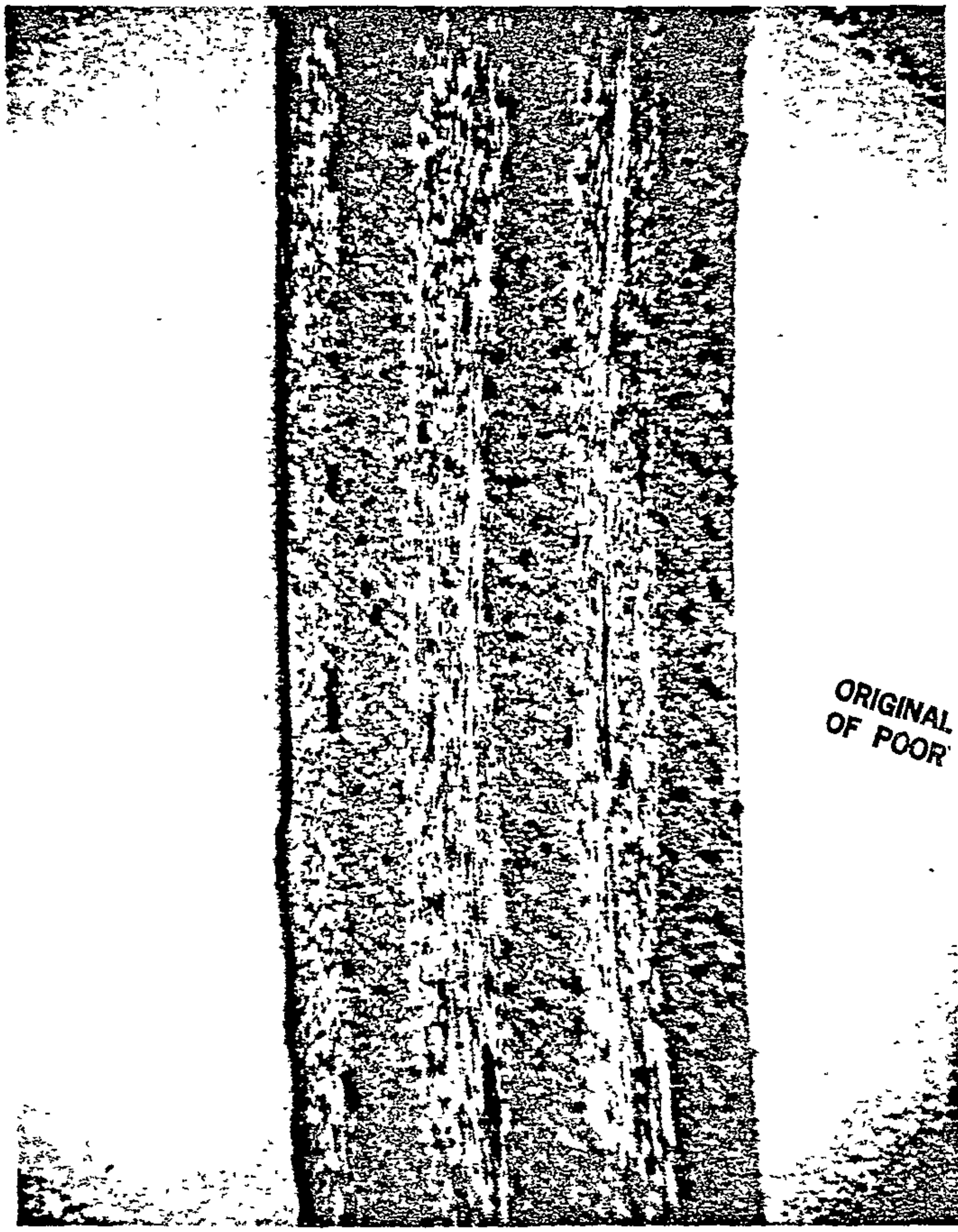


±30° WINDING ANGLE

Bright Light Inspection Records



Bright Light Inspection Records
40-Inch Diameter Drum Flat Specimens



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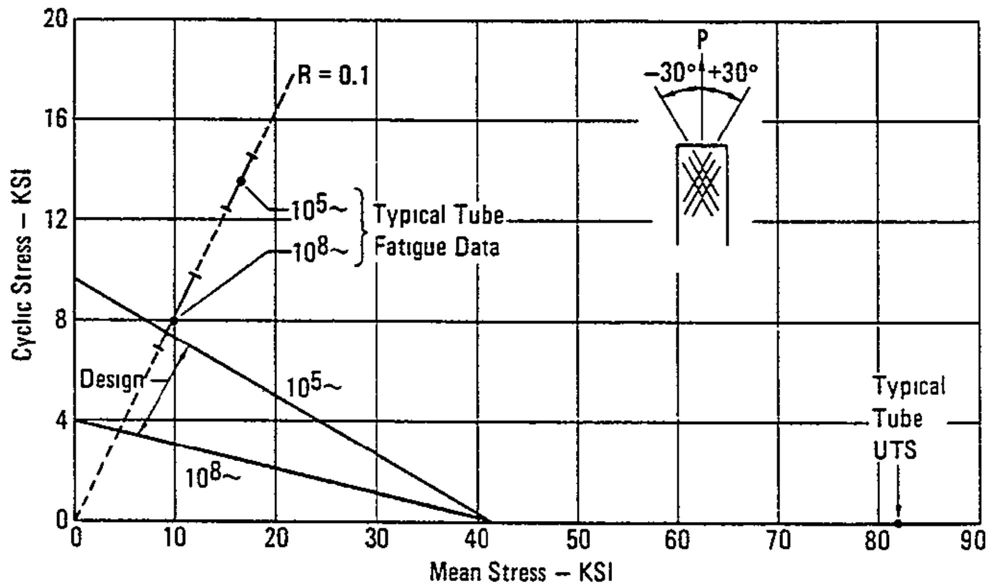
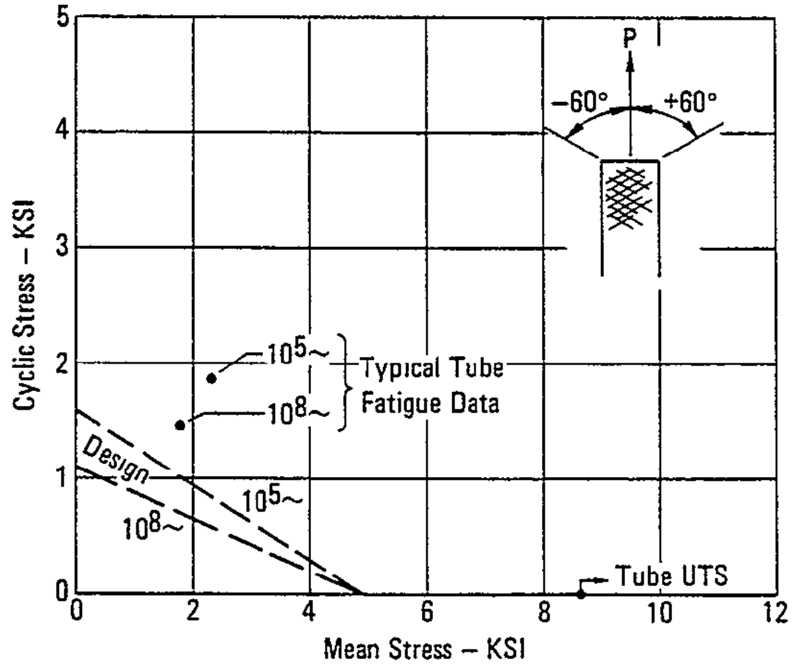
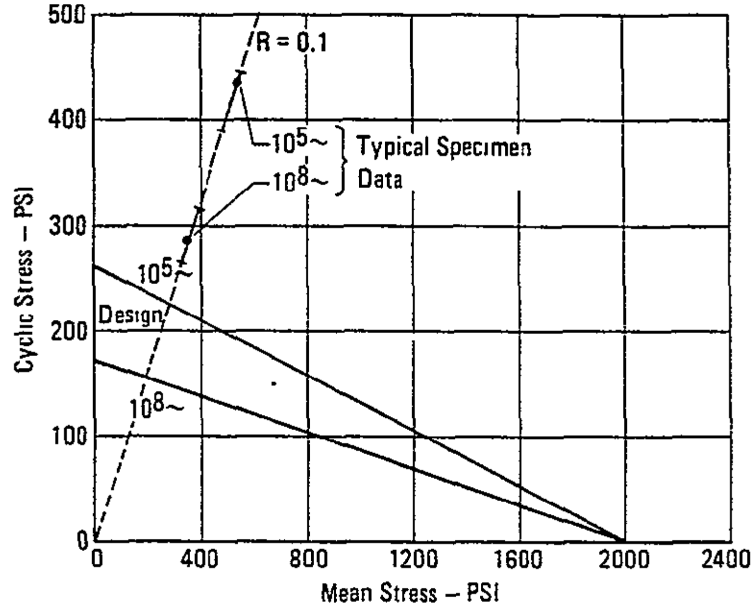
Wind Turbine Blade Filament Wound E-Glass/Epoxy
4.79 Inch Diameter Test Cylinder



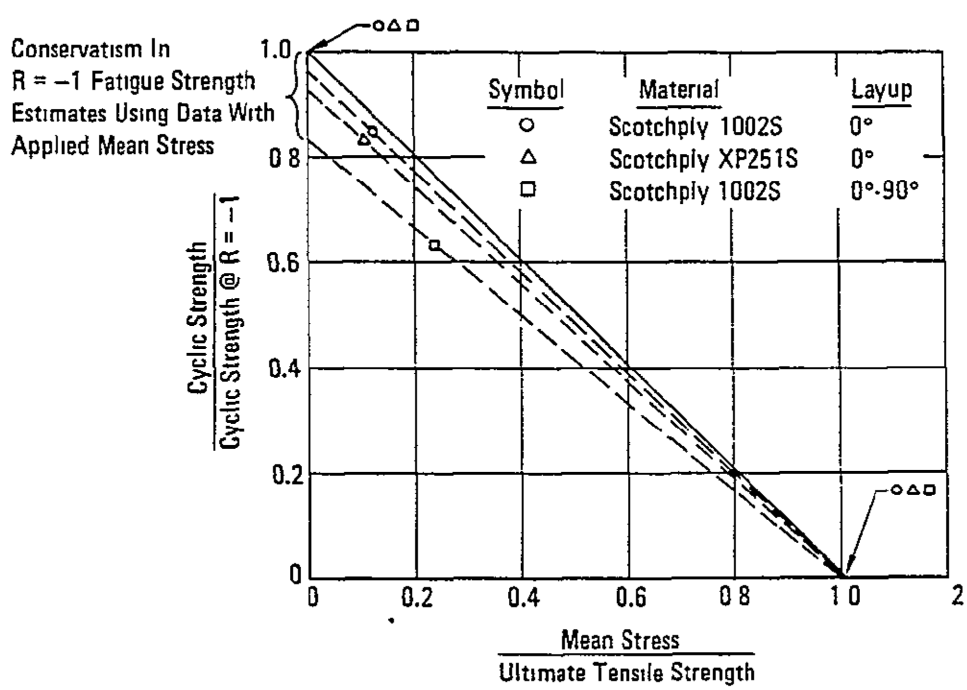
Modified Goodman Diagram for Wind Turbine Blade Material
Room Temperature Properties



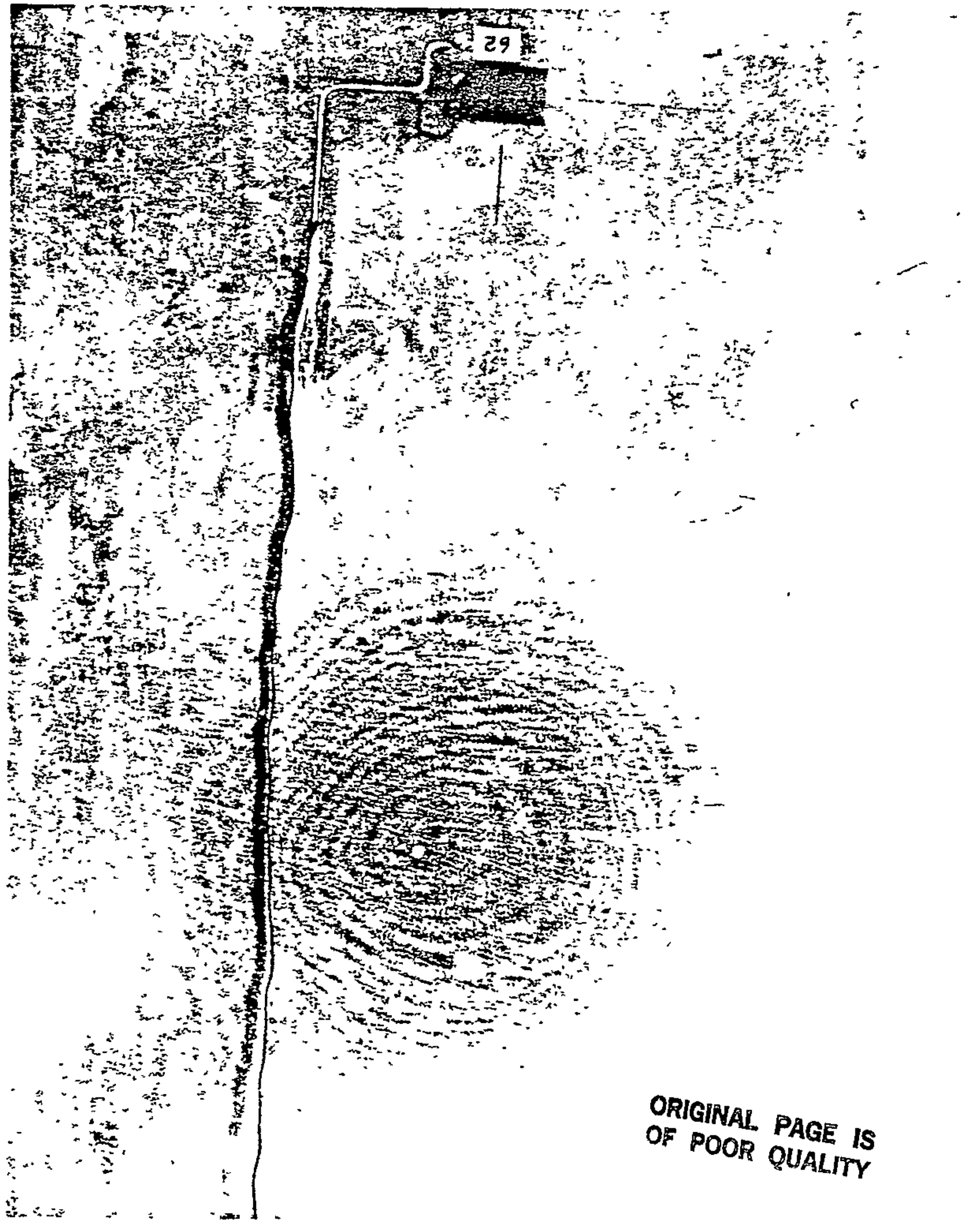
Modified Goodman Diagram for Wind Turbine Blade Material
Room Temperature



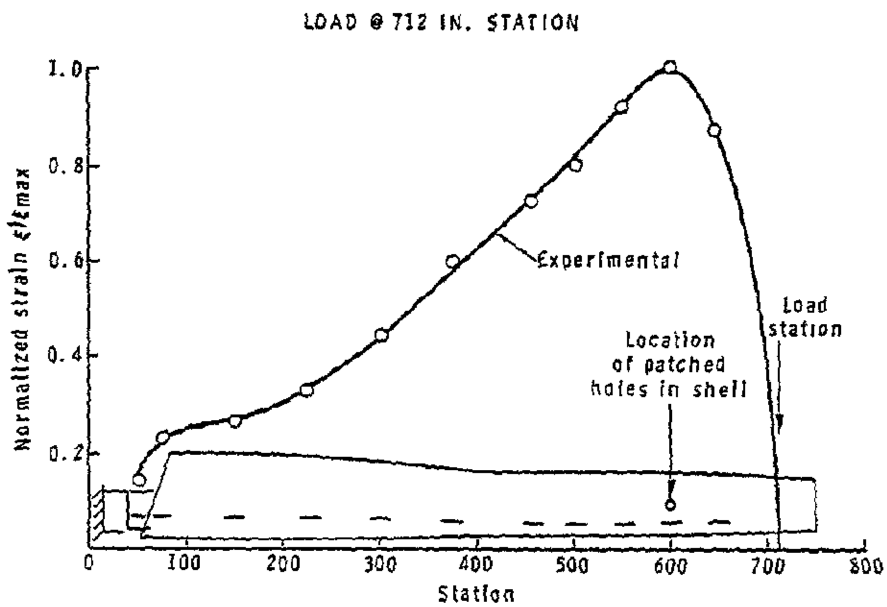
Modified Goodman Diagram for Interlaminar
Shear Strength at Point of Manufacturing Interruption
Room Temperature



FRP Fatigue Strength
10^7 Cycles



Manufacturing Prototype Blade
Shell Repar 600" Station



Mod "0" Manufacturing Prototype Blade Longitudinal Strain Distribution

Mod-0/Mod-1 Common Technology

- Integrally wound monolithic structure
- Spar/shell design
- Integrally bonded inner adapter sleeve
- Bonded outer adapter sleeve
- Collapsible spar and shell mandrels
- Filament winding technique

Technology Needs for Increased Length

- Mandrel deflection assessment
- Mandrel removal demonstration
- Filament winding procedures scaled to larger size
- Dimensional accuracy assessment
- Weight prediction verification
- Material strength demonstration

Above needs addressed in demonstration
spar program

Mod-1 Blade Manufacturing Process Controls

To include

- Materials receiving inspection
- Winding program control
- Resin mixing
- Mandrel assembly & preparation
- Layup spot check
- Weight control
- Cure time & temperature
- Component detail inspection

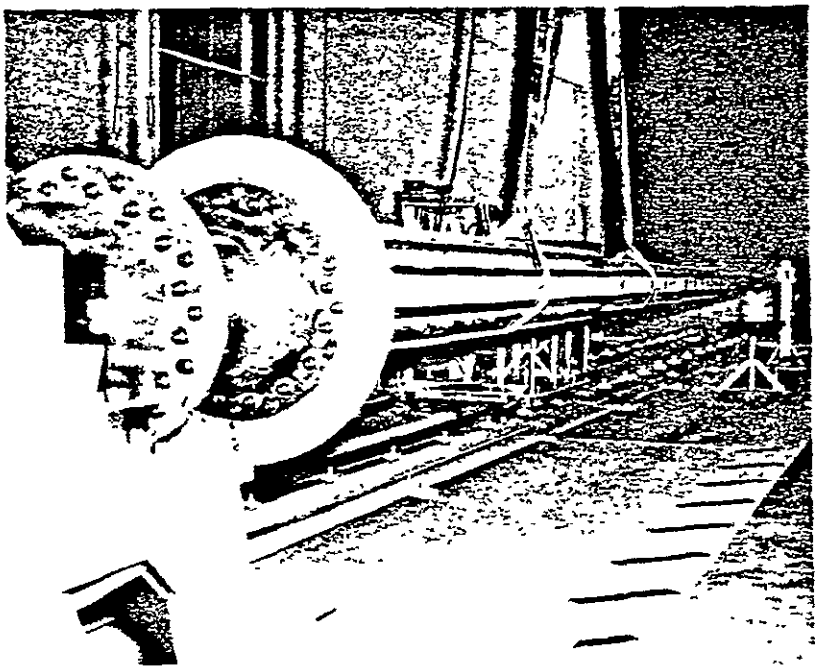
Wind Turbine Blade Weight Control

- For each ply drop section:

  Record weight of fiber & resin used
  Subtract weight of wet & dry scrap
  Subtract est. wt. of end scrap
  Compare with control weight

- Adjust as necessary in subsequent layers by:

  Adjusting resin content within tolerance
  Adjusting ply length within tolerance



Mandrel Shaft Installation

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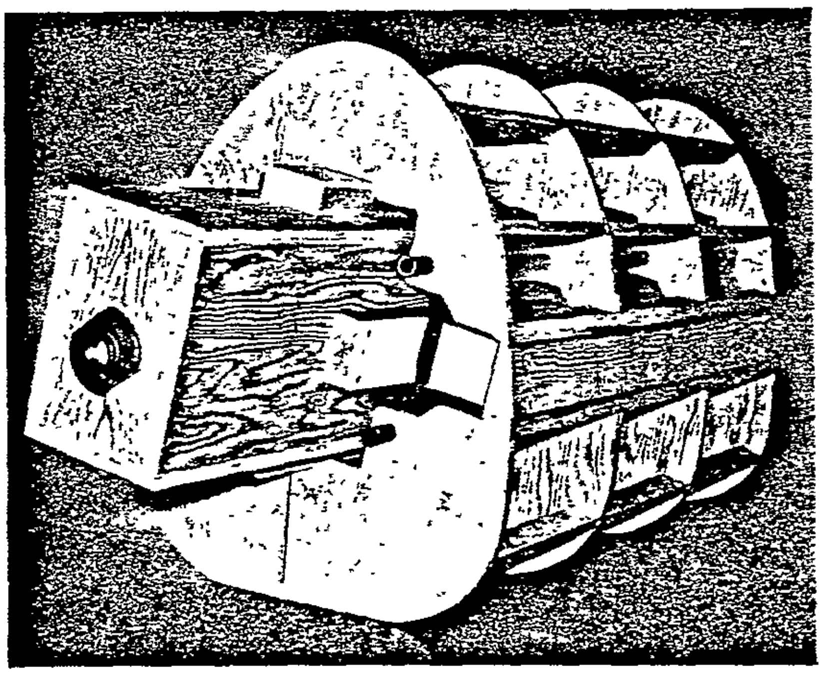
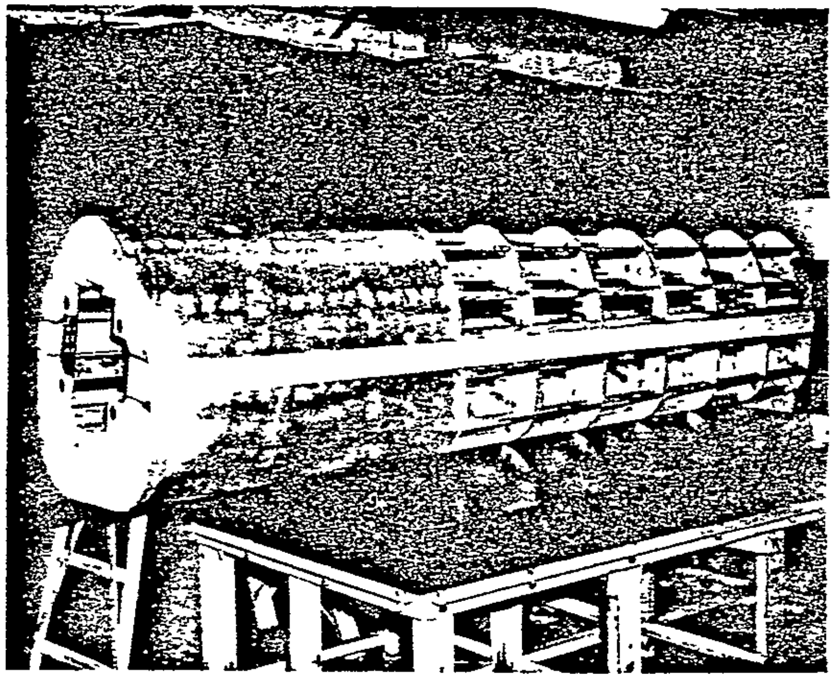
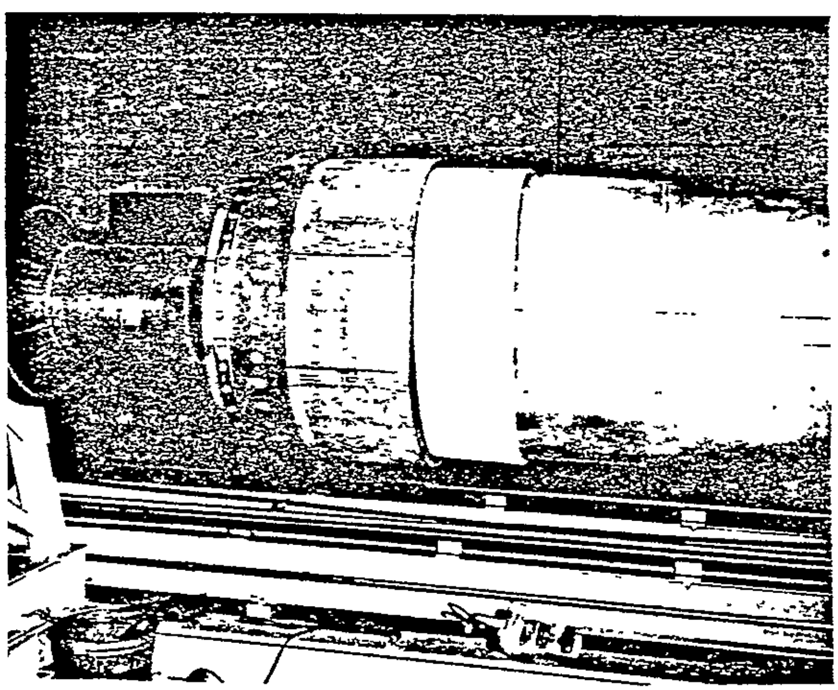


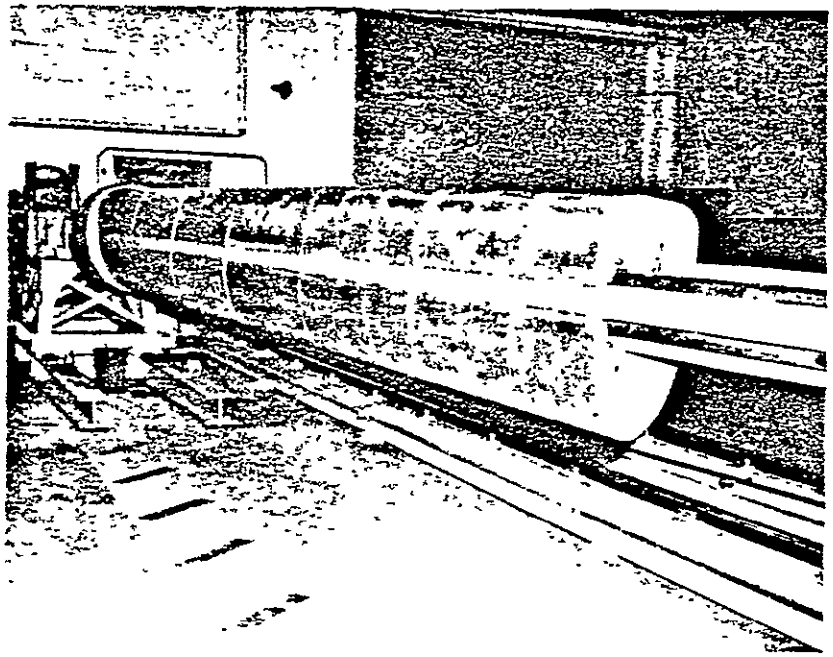
Spar Mandrel Former



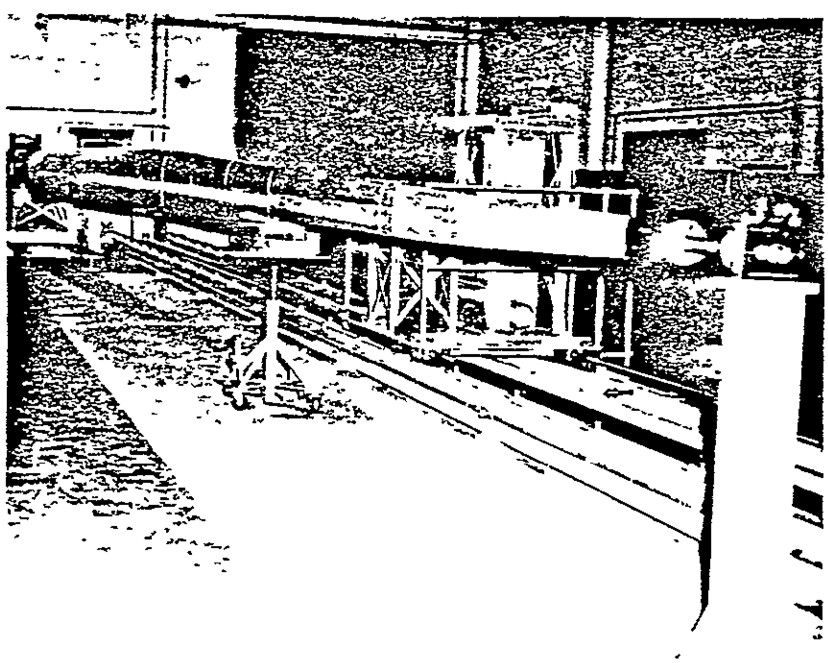
Spar Mandrel Former Assembly



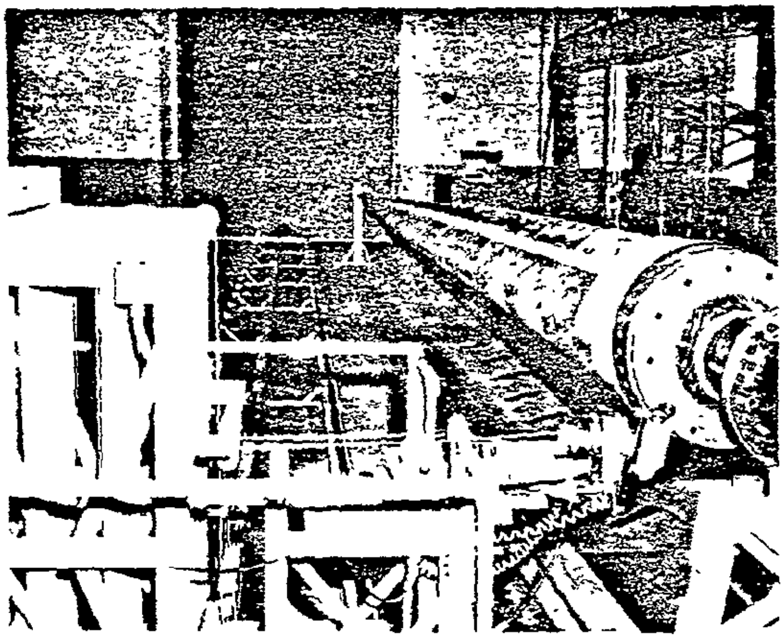
Spar Retention to Airfoil Transition



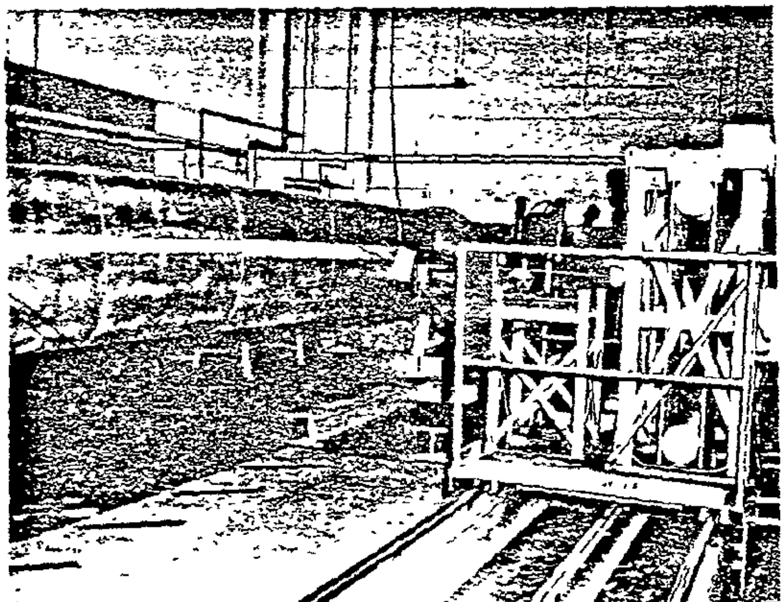
Spar Mandrel Buildup



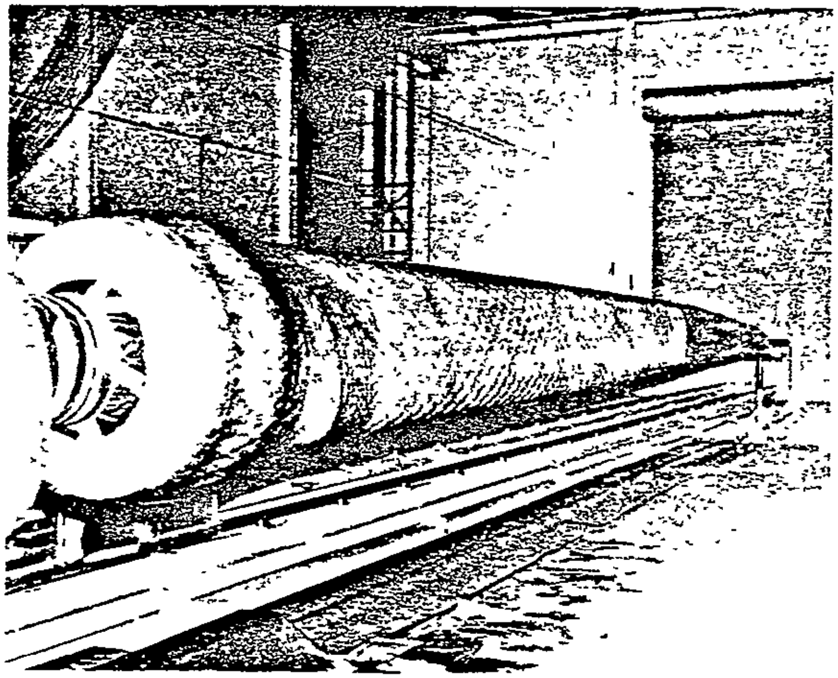
Complete Spar Mandrel



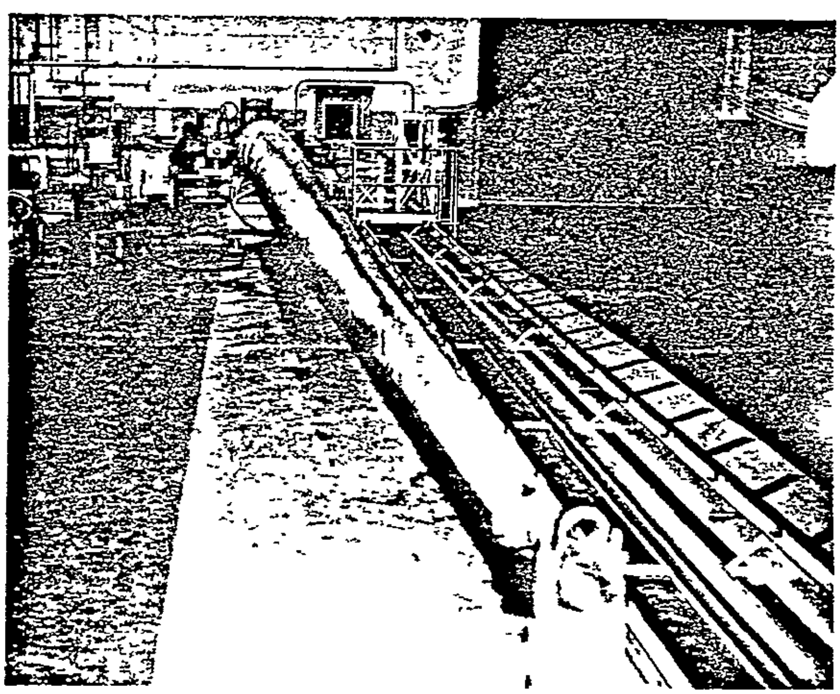
Spar Trial Winding



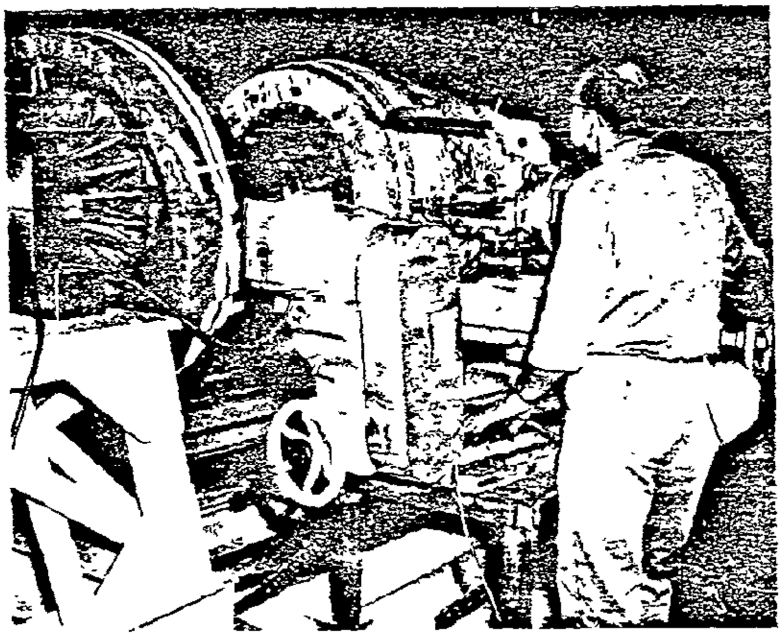
Filament Delivery System



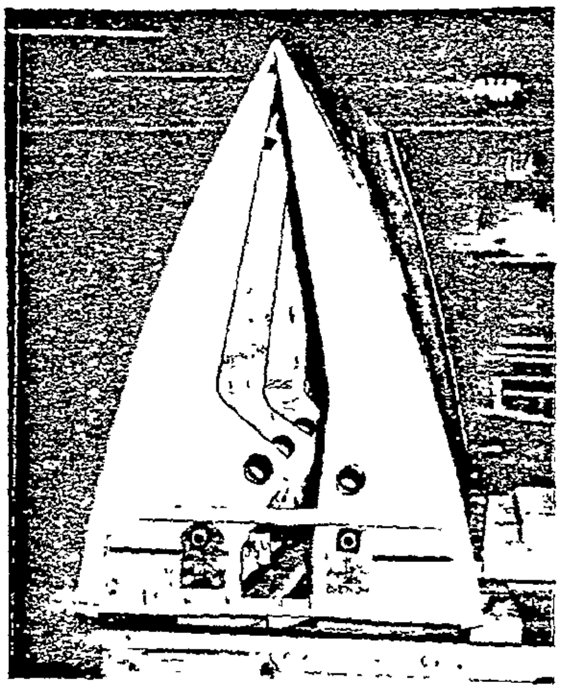
Spar Wound to 345-Inch Station



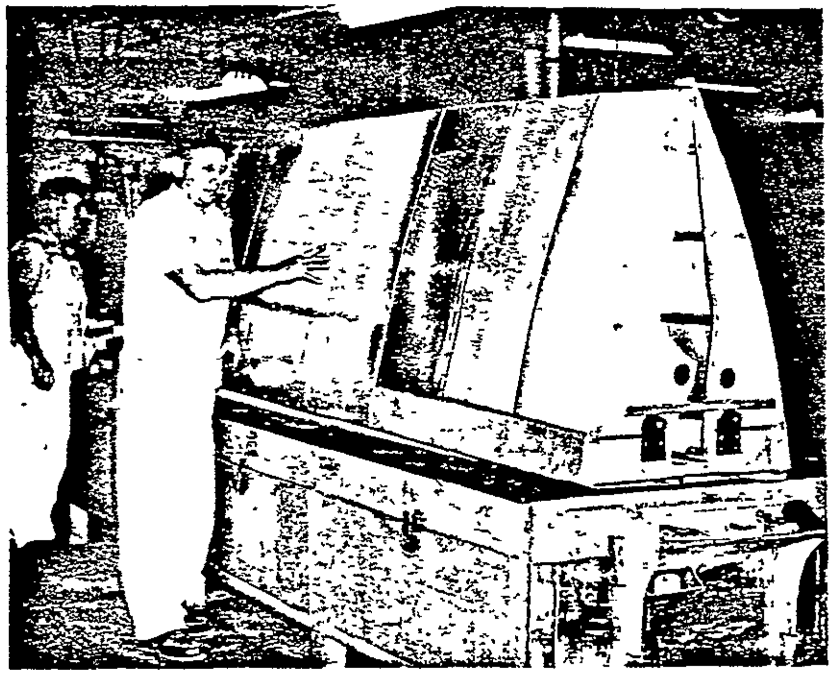
Finish Wound Spar with Peel Ply



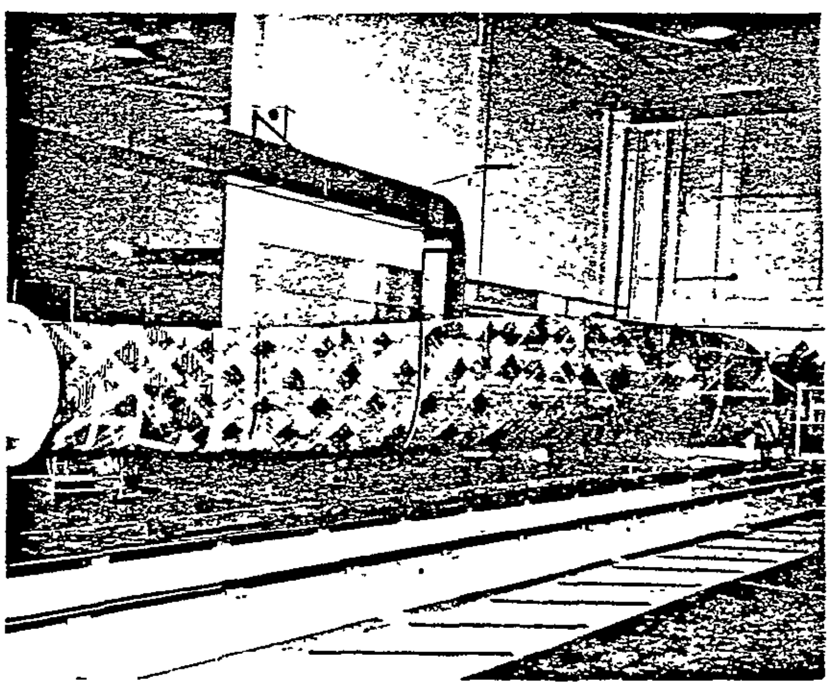
Retention Machining



Shell Mandrel Former



Shell Mandrel Former Assembly

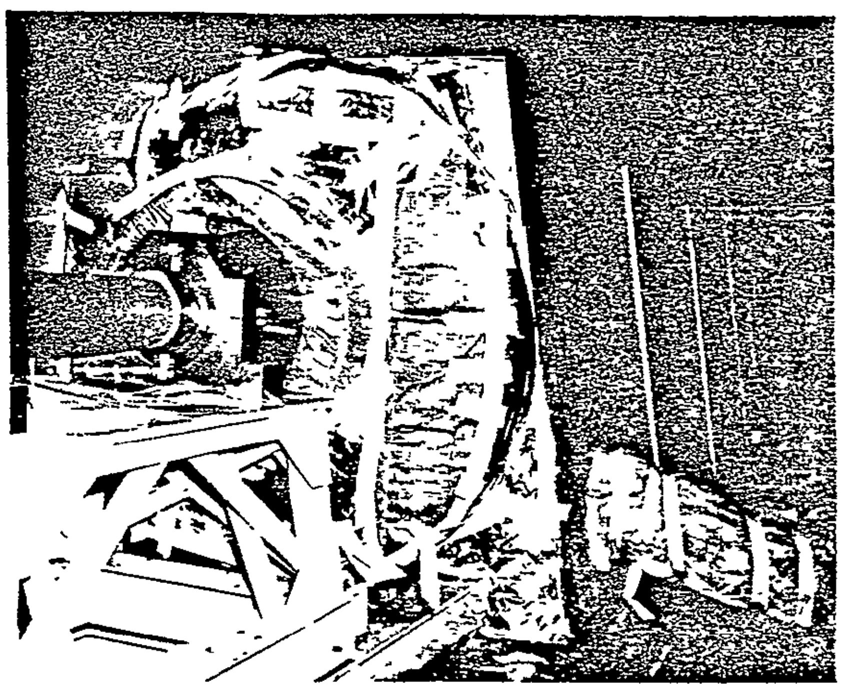


Shell Helical Winding

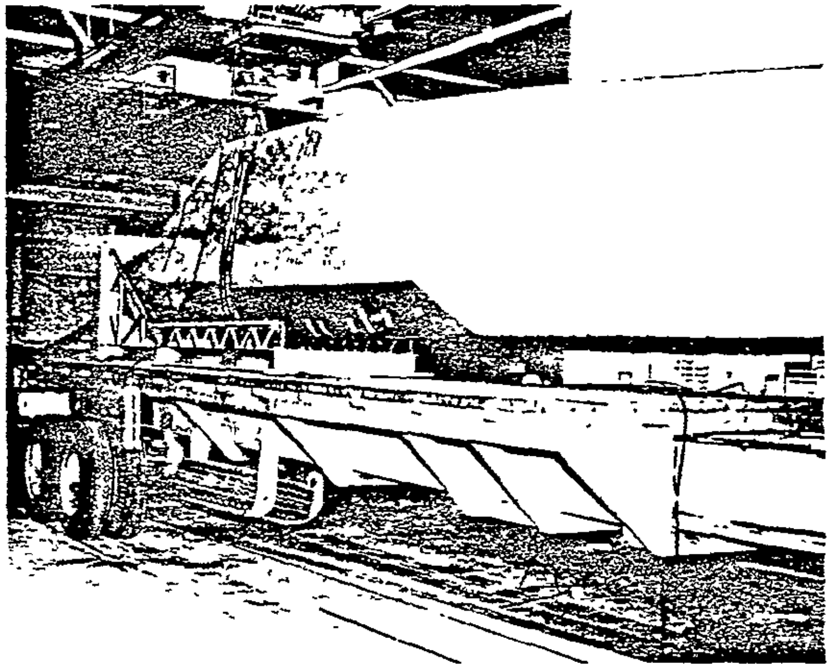
- Use low circuit patterns

Reduces cross overs
Eliminates band buildup voids
Verified on demo span

- Increase shell layup angle
- Machine modifications



Final Blade Cure



Finished Blade in Shipping Fixture

Demonstration Spar Program

Introduced to reduce jeopardies

- Large filament wound structures
- Mandrel deflection
- Mandrel removal

Specific Objectives

Verify:

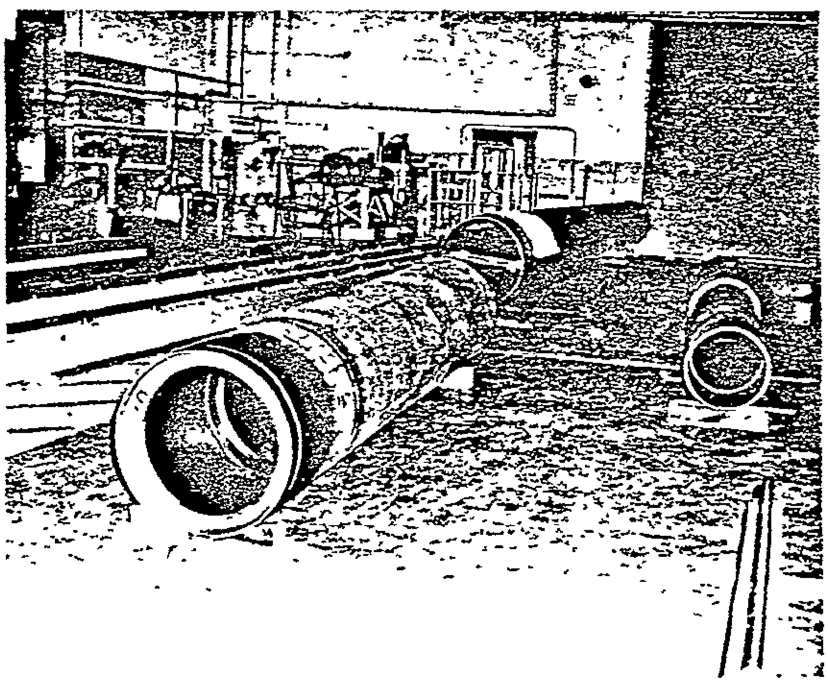
- Mandrel deflection
- Mandrel removal
- Filament winding process
- Dimensional accuracy
- Weight
- Spar physical properties

Spar Definition

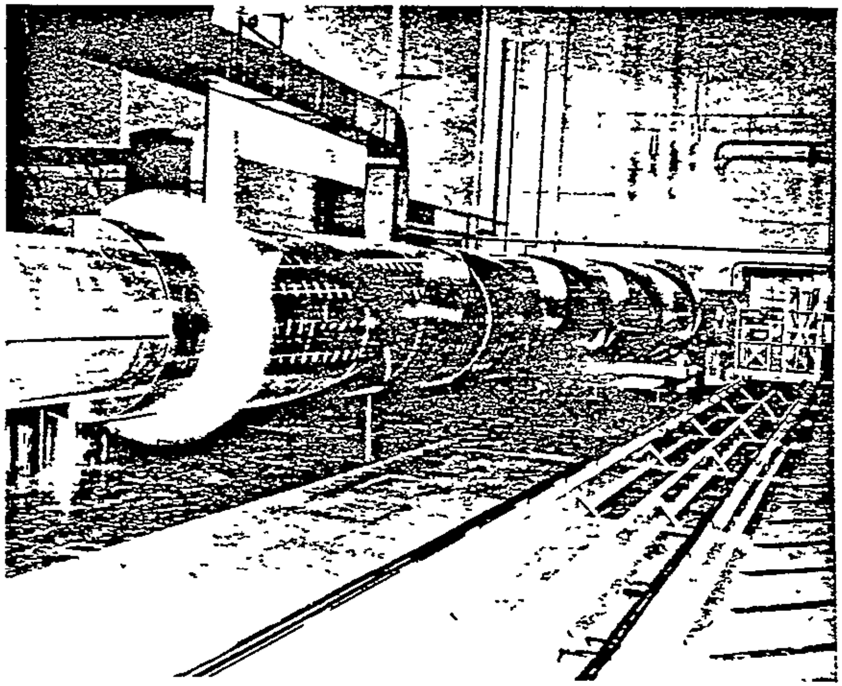
Length	98.5 Ft.
Weight	7800 Lb.
Winding angle	± 30°
Root size	67 In. x 58 In.
Maximum wall thickness	1.0 In.

Program Accomplishments

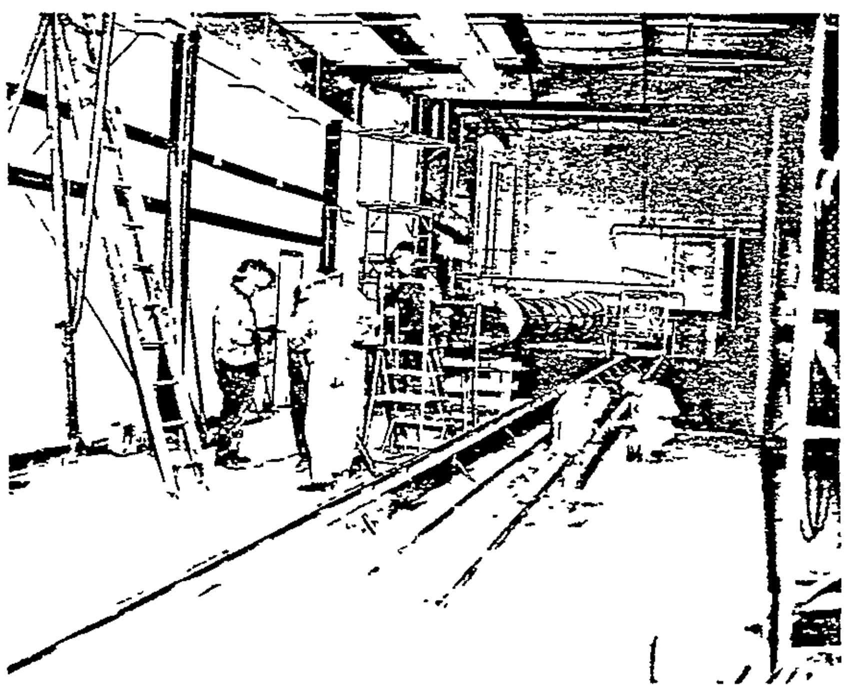
- Facilities complete
- Tooling complete
- Excellent filament winding patterns developed
- Predicted mandrel deflection confirmed
- Mandrel removal successfully accomplished



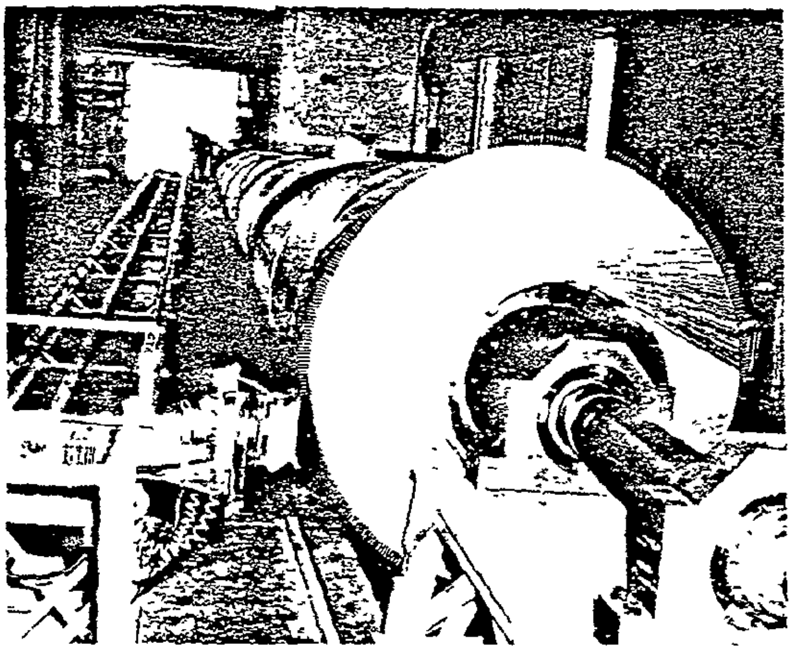
Center Mandrel Shaft Parts



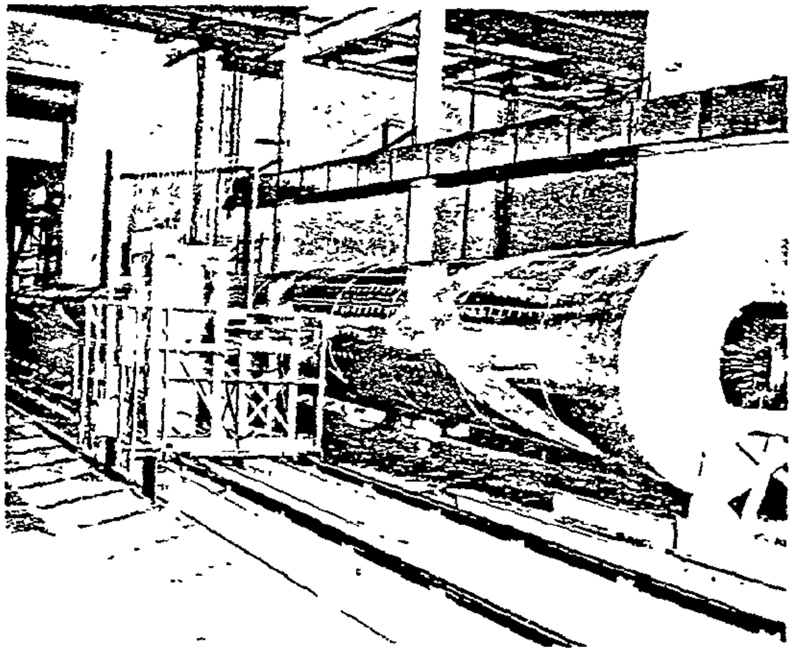
Dry Trial Winding to Second Generating Disk



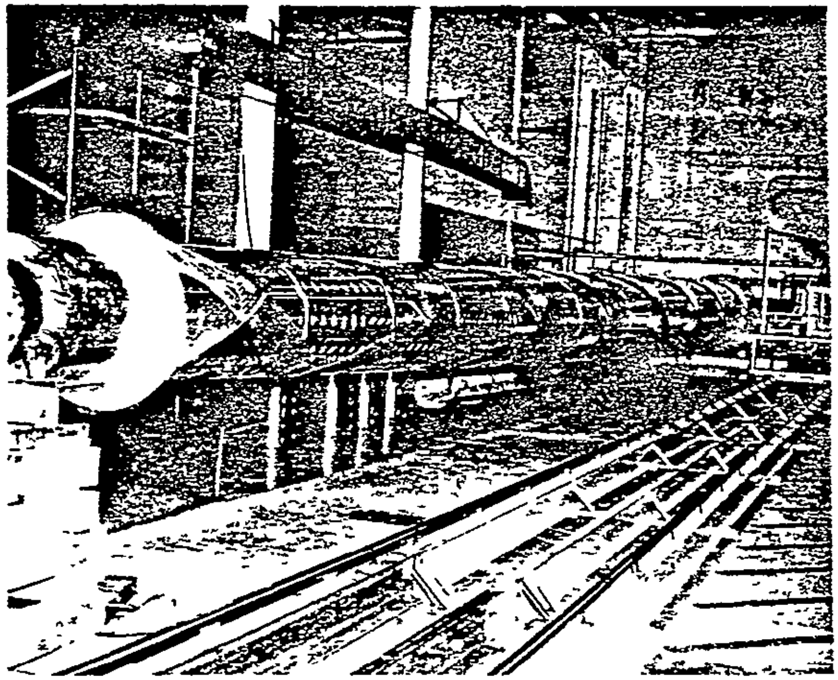
Mandrel Assembly



Dry Trial Winding Around Inboard Generating Disk



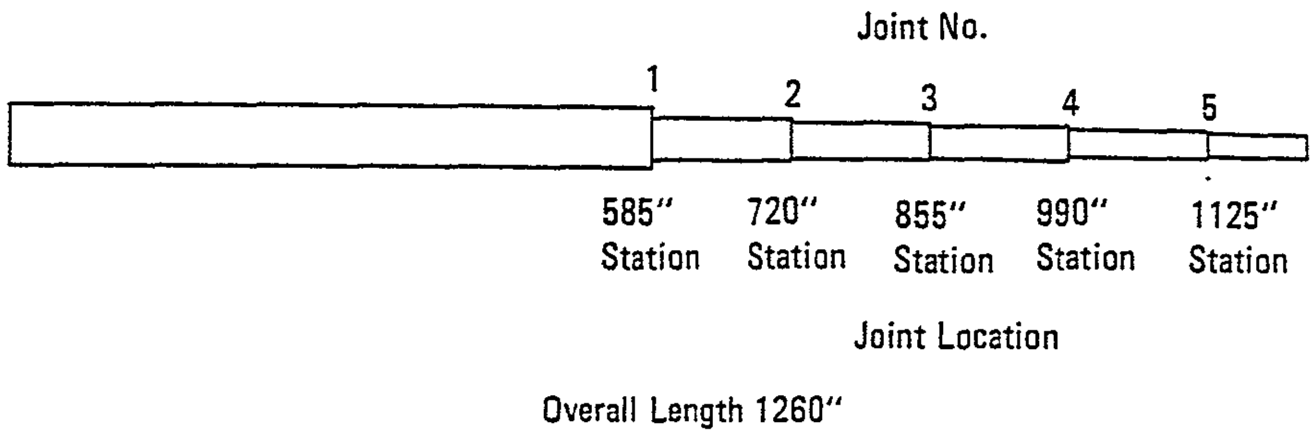
Dry Trial Winding Around Inboard Generating Disk



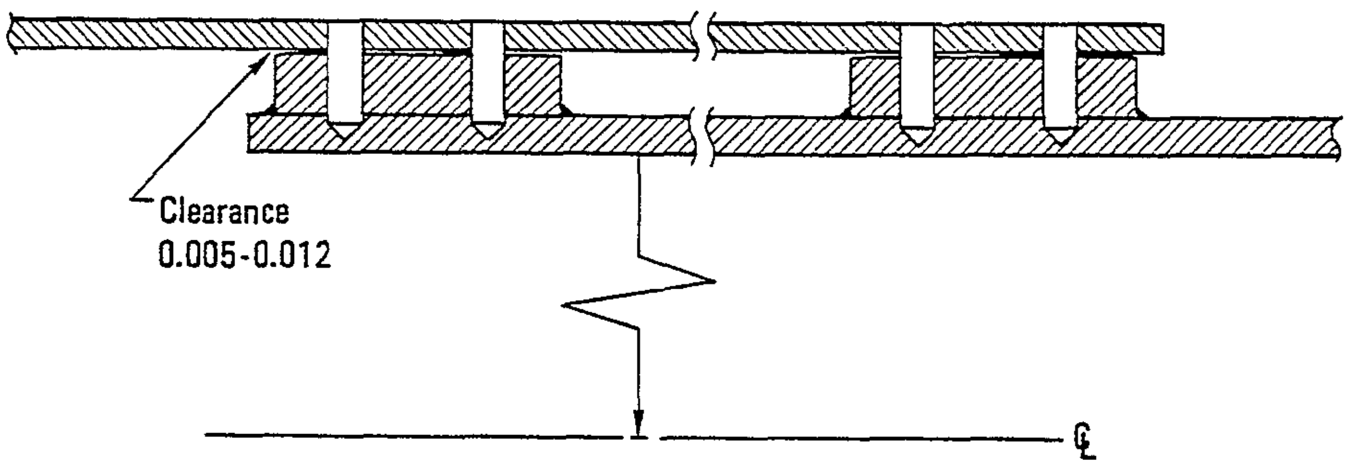
Dry Trial Winding Around Outboard Generating Disk

--- Demonstration Spar Problem

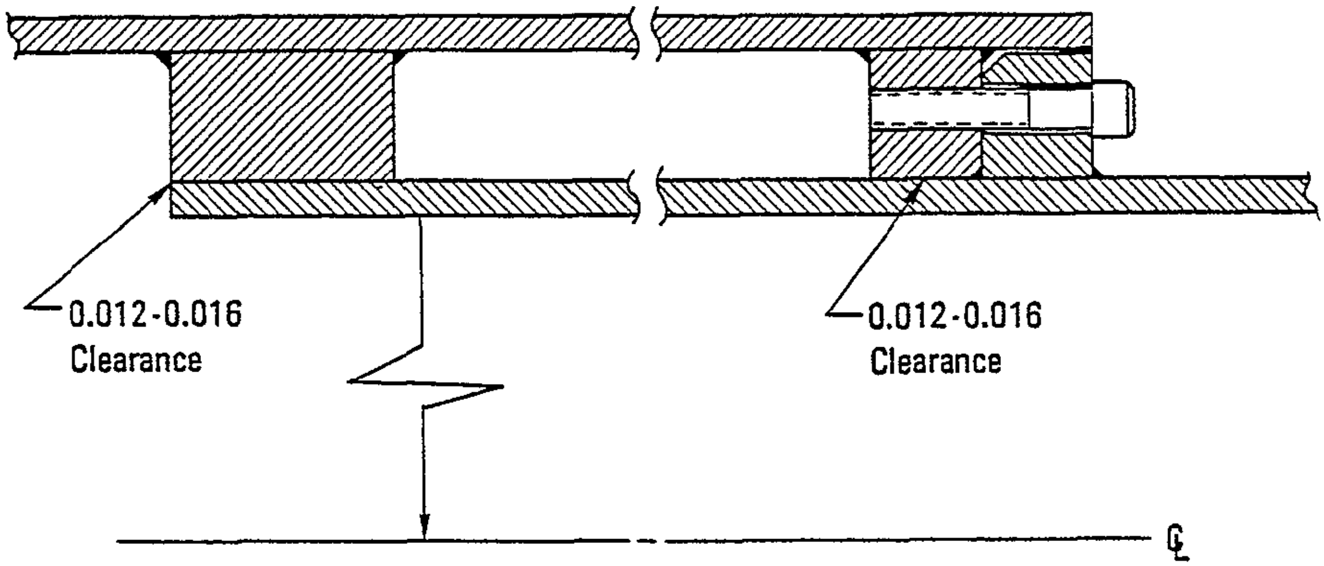
1. Program on schedule thru 2/17/77
2. Winding complete through 3 of 4 turn around disc locations
3. Pins in mandrel shaft joints fractured late 2/18/77 while winding to final turn around disc
4. Four of five joints beefed up with added pins – 2/22/77
5. One layer wound full length – 2/24/77 – bolts in fifth joint fractured and mandrel twisted about 585 in station
6. Cured partially completed spar – 2/28/77
7. Mandrel removed – 3/3/77
8. Soar inspection – 3/5/77



Demonstration Spar Mandrel Shaft



Mandrel Shaft Joints 2-5



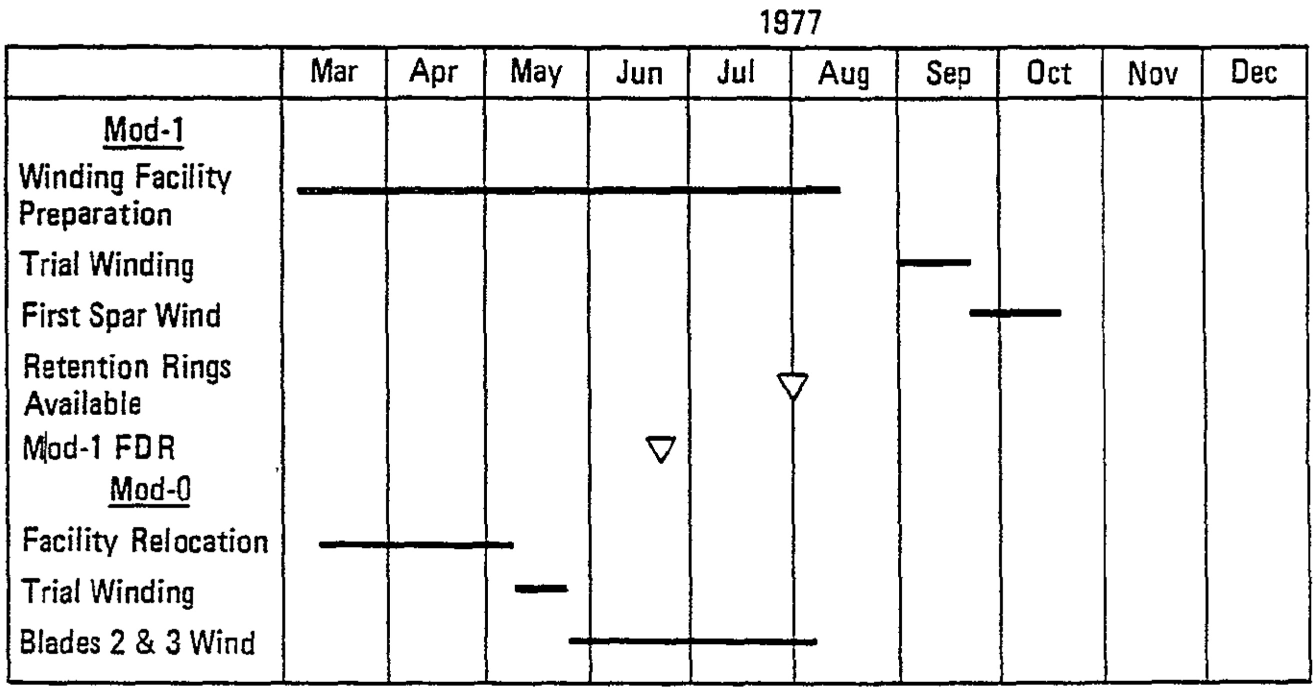
Mandrel Shaft Joint 1

Mod-1 Mandrel Design

Actions to preclude demo spar mandrel problems

- Review tool concepts
- Tool stress analysis
- Tool design review
- Tool hardware review
- Mandrel inspection after trial winding

Current thinking is to reduce loads on tooling by removing spar mandrel prior to shell winding



Mod-0/Mod-1 Blade Schedules

Spar Inspection/Testing

- Tap test
- Bright light inspection
- Weight
- Dimensional inspection

Demo Spar/Mod-1 Spar Comparisons

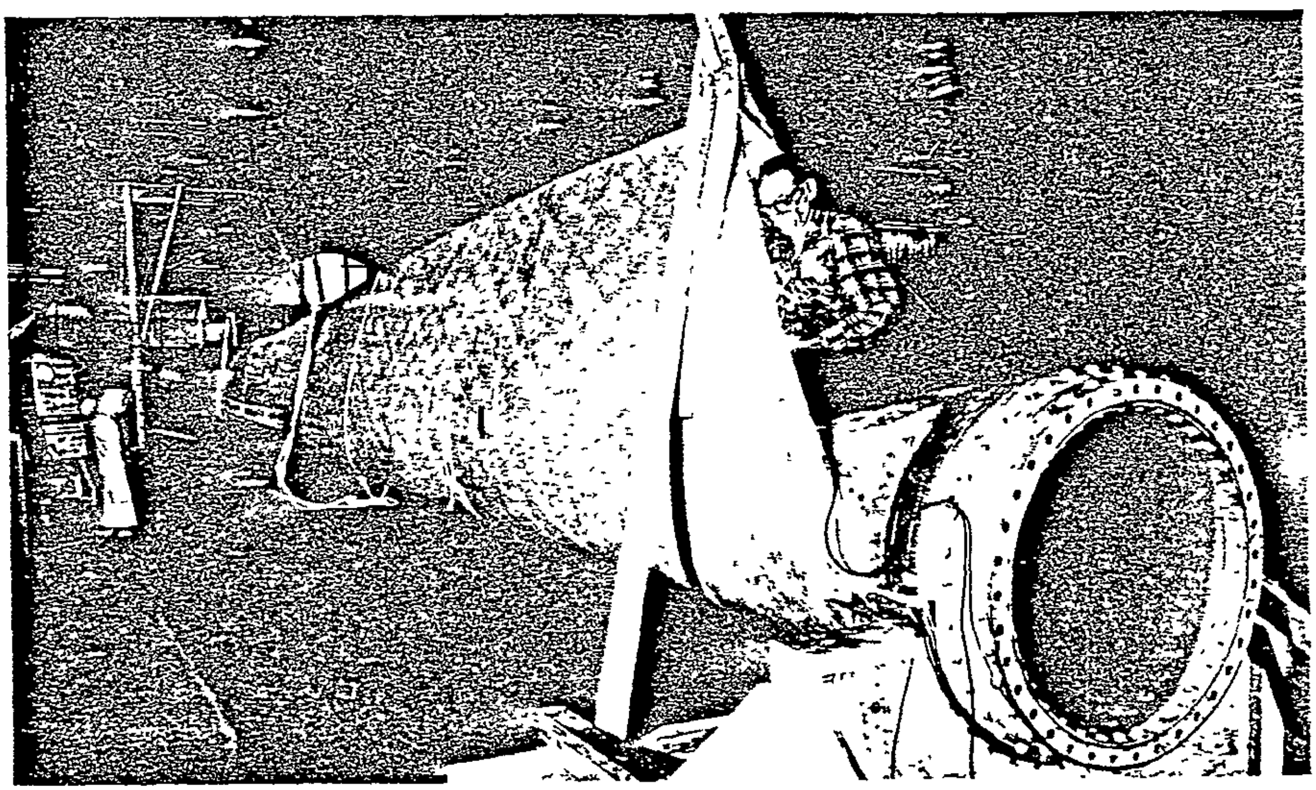
	Demo Spar	Mod-1 Spar
Material	. . . . Same	. . . .
Length	98.5 Ft.	100.7 Ft.
Winding angle	± 30°	± 30°
Max section	67 in. x 58 in.	69 in. x 56 in.
Weight	7800 lb.	9300 lb.

Mod 1 Demo Spar Weight

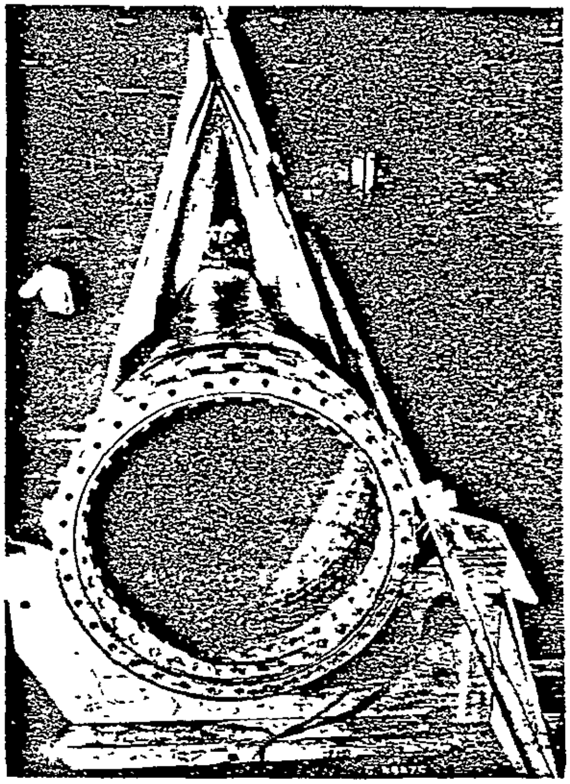
Estimated thru 17th layer -	6030 #
Estimated wt. add'l layers -	1720 #
Total estimated	7750 #
Total design wt. @ completion (± 10% wt. tol.)	7800 ± 10%

GENERAL INSPECTION TEST COVERAGE

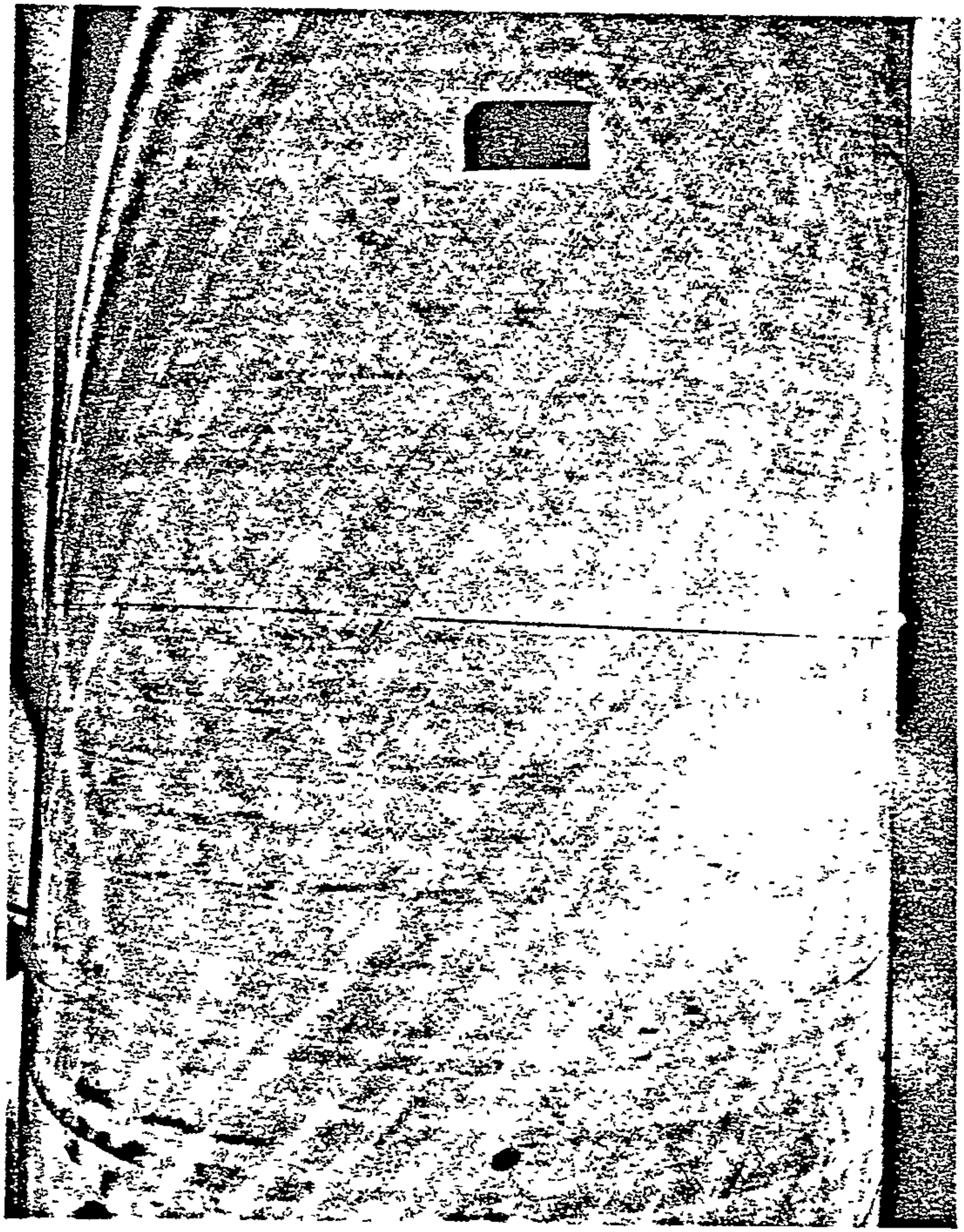
- Visual
  - During winding
  - Final overall
- Tap test
  - Spar
  - Shell
- Bright light
  - Entire completed blade
- X-ray/fluoroscope
  - Recommended for recording of metal ring area



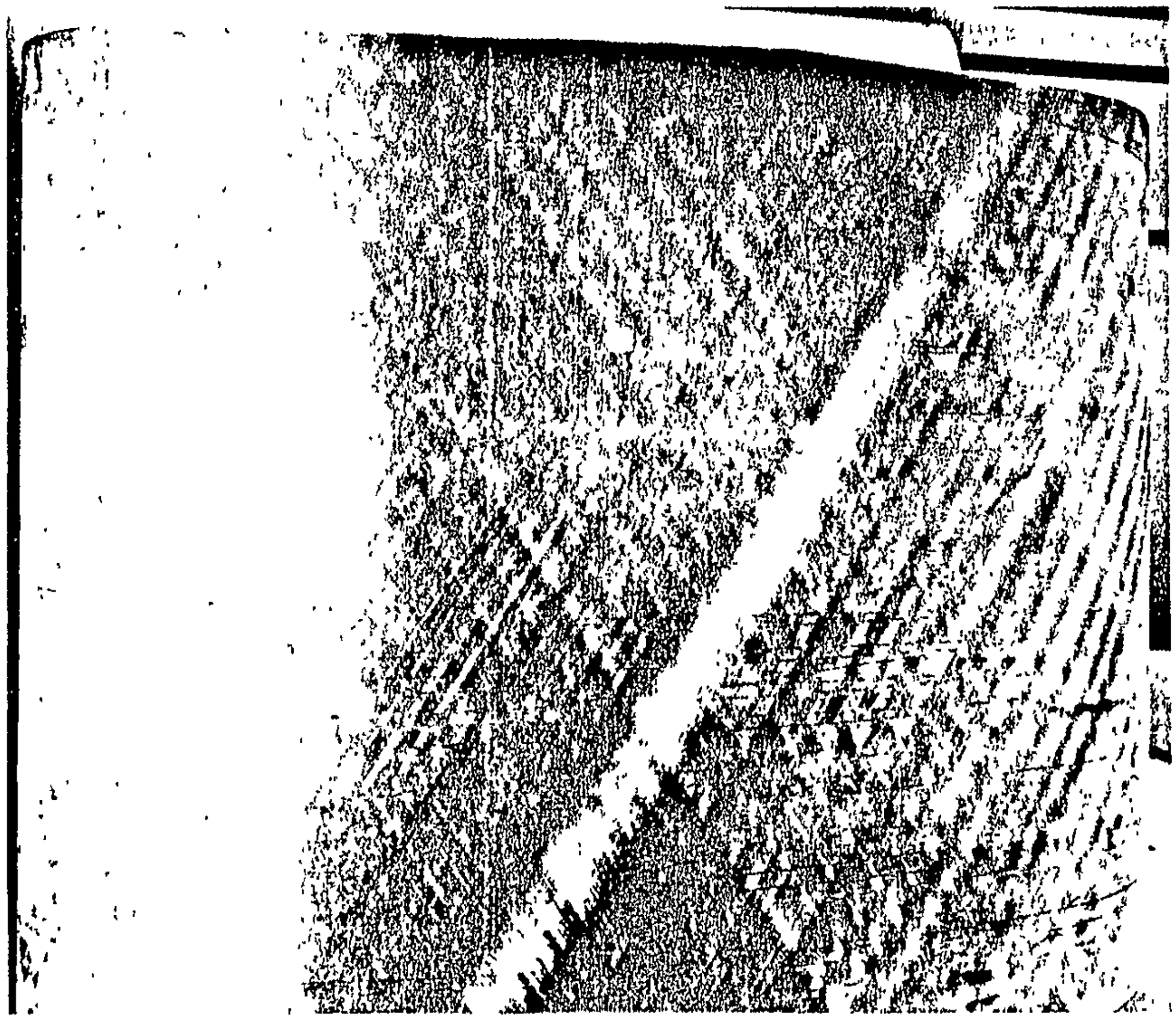
Bright Light Inspection



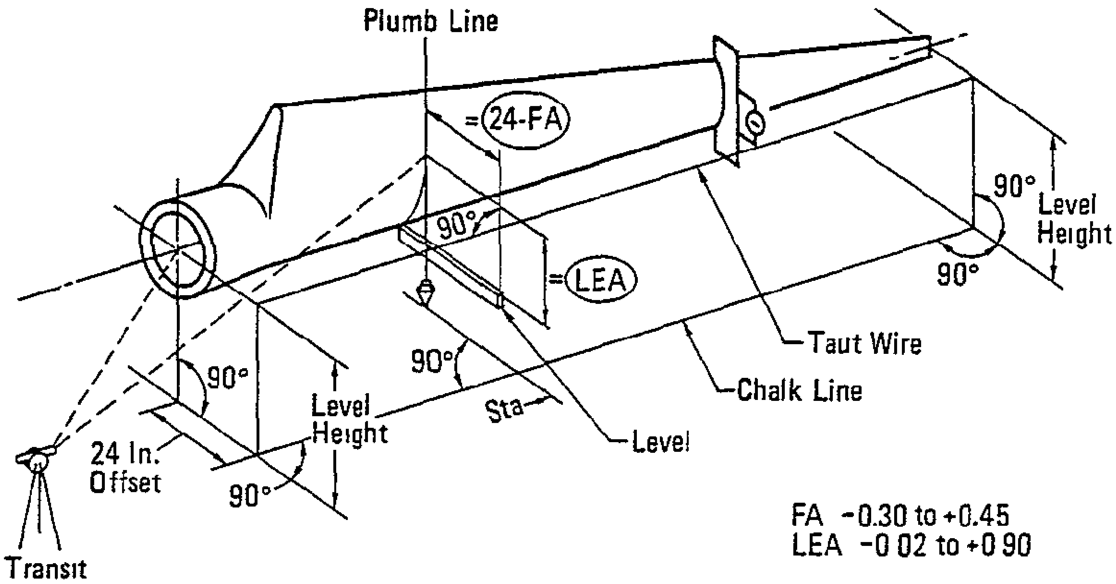
Blade Internal Inspection



Satisfactory Laminate Bright Light Test



Unsatisfactory Bright Light Test



Wind Turbine Blade Dimensional Inspection

Mod-1 First Article Inspection

Width

Blade thickness

Contour

Twist angle

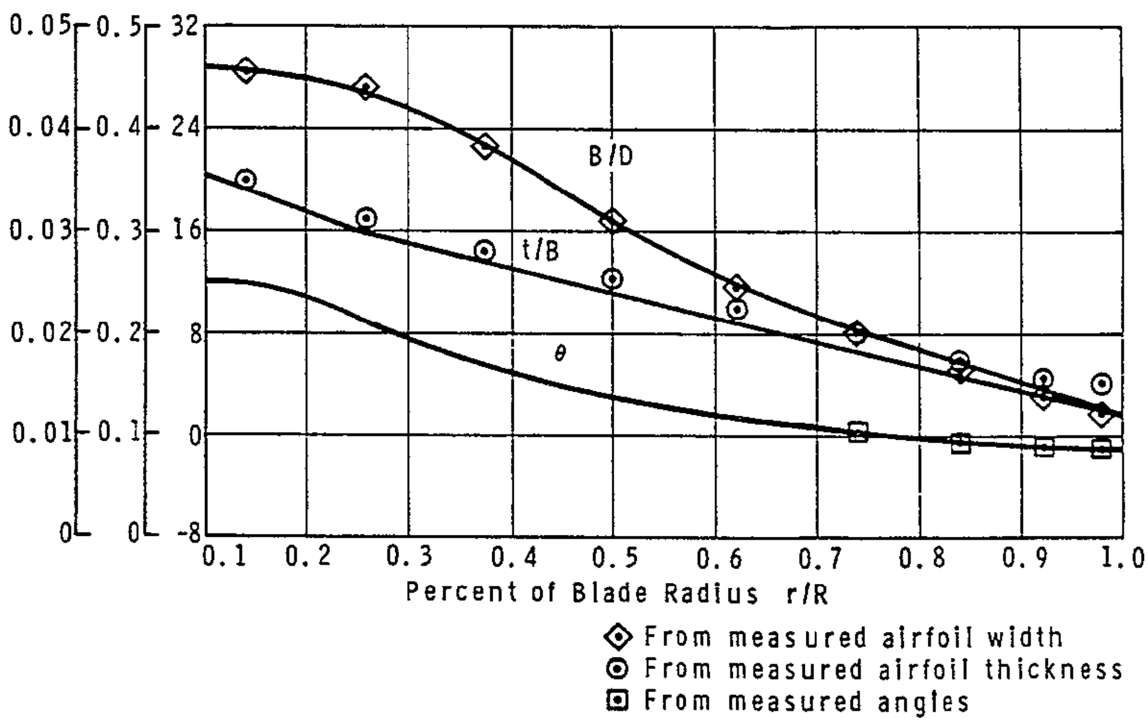
Shell thickness

Leading edge alignment

Face alignment

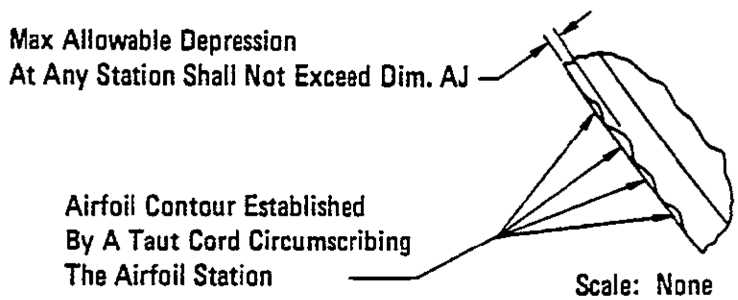
- Level blade between retention center and tip airfoil
- Establish parallel 24 in. offset taut line and chalkline
- Rotate blade from keyway to make station vertical
- Mark station with plumb line
- Project centerline on blade
- Measure dimension from taut line to blade surface
- Subtract from 24 in. to get F.A.
- Place level on blade leading edge
- Measure along plumb line from level to centerline projection to get L.E.A.

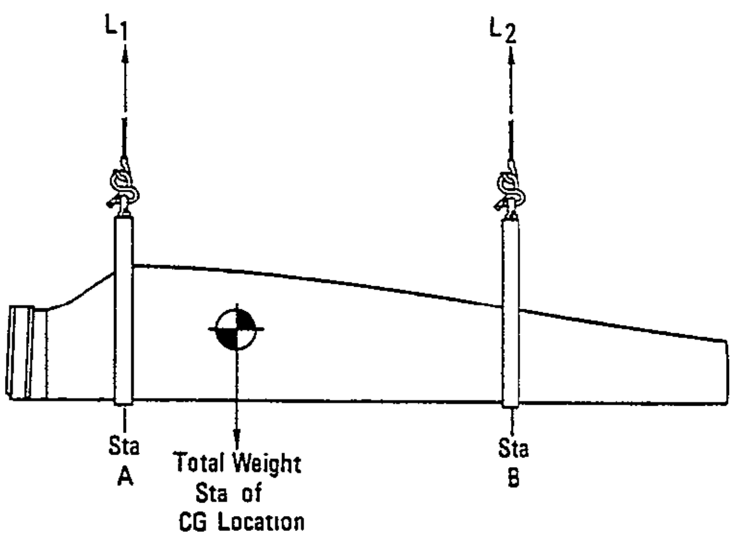
MANUFACTURING PROTOTYPE BLADE MEASURED AERODYNAMIC SHAPE



Contour Inspection

STA	AJ
75.00	~
105.00	0.053
150.00	0.051
195.00	0.049
240.00	0.046
285.00	0.042
330.00	0.039
375.00	0.037
420.00	0.036
465.00	0.035
510.00	0.033
555.00	0.031
585.00	0.030
630.00	0.028
690.00	0.027
735.00	0.026

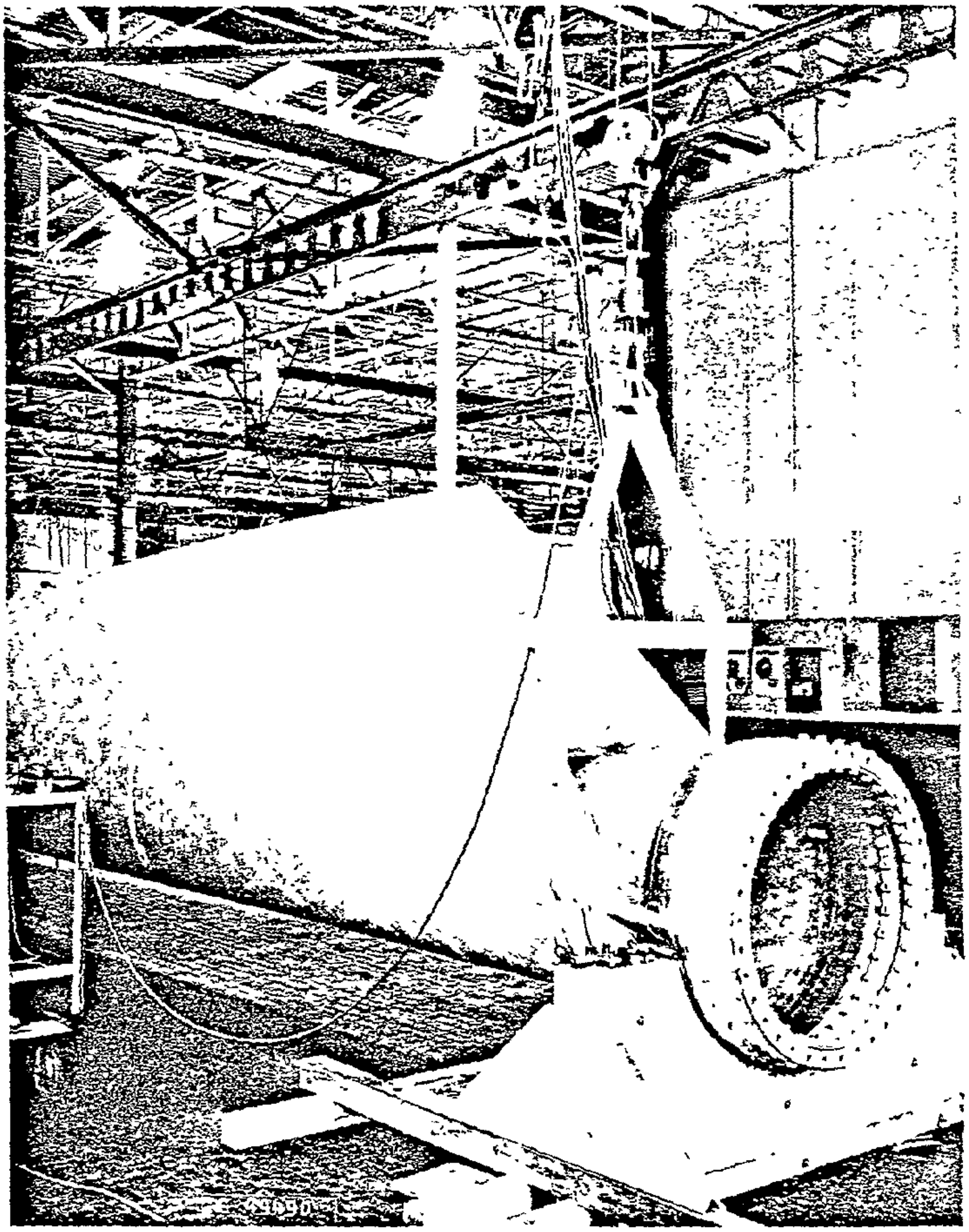




2412 Lbs Total Weight = $(L_1 - Tare_1) + (L_2 - Tare_2)$

$$\text{CG Location} = \frac{(L_2 - Tare_2)\quad (B - A)}{\text{Total Weight}}$$

Weight and Moment Determination



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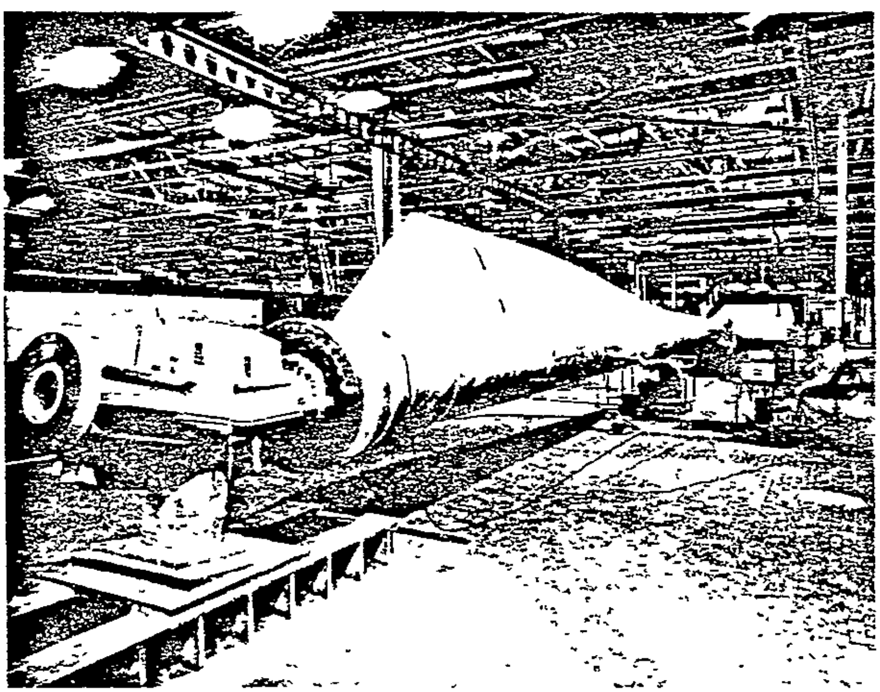
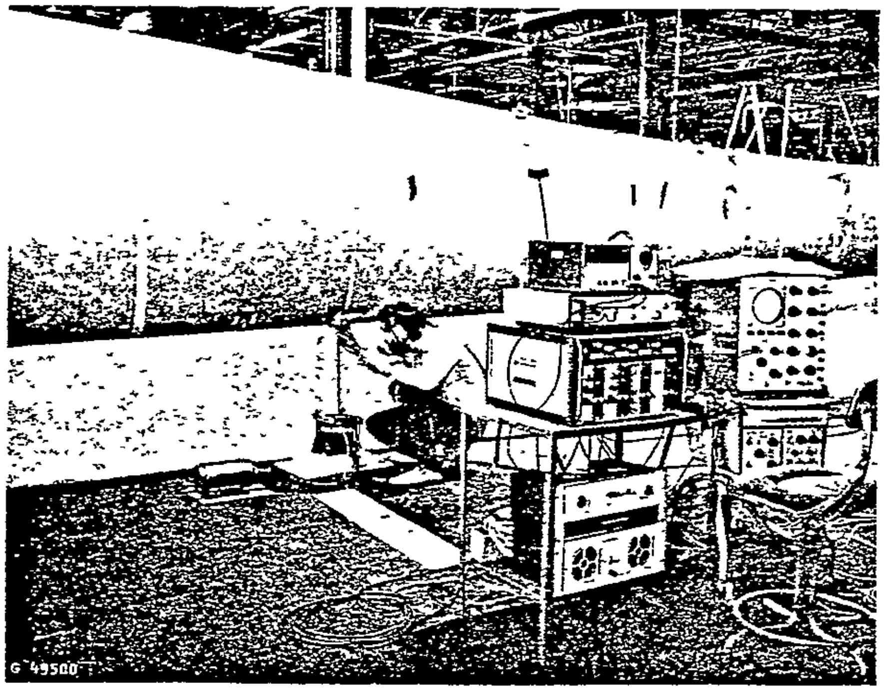
Blade Weight Measurement

Experimental Modal Analysis

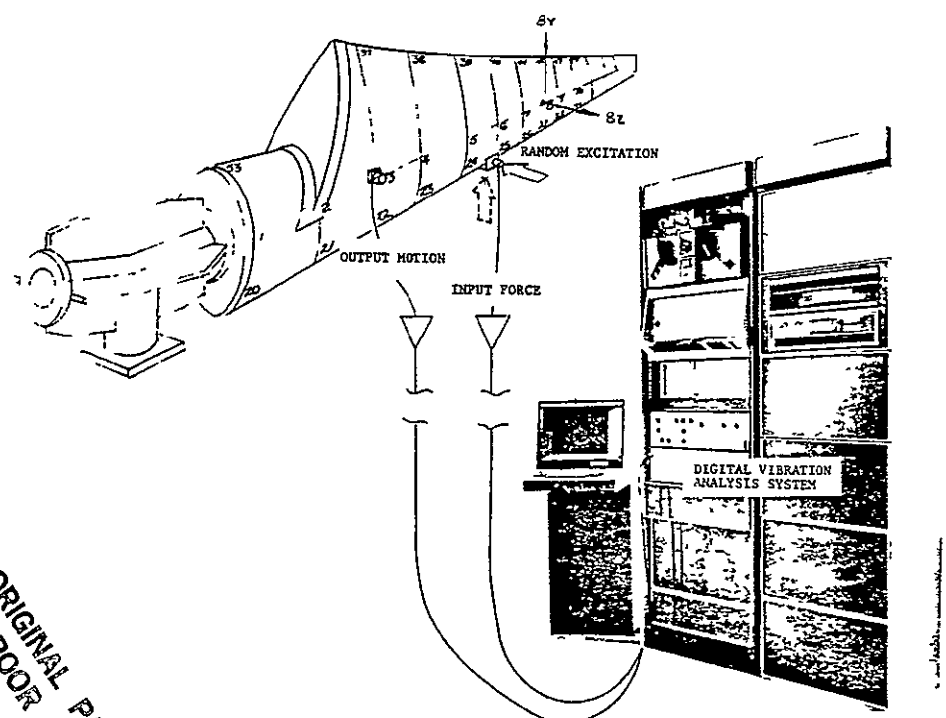
- Determines blade resonant frequencies and mode shapes
- Performed with blade cantilever mounted T.E. up
- Blade randomly excited with a shaker
- Accelerometer responses recorded at intervals along blade
- Responses computer plotted for frequency peaks
- Displacement mode shapes computer synthesized and plotted
- Test successfully performed on mfg. prototype blade



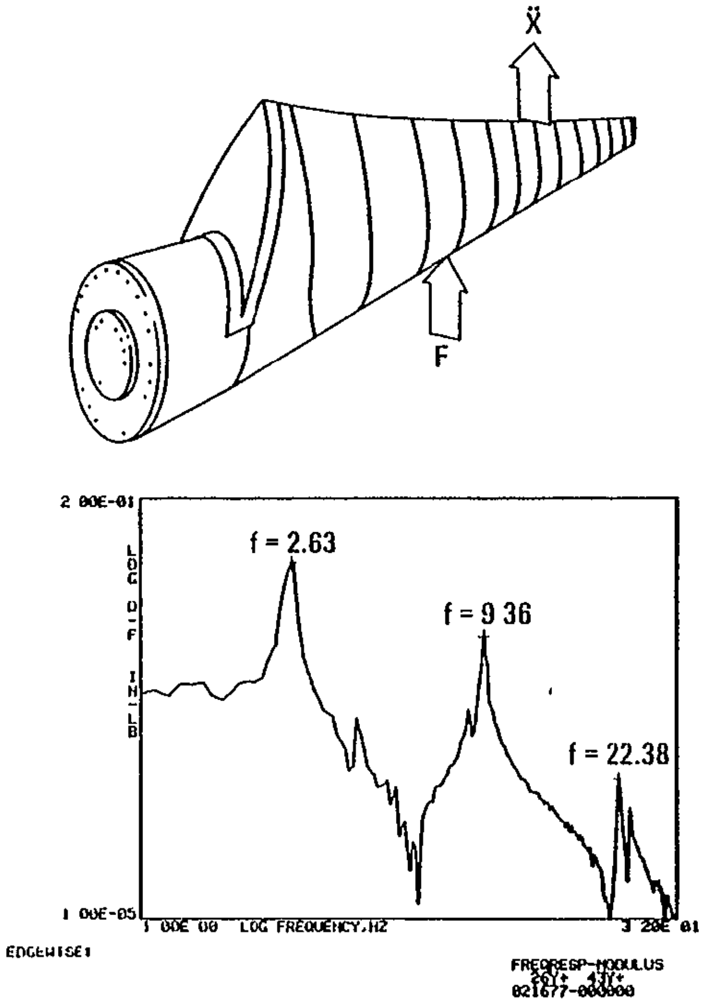
Test Arrangement



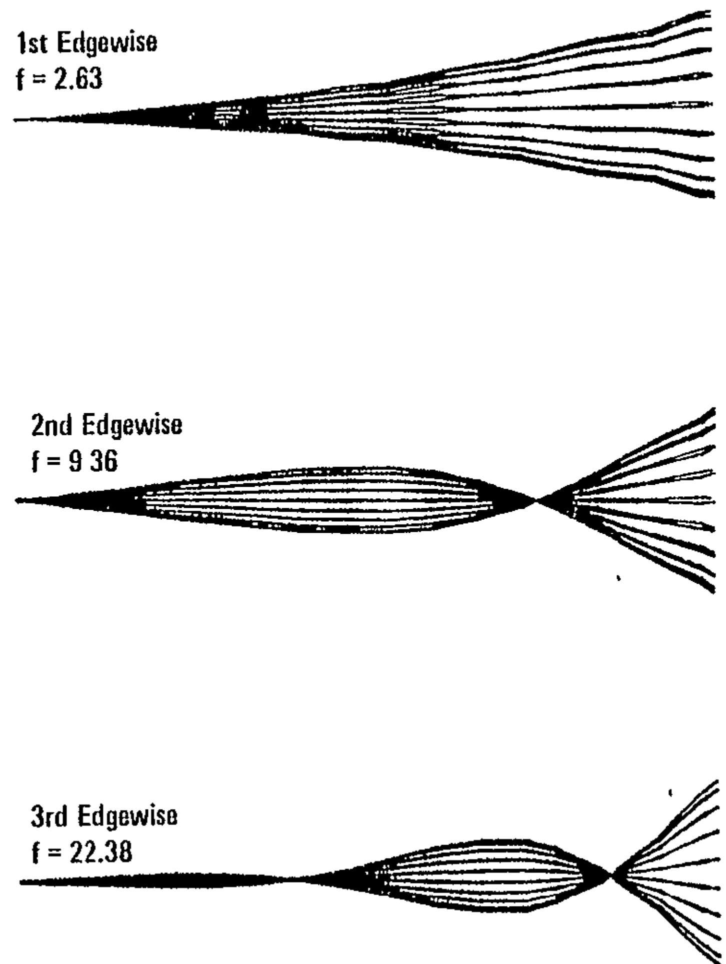
EMA Shaker Installation



Test Arrangement

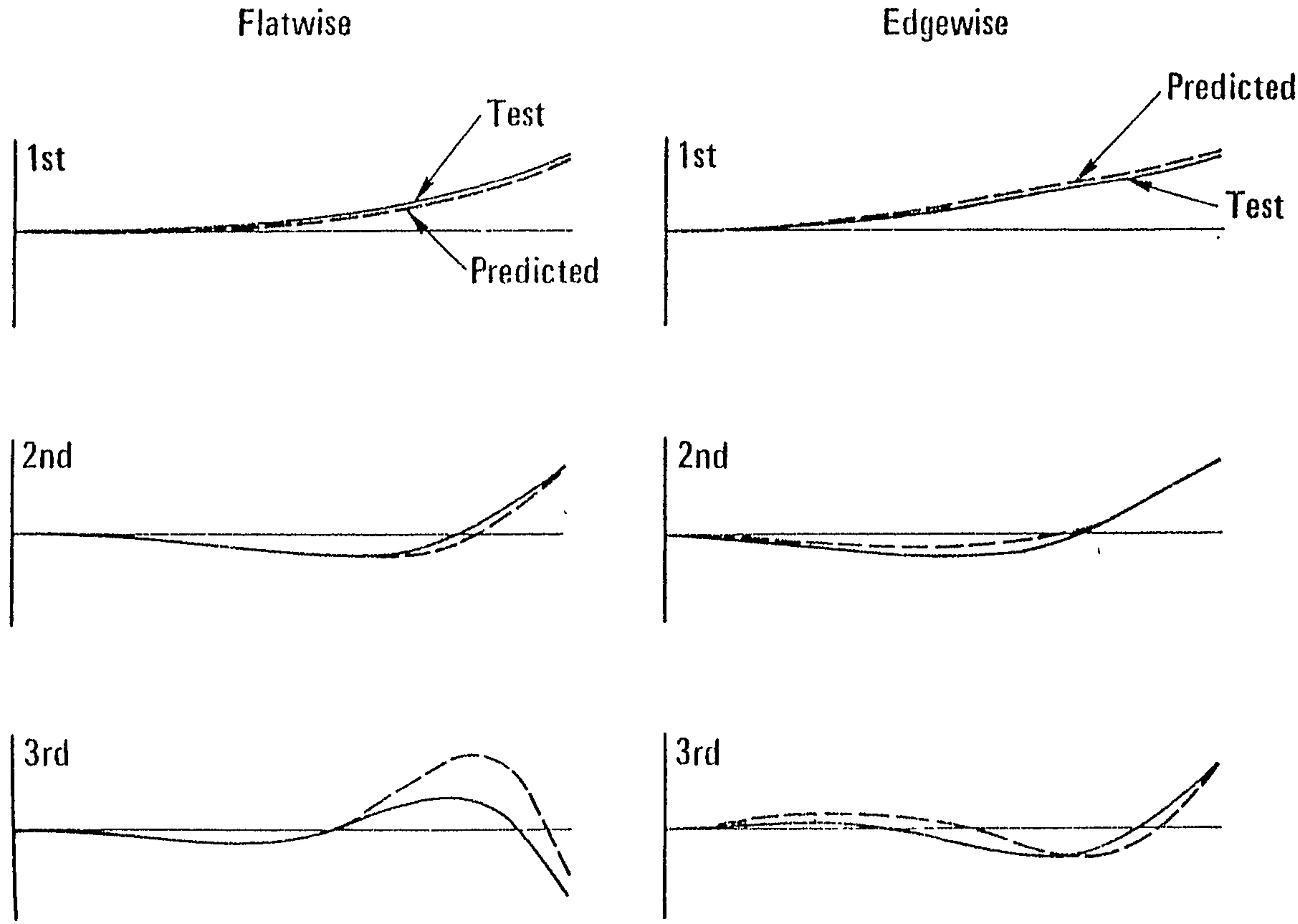


Typical Edgewise Transfer Function



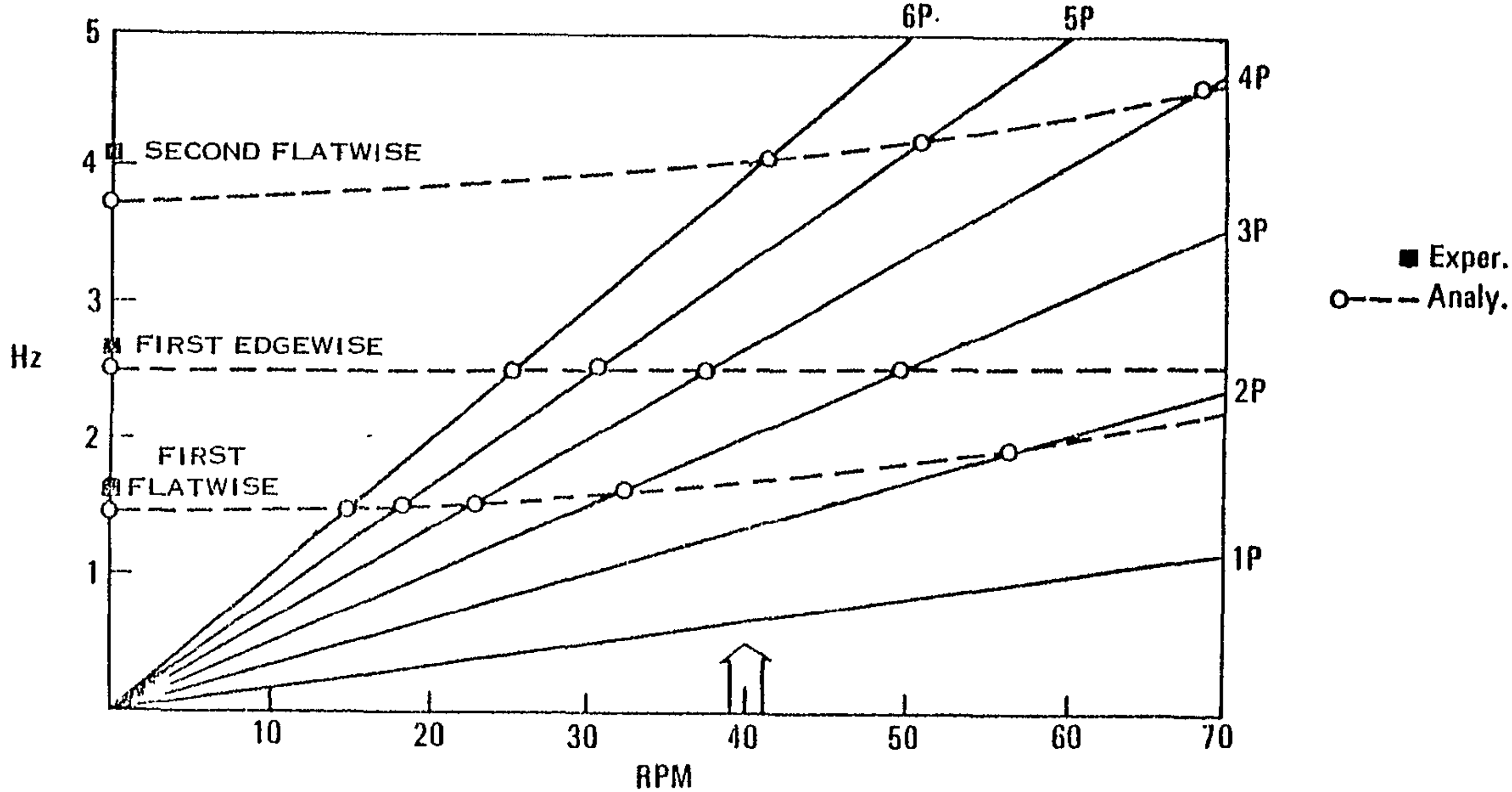
Typical Experimental Mode Shapes

MANUFACTURING PROTOTYPE – MODE SHAPES



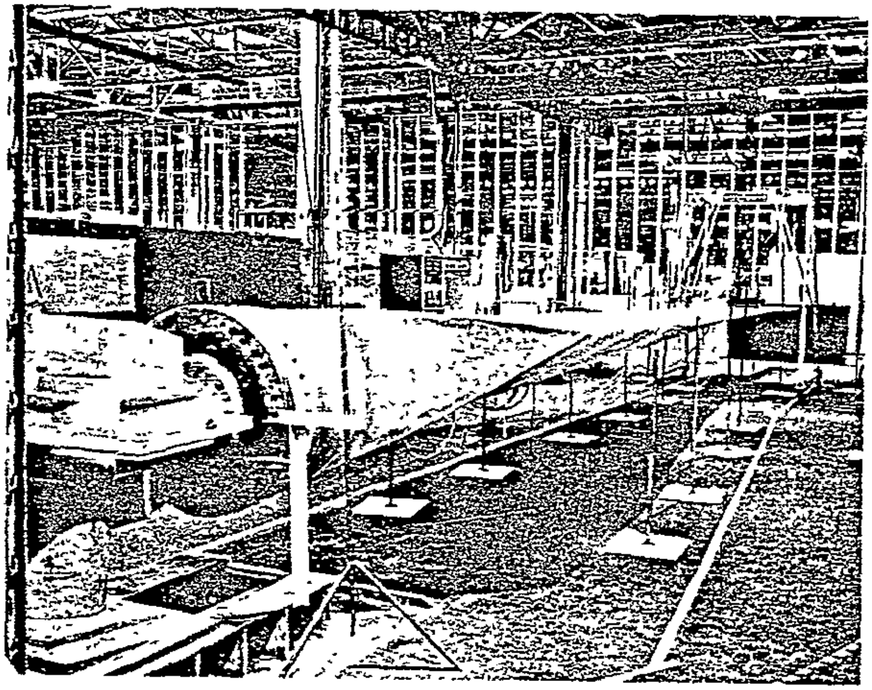
COMPARISON WITH ANALYTICAL MODEL
MANUFACTURING PROTOTYPE IN TEST MOUNT

Mode	Frequency – Hz Analytical	Experimental
1st Flatwise	1.45	1.53
1st Edgewise	2.51	2.63
2nd Flatwise	3.73	4.05
3rd Flatwise	8 08	8.59
2nd Edgewise	9.86	9.36
3rd Edgewise	24.75	22.38
Static Torsional	28.59	29.26

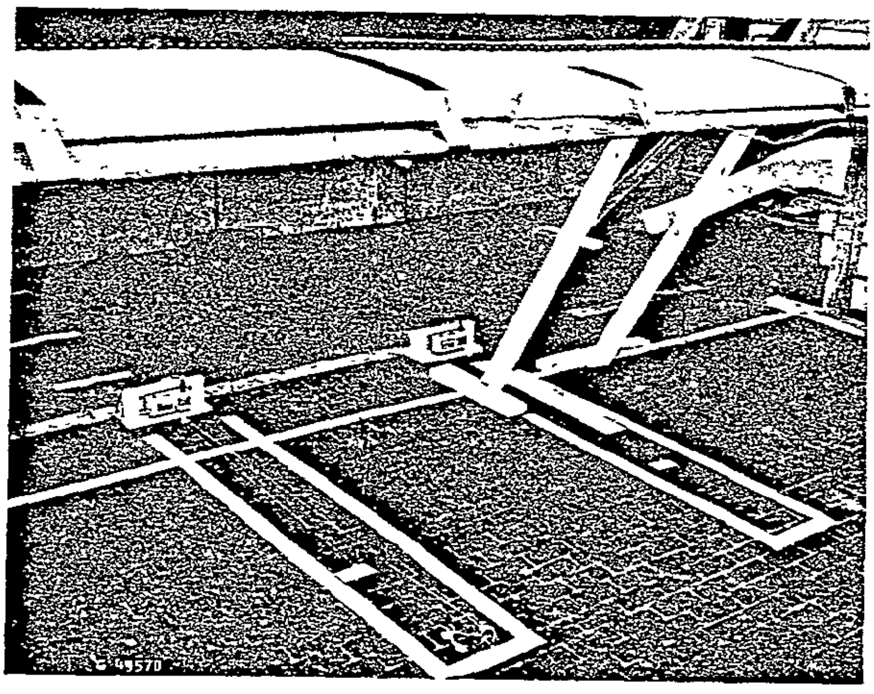


BLADE INSTRUMENTATION CALIBRATION

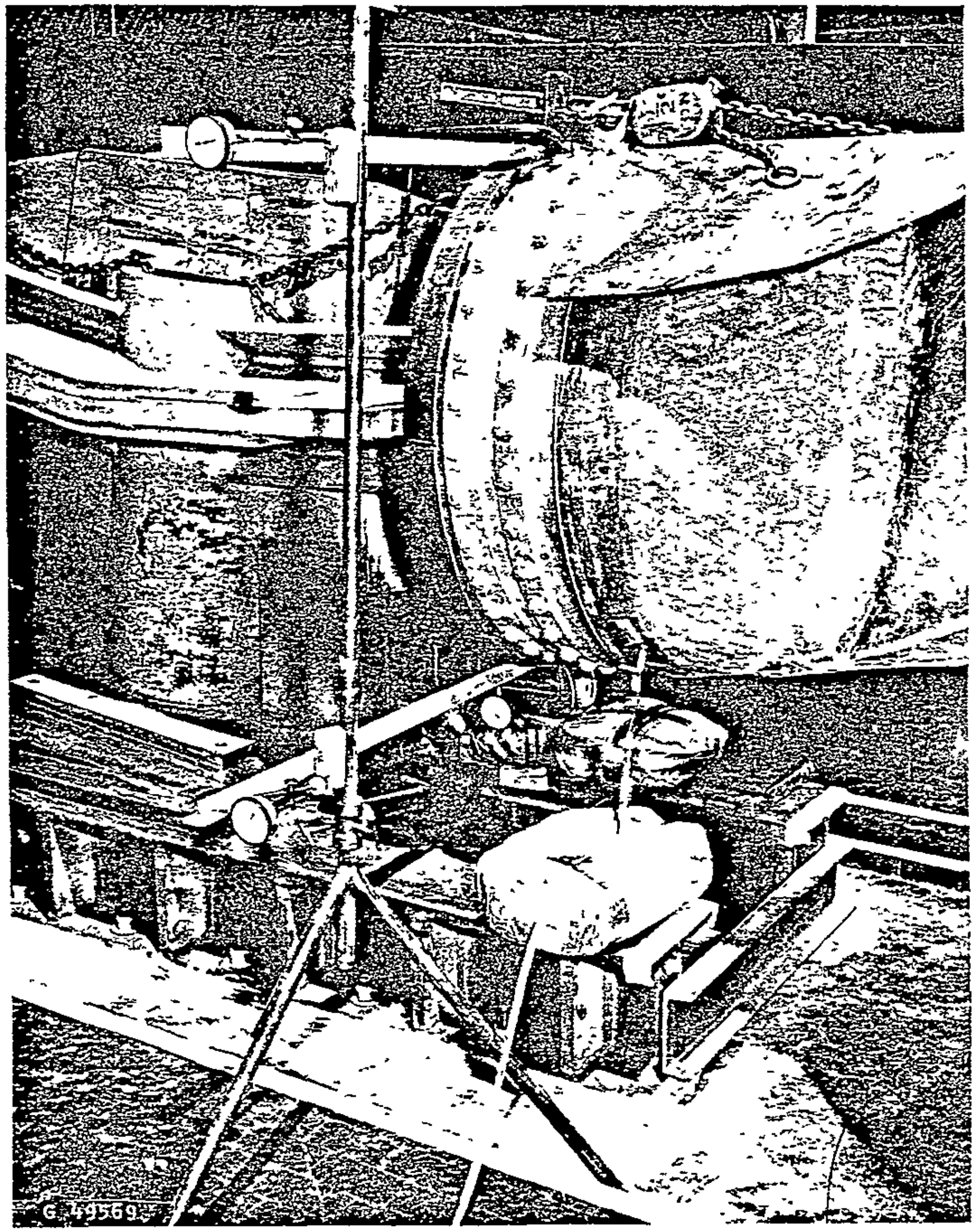
- Relates strain gage outputs to applied loads
- Performed with blade cantilever mounted
- Blade deflected with concentrated load
- Similar ESA testing successfully performed on manufacturing prototype Mod-0 blade
- Proof load testing successfully performed on manufacturing prototype Mod-0 blade



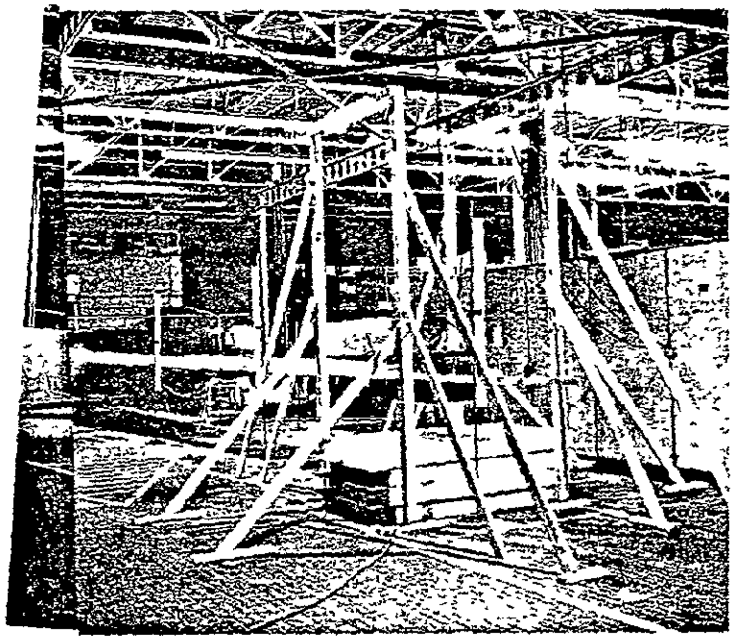
ESA Test Setup



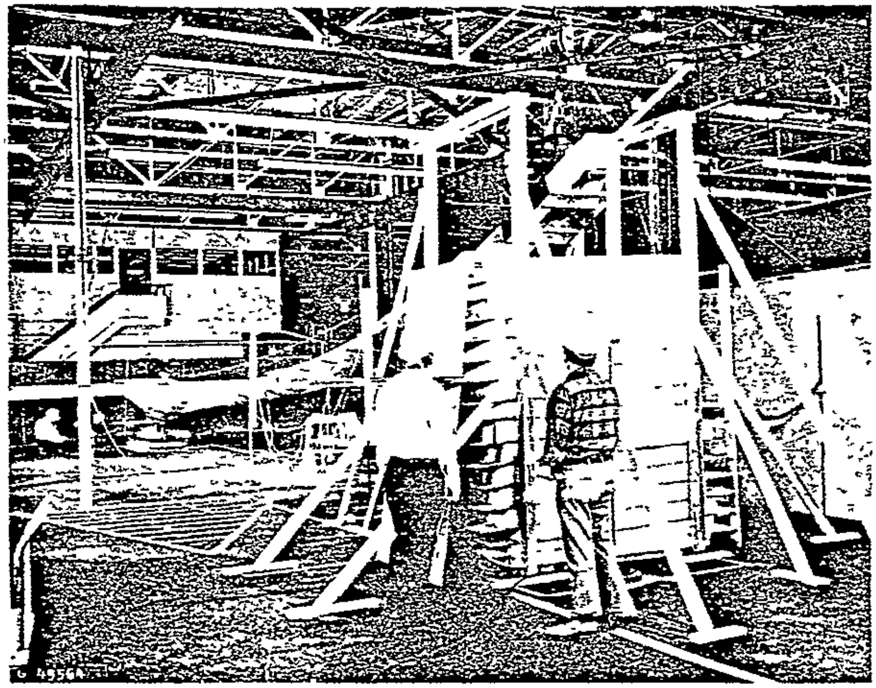
Deflection Measurements



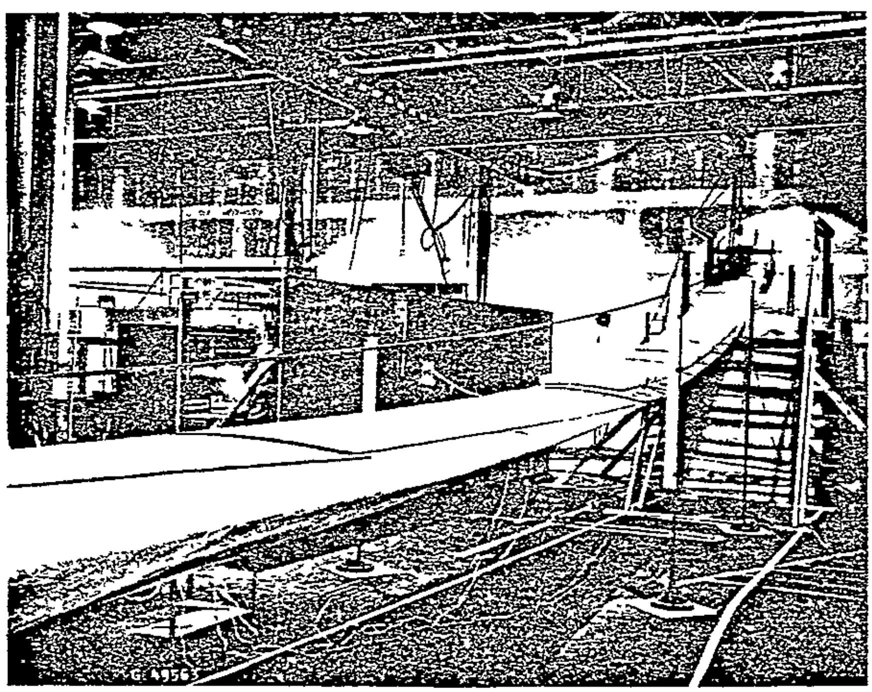
Hub Deflection



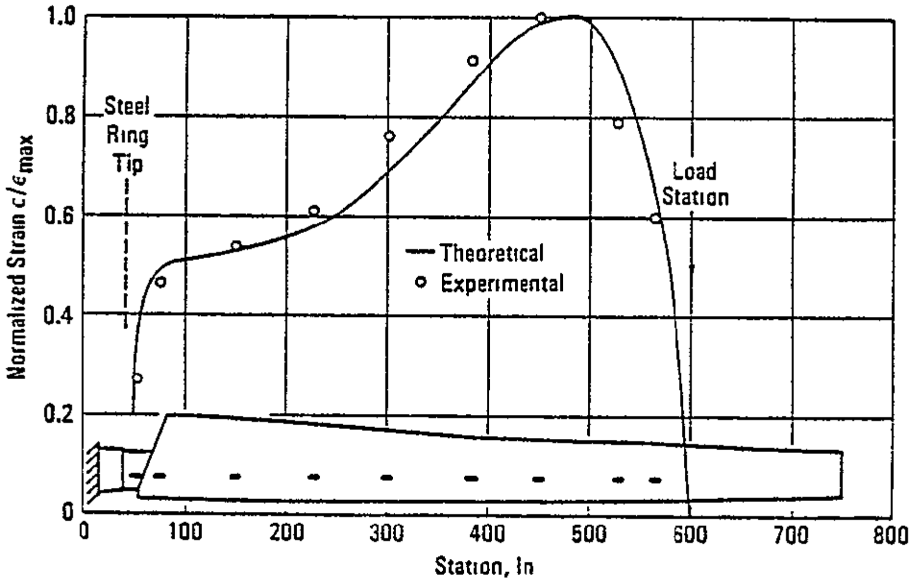
Blade Static Load Test Setup



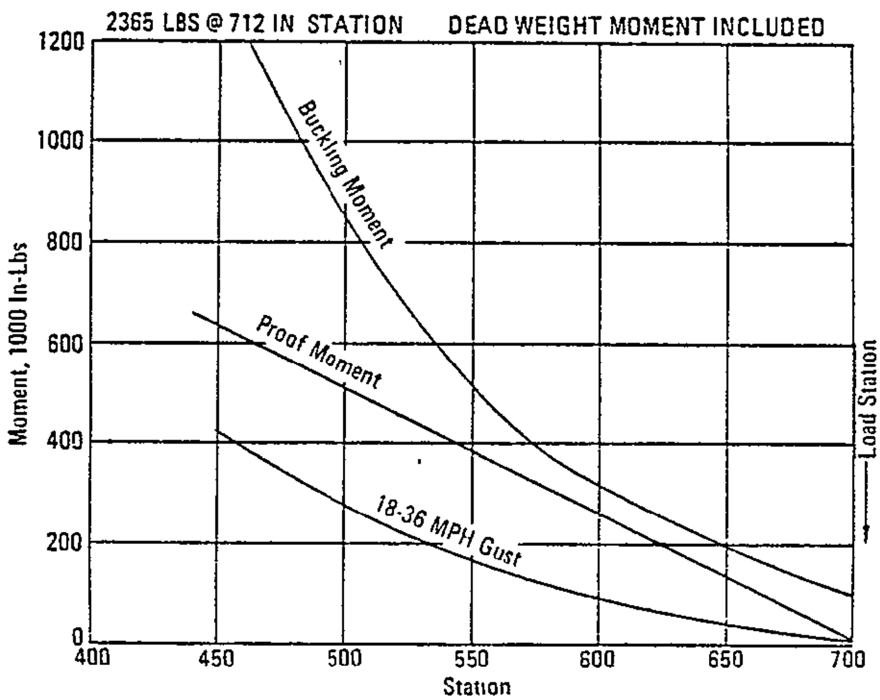
Blade Under Static Load



Blade Test



Mod-0 Manufacturing Prototype Longitudinal
Strain Distribution Load at 600-inch Station



Mod-0 Manufacturing Prototype Blade Proof Moment Plot

Mod-1 Blade Winding Machine Operations Projection

Mod-0 experience

Manufacturing prototype required 200
Actual machine hours

X.67 Factor for development problems during winding

X.5 Factor for production 2 to 1 speed increase in system

67 Machine hours net production estimate

Mod-1 projection

402 machine hours net production

equals 67 x 7.5 $\frac{\text{Mod-1 FRP}}{\text{Mod-0 FRP}}$ factor

x $\frac{4}{5}$ factor for size effect on per pound cost estimates

Recommendations

- The PDR Mod-1 blade configuration be approved for detail design
- Approval be given for ordering long lead time hardware
- Additional testing be considered to obtain more experience with blade
- Evaluate the effect of certain changes being considered

Long Lead Hardware

Approval to procure the following long lead time hardware is requested

- Blade retention bearings
- Blade retention ring material
- Blade tooling

Additional Testing - Mod-0

Install & run composite blades on Plum Brook machine to:

- Obtain early load/stress verification
- Obtain early confirmation of blade dynamic structural action
- Early confirmation of blade integrity under operating conditions
- Evaluate environmental effects

Mod-1 blade is a large scale version of Mod-0 blade

Additional Testing - Mod-1

Fabricate spare blade for structural testing

ESA Test (Experimental stress analysis)

- Determine stress distribution in retention and selected airfoil areas
- Determine blade axis
- Apply concentrated static loads
- Temporary local hard points

Fatigue Test

- Confirm that blade strength & design limits are compatible

EMA Test (Experimental modal analysis)

- Conduct before and after Fatigue Test
- Use to determine that blade structure not effected by fatigue loading

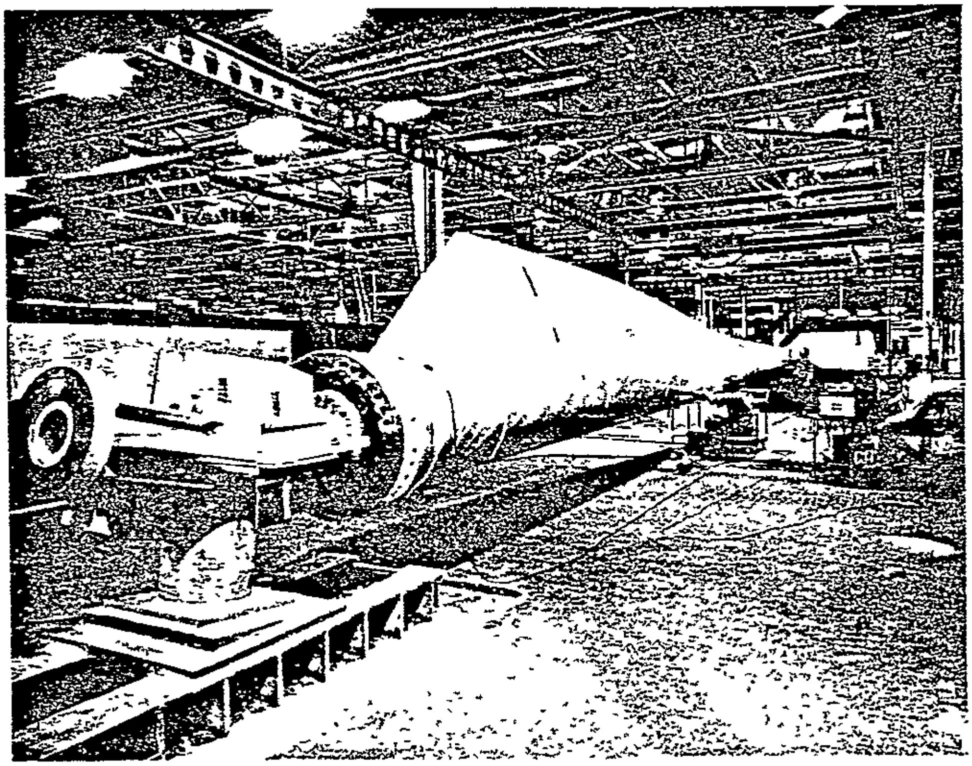
Buckling Test

- Verify satisfactory buckling capacity

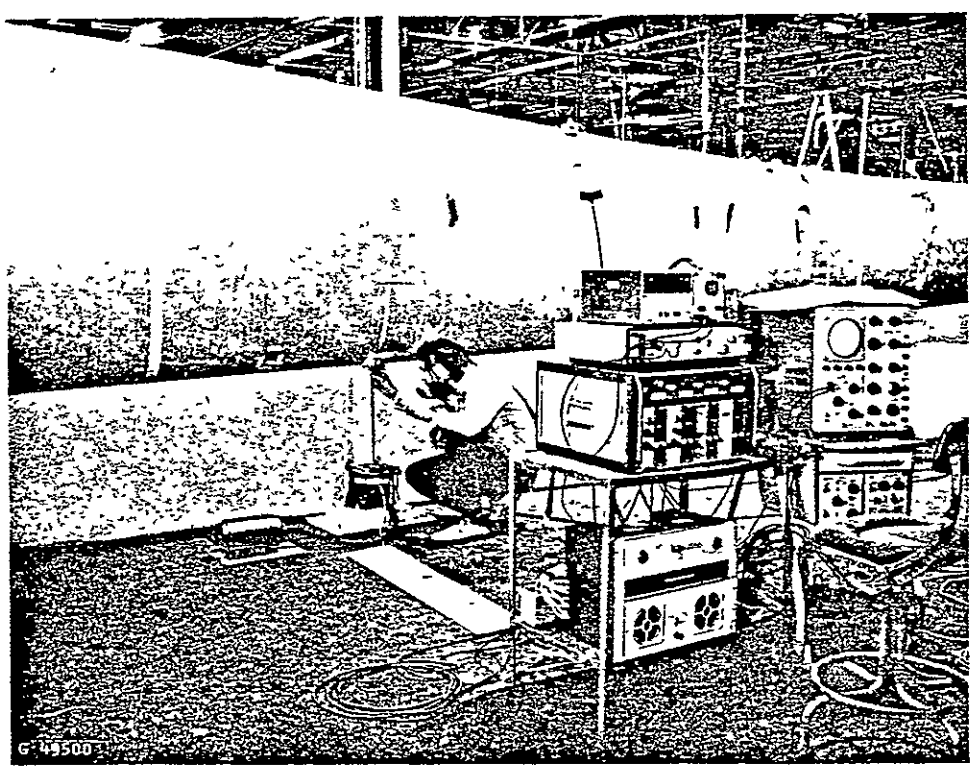
Retention Proof Test

Changes Being Considered

- Incorporate design factors from code verification
- Modify conditions of load case 2
- System criteria affecting pitch change stiffness/ blade stability



EMA Test Setup



EMA Test

APPENDIX A

QUESTIONS AND ANSWERS

1. A plot of C_L vs. r/R at design conditions where:

   C_L - lift coefficient
   r - local blade radius
   R - blade tip radius

2. A plot of B C_L/R vs. r/R showing the actual design values compared to the ideal values at the design condition where:

   B - local blade chord

3. A summary of the aerodynamic design conditions analyzed. Are these conditions sufficient to design the WTG or are there more critical conditions that should also be considered?

4. Why does the peak blade stress occur at the 65% position of the blade radius? Why isn't the blade designed for a uniform stress with a view towards reducing blade weight?

5. To what degree does the shape of the lift coefficient vs. angle of attack curve near the stall point affect aerodynamic design, weight, or blade stress? Does a "flat top" lift curve have any benefits in off-design performance? Have blunt-base airfoils been considered for use near the hub region of the rotor to reduce the local drag coefficient and increase the local lift coefficient? (refer. Hoerner-Fluid-Dynamic Drag, pg. 3-20 to 3-22).

Additional Aerodynamic Information on Mod-1

In response to Mr. Puthoff's request for additonal aerodynamic information, the following is presented.

1. The lift coefficient, C_L, distribution along the blade for the rated velocity at -1° blade angle at 3/4 radius is presented in figure 1. C_L is plotted versus r/R where

   r - local blade radius
   R - blade tip radius

The lift distribution is based on uniform flow. It should be recognized that this curve varies azimuthally due to wind shear and tower shadow.

2. The bC_L/R distribution corresponding to the C_L distribution is compared to the optimum bC_L/R distribution in Figure A-2.

b - local blade chorde

The drop-off from the optimum in the shank region is due to these sections having narrower chords than those specified for the optimum because of practical considerations. This deviation from the optimum results in some degradation in performance.

3. A parametric study spanning blade activity factor, planform shape, twist distribution, diameter and rpm was made to select the minimum diameter with maximum yearly power out.

4. The blade is proportioned to meet all of the seven load cases. As shown in Figure A-3, the peak total compressive stress is around 0.75 radius for gusting and high wind velocity conditions. The rated condition and the down-gust condition produce relatively flat stress distributions. In addition, the critical speed locations influence the stiffness and mass distribution which, in turn, affect the stress distribution for a given applied moment.

5. The NACA 230XX airfoil family was selected from both aerodynamic and manufacturing considerations and is an excellent family for the application of relatively thick airfoil sections characteristic of wind turbines. Figure A-4 shows the airfoil aerodynamic characteristics of a NACA 23015 section, typical of the thickness at 85% radius. Lift and drag coefficients versus angle of attack are shown for smooth airfoils and NACA roughness, both based on test data, and NASA roughness which is based on trends define for other airfoil sections. The surface of the Mod-1 blades is more closely representec by the NASA roughness criteria. Although the stall characteristics are rather abrupt, i should be noted that the blade does not operate near stall during normal operation.

   During gusts, the blades do operate in the stall region. A study of the relative torsional stabilities of the NACA 230XX and an airfoil with a "flat top" C_L distribution was made for the Mod-0 blades using the Steinman stall flutter analysis. It was found that the airfoils with the "flat top" C_L are slightly more stable in forward gusts while the NACA 230XX airfoils are somewhat more stable in reverse gusts. The differences were small and it was concluded that neither has a clear advantage over the other with respect to flutter.

   Airfoils generally greater than 25% thick can derive some benefit by blunting the trailing edge. A limited amount of data (Figure A-4) shows that although minimum drag is increased, the maximum lift coefficient is increased with a corresponding delay in drag rise. Thus it would appear that for the inner 40% of the Mod-1 blades, a potential performance benefit could be obtained by blunting the trailing edge. Available data have indicated that below 25% thickness such blunting of the trailing edge would result in performance losses.

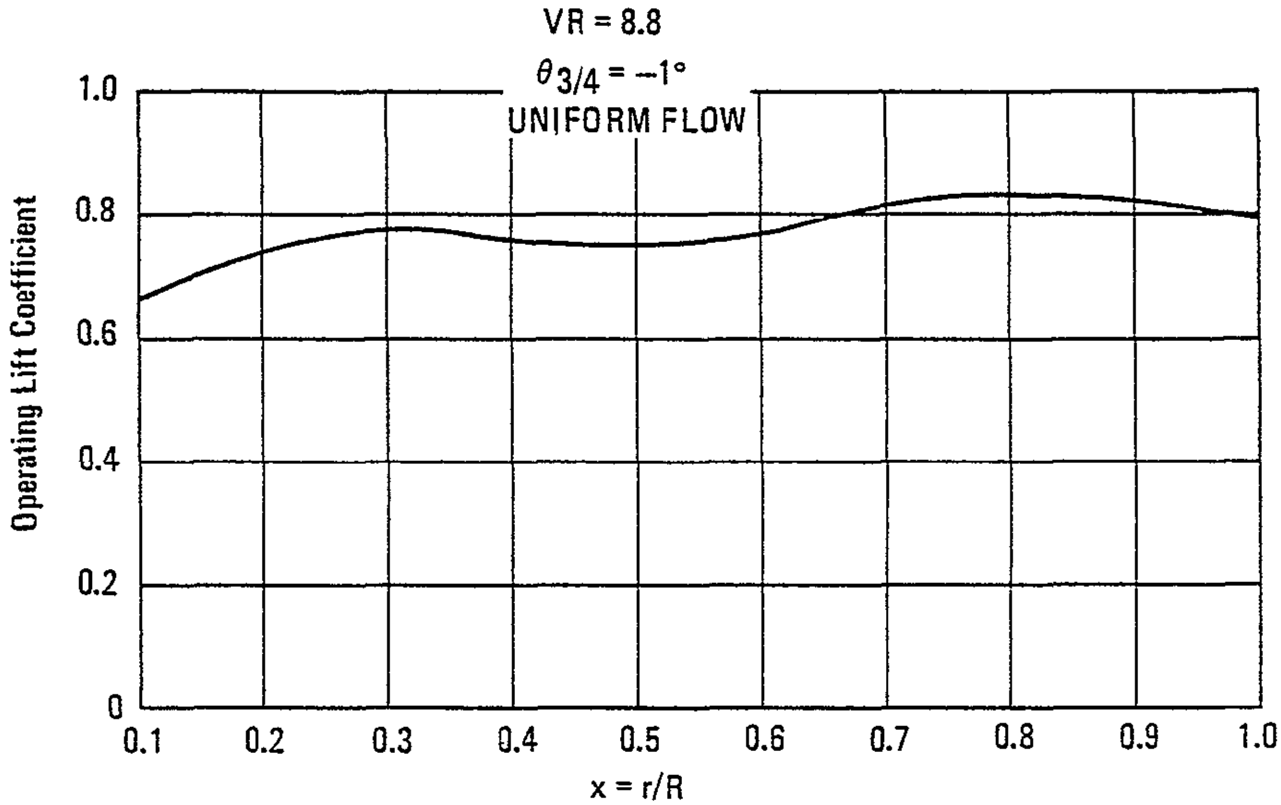


Figure A-1. Mod-1 Lift Coefficient Distribution at Rated Velocity

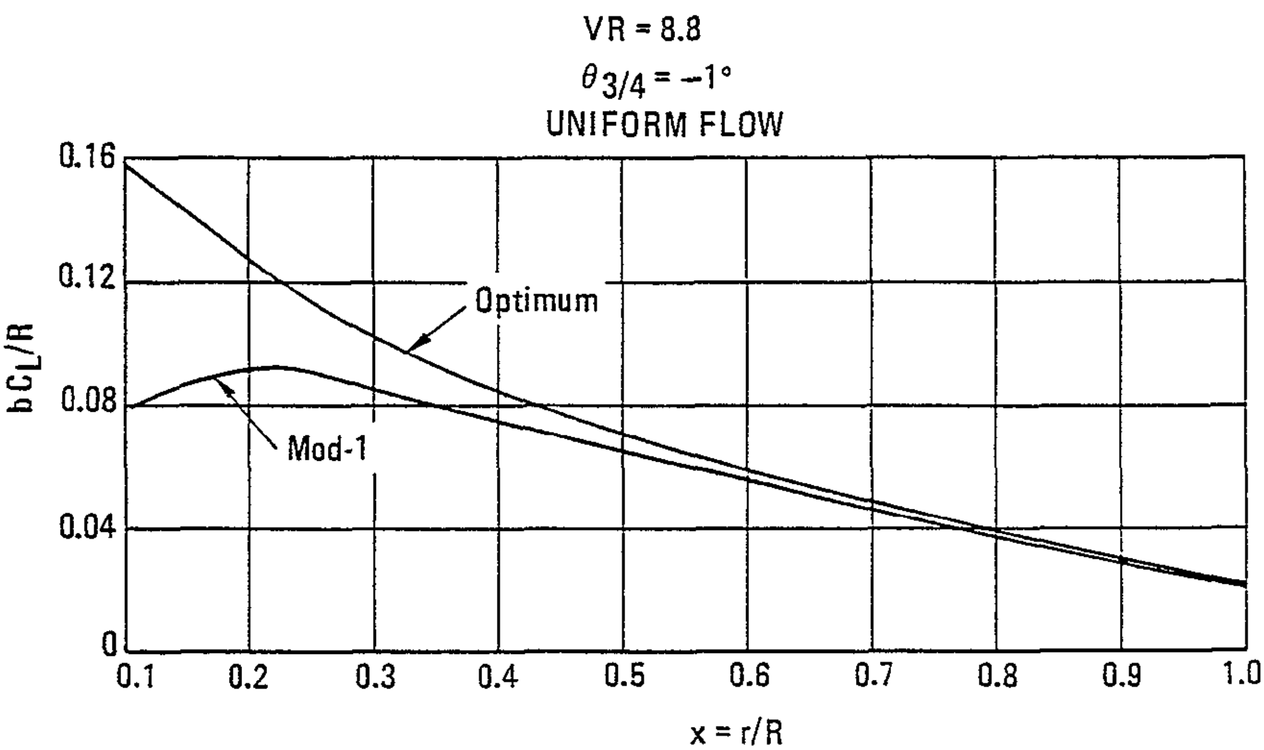


Figure A-2. Mod-1 Loading Distribution

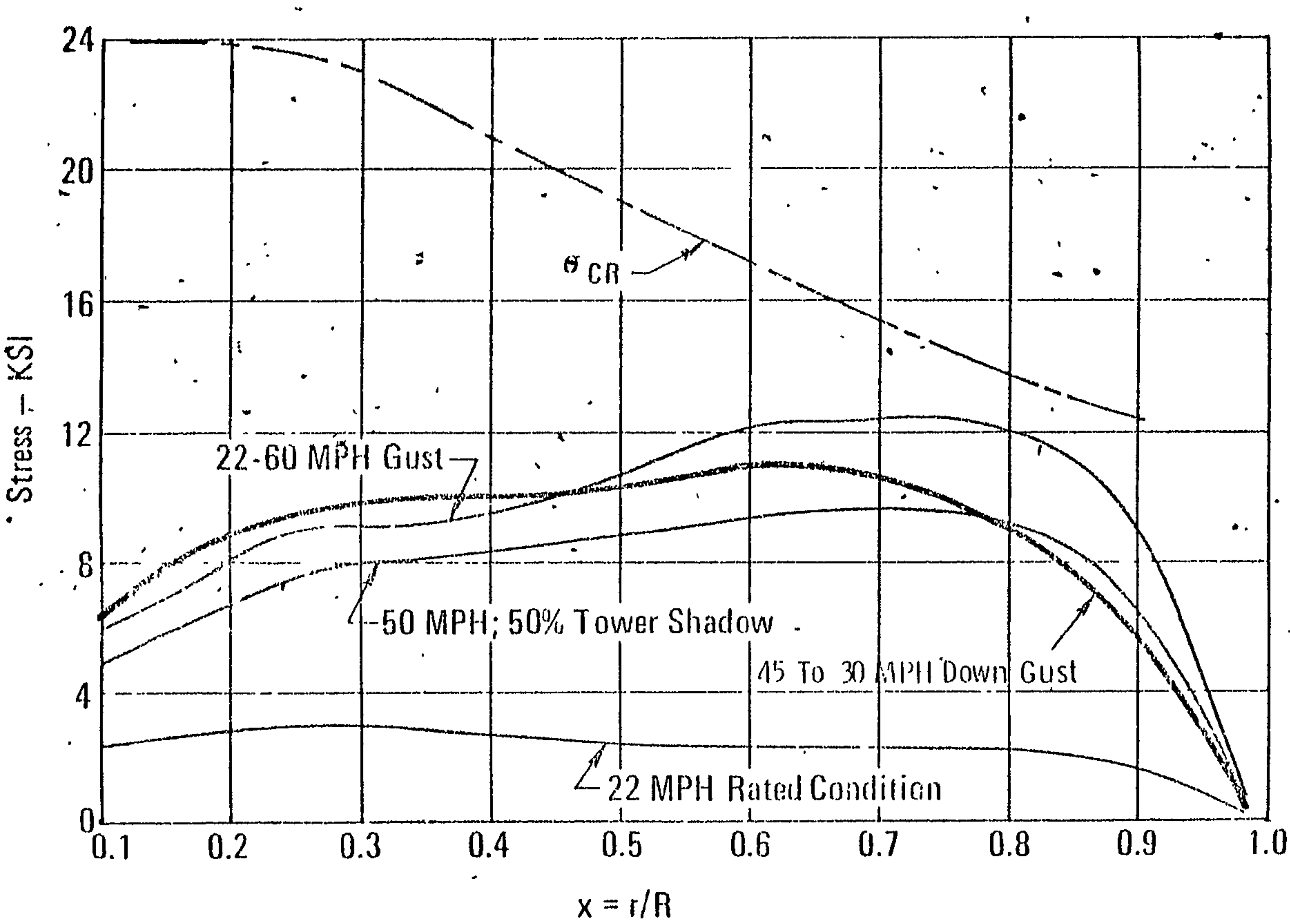


Figure A-3. Buckling Capacity

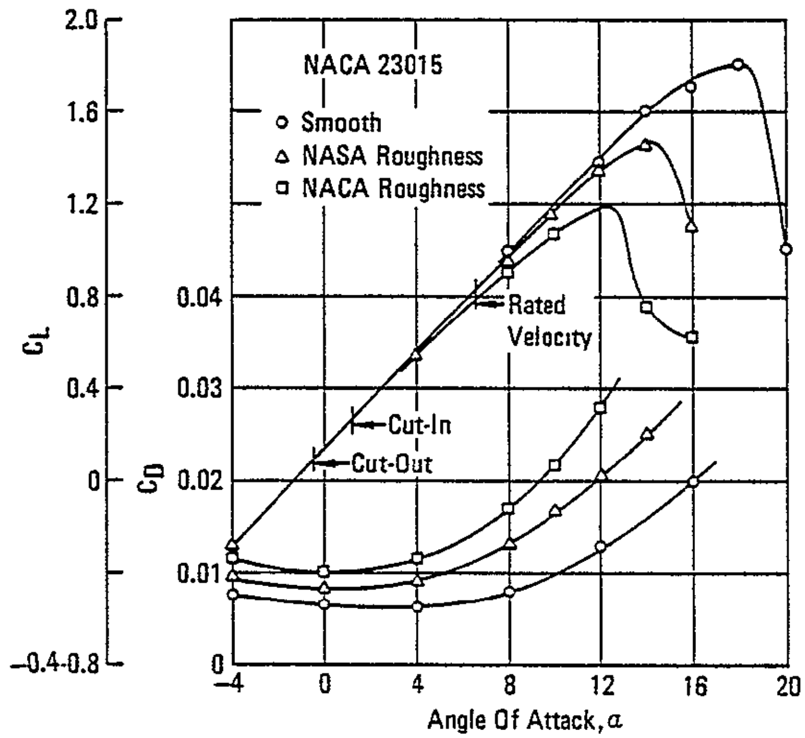


Figure A-4. Airfoil Aerodynamic Characteristics

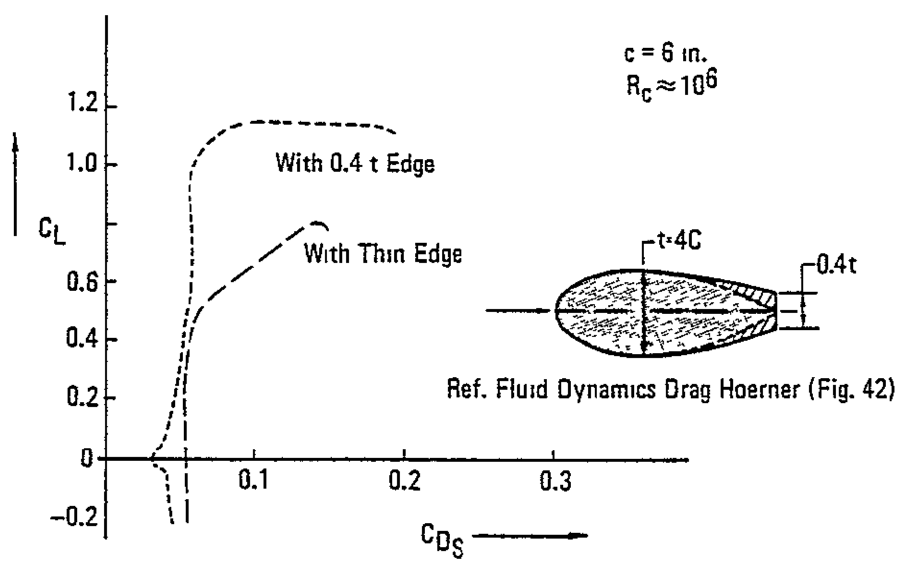


Figure A-5. Lift and Drag of a 40% Thick Airfoil Having Square Ends

APPENDIX F

SYSTEM TRADE STUDY

Abstract

This appendix is a report on the results of a parametric system trade study, entitled "Mod 1 Parametric Trade Study - Final Report", dated March 30, 1979. The report consists of copies of slides that were used in a presentation at NASA - Lewis Research Center, with some additional explanatory test.

MOD - 1 PARAMETRIC TRADE
STUDY
FINAL REPORT

DRAFT - MARCH 30, 1978

FOR

NASA - LEWIS RESEARCH CENTER

WIND ENERGY PROJECT OFFICE

GENERAL ELECTRIC CO.

SPACE DIVISION

ADVANCED ENERGY PROGRAMS

TABLE OF CONTENTS

1.0 INTRODUCTION

This study was authorized and outlined by NASA Lewis Research Center Management for the expressed purpose of documenting a compilation of tradeoff data. This data was prepared and analyzed for the purpose of developing a low weight, low cost wind turbine generator using the experience of the MOD-1 hardware design and fabrication, and the in-place personnel and analytical tools. G.E. was directed to prepare a conceptual design of a MOD-1 class utility system with a goal of achieving 400,000 lbs (total weight at the base of the tower). The system parameters outlined by NASA Lewis included the following:

- 2000 kilowatt class machines
- Utility class power
- Horizontal Axis wind turbine generator
- Less than 400,000 lb. system.

The current MOD-1 system, to be delivered and installed in 1978, weighs approximately 696,000 lbs. (weight at the base of the tower). General Electric was instructed to use any combination of factors, subsystems, components or design alternatives that could achieve the results of a less than 400,000 lb. system. Risks and areas for possible future development in the event that state-of-the-art technology cannot be used, should be identified for future potential research and engineering development tasks. The duration of the study was three months, formal kick-off was October 12, 1977.

This document represents the final report of this study.

STUDY OBJECTIVES,
APPROACH AND
RESULTS

STUDY OBJECTIVES

APPROACH

ANALYSIS FLOW

3 CANDIDATE SYSTEMS

RESULTS

1.1 STUDY OBJECTIVES

In order to establish reasonable credibility for a study of this nature, GE was instructed to use the MOD-1 operational requirements as a baseline. Equipment hardware configuration could deviate from the MOD-1 system, but its operational characteristics should be identical with any exceptions noted. GE examined deviations from the operational requirements such as hurricane conditions, low temperature conditions, etc., but determined no deviations were required. The MOD-1 operational requirements were therefore met completely.

In addition to the less than 400,000 lb. goal, GE felt it was necessary to impose cost restrictions in order to ultimately achieve a competitive system. For second unit costs, we established bogeys for each major subsystem both in weight and cost that totaled 400,000 lbs and $1000 per kilowatt and established 5¢ per kilowatt-hour as a cost of energy goal.

GE has participated significantly in many system and economic analysis studies. However, in order to achieve the best learning from MOD-1 hardware design experience, it was imperative to use the in-place design team for the tradeoff study. Similarly, existing analytical tools for the determination of dynamics and stress of systems and components, those that had been proven against MOD-0 operational data and used for the MOD-1 design, were to be utilized. Furthermore, in order to establish hardware credibility, GE utilized common MOD-1 functions and design concepts where applicable.

STUDY OBJECTIVES

	MOD-1	STUDY GOALS
WIND REGIME	18 MPH	18 MPH
RATED POWER	1818 kW	2000 kW
LIFE	30 YEARS	30 YEARS
WEIGHT	696,000 LBS.	$<$ 400,000 LBS.
2ND UNIT COST ($/kW)	$2035	$1000
COST OF ENERGY (¢/kW-HR)	11	5

KEY OBJECTIVE: ACHIEVE $<$ 400,000 POUND SYSTEM WITH MOD-1 ENERGY PERFORMANCE

1.2 APPROACH

The Analysis Flow diagram on Page 1, describes the methodology of the study approach. Using the MOD-1 as the system base-line, major weight and cost drivers were identified, ranging from dynamic loads to installation procedures. Each major subsystem including assembly and test/erection and installation were examined to determine the critical weight, cost and design drivers that forced the MOD-1 into a 696,000 pound machine. Candidate solutions that approached the weight and cost bogeys were subsequently identified for each of the major subsystems and components.

APPROACH

- IDENTIFY MAJOR COST & WEIGHT DRIVERS
  - FATIGUE & STEADY STATE LOADS
  - MASSIVE STRUCTURAL COMPONENTS
  - OVERLY COMPLEX FUNCTIONS
  - INSTALLATION PROCEDURES
- REDUCE LOADS
  - UPWIND ROTOR
  - TEETERED ROTOR
  - 3 BLADES
  - NATURAL FREQUENCY PLACEMENT
- SIMPLIFY AND/OR ELIMINATE SUBSYSTEM FUNCTIONS
  - REDUCED MOVING PARTS
  - LESS STRUCTURE
  - CONTROL SYSTEM SIMPLICITY
- SIMPLIFY FABRICATION & ERECTION TECHNIQUES

TAKE FULL ADVANTAGE OF MOD-1 LEARNING

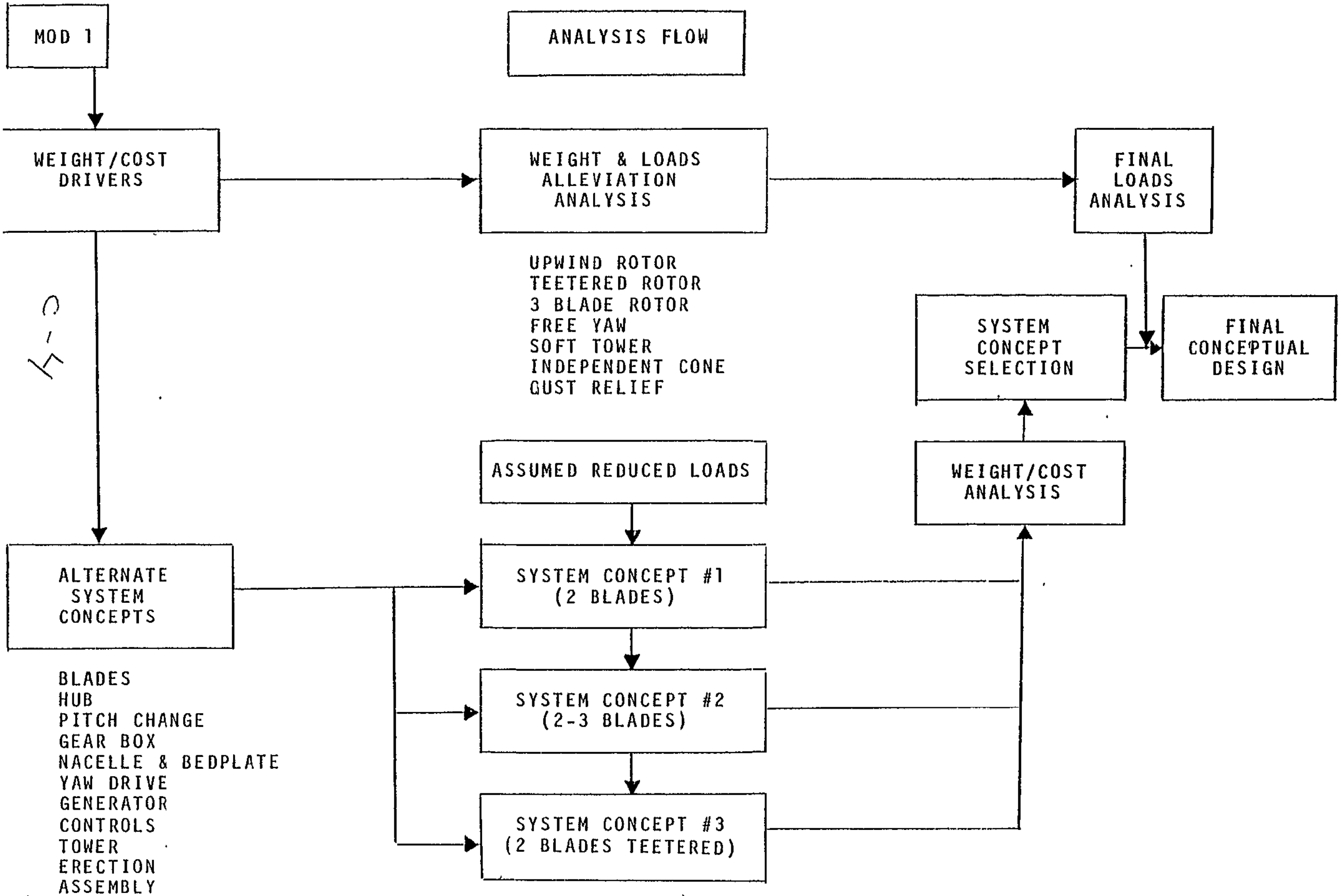
Two parallel efforts were then initiated:

1) To determine an assumed weight and load alleviation from MOD-1 system utilizing up-wind rotor, teetered rotor, 3-blade rotor and soft-tower (bending frequency placement). Neither time nor funds allowed the study of free-yaw, independent coning or gust relief solutions. Existing codes were not equipped for these analyses. Prior to the investigation of subsystems hardware reduction/simplification, we assumed results of the load alleviation would provide approximately 25% load reduction, which could be directly related to the size, weight and durability of major components. With that assumption in hand, the second part of the parallel design effort was initiated.

2) To define alternate system concepts utilizing new subsystems with reduced moving parts, reduced structure and increased overall system simplicity. Three system concepts were thereby defined that would achieve at least 25% load alleviation, each of which would incorporate a combination of the new subsystem configurations.

   Certain reduced in-size components were identified, applicable to all three concepts while other components required individual approaches for each concept. Preliminary weight and cost analyses were conducted on the three concepts, each of which had been detailed conceptually to the same degree of completion. Based on weight and cost considerations, one system was selected and a final simulated complete set of loads was conducted to assure the feasibiltiy of the concept. This final simulated load analysis was to establish the validity of the 25% loads reduction assumed earlier in the study and provide for additional weight and cost system and subsystem reduction, if practicable, which would then feed into a final concept design.

Throughout the study, while evaluating various component and subsystem candidates, both simplicity of assembly/fabrication techniques and processes as well as erection/installation costs and methods dominated the design effort.


MOD 1
ANALYSIS FLOW
WEIGHT/COST DRIVERS
WEIGHT & LOADS ALLEVIATION ANALYSIS
FINAL LOADS ANALYSIS
UPWIND ROTOR
TEETERED ROTOR
3 BLADE ROTOR
FREE YAW
SOFT TOWER
INDEPENDENT CONE
GUST RELIEF
SYSTEM CONCEPT SELECTION
FINAL CONCEPTUAL DESIGN
ASSUMED REDUCED LOADS
WEIGHT/COST ANALYSIS
ALTERNATE SYSTEM CONCEPTS
SYSTEM CONCEPT #1 (2 BLADES)
SYSTEM CONCEPT #2 (2-3 BLADES)
SYSTEM CONCEPT #3 (2 BLADES TEETERED)
BLADES
HUB
PITCH CHANGE
GEAR BOX
NACELLE & BEDPLATE
YAW DRIVE
GENERATOR
CONTROLS
TOWER
ERECTION
ASSEMBLY

1.3 THREE CANDIDATE SYSTEMS

The final three concepts include an up-wind reduced MOD-1 (2 blades fixed hub) an up-wind epicyclic gear configuration (3 blades fixed hub) an upwind or down-wind integral gear box (2 blades teetered hub). Each concept met the weight goal of less than 400,000 pounds and each came close to meeting the cost goals. The final loads determination provided an added bonus in both weight and cost reduction because loads were actually alleviated by approximately 40% rather than the assumed 25%. These weights have accounted for this additional load alleviation factor.

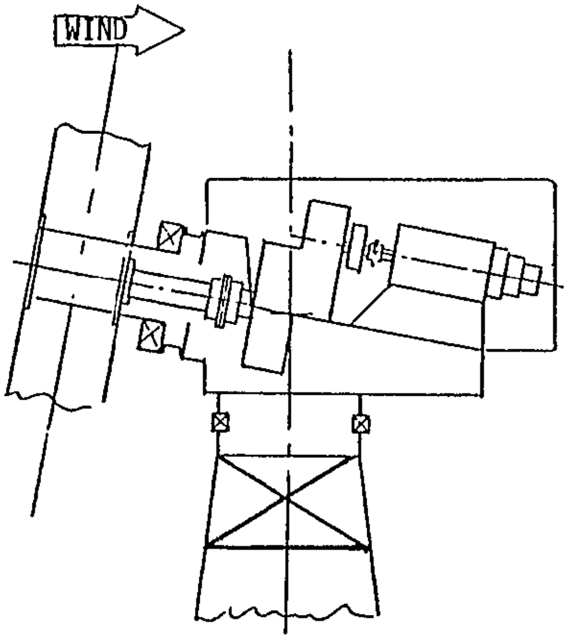
THREE CANDIDATE SYSTEMS

SYSTEM PARAMETERS	MOD-1	CONCEPT #1	CONCEPT #2	CONCEPT #3
# BLADES	2	2	3	2
PITCH CHANGE	FULL SPAN	PARTIAL SPAN	PARTIAL SPAN	PARTIAL SPAN
HUB	FIXED	FIXED	FIXED	TEETERED
ROTOR LOCATION	DOWNWIND	UPWIND	UPWIND	DOWNWIND
GEAR DRIVE	PARALLEL SHAFT	PARALLEL SHAFT (MOD-1)	EPICYCLIC	PARALLEL SHAFT (MOD-1)
ELECTRICAL GENERATION	CONSTANT SPEED SYNCHRONOUS	CONSTANT SPEED SYNCHRONOUS	CONSTANT SPEED SYNCHRONOUS	CONSTANT SPEED SYNCHRONOUS
YAW DRIVE	HYDRAULIC/ PINION	HYDRAULIC/ ACTUATOR	HYDRAULIC/ ACTUATOR	HYDRAULIC/ ACTUATOR
TOWER	TRUSS-STIFF	TRUSS-SOFT	SHELL-SOFT	SHELL-SOFT

Common elements among the concepts include the following.

- Two blades (MOD-1 design blade)
  - Concept #1
  - Concept #3
- Partial Span Control (replacing the MOD-1 pitch change mechanism)
  - Concept #1
  - Concept #2
  - Concept #3
- Fixed Hub
  - Concept #1
  - Concept #2
- MOD-1 geardrive
  - Concept #1
  - Concept #3 (outer structural housing modified to replace bed-plate and yaw bearing support)
- MOD-1 electrical generation and controls subsystem
  - Concept #1
  - Concept #2
  - Concept #3
- Hydraulic Actuator Yaw Drive
  - Concept #1
  - Concept #2
  - Concept #3
- Approximately 1.2P tower
  - Concept #1 (truss tower)
  - Concept #2 (conical shell tower)
  - Concept #3 (conical shell tower)

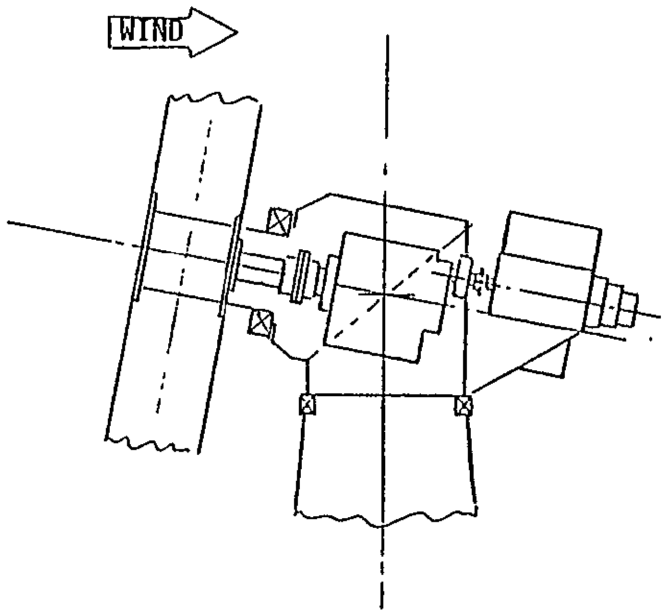
THREE CANDIDATE SYSTEMS



DESIGN CONCEPT #1
(REDUCED MOD - 1)

- FIXED HUB
- 2 BLADES
- UPWIND ROTOR
- PARTIAL SPAN CONTROL
- MOD-1 GEARBOX
- MOD-1 ELEC. GEN.
- TRUSS TOWER (SOFT)

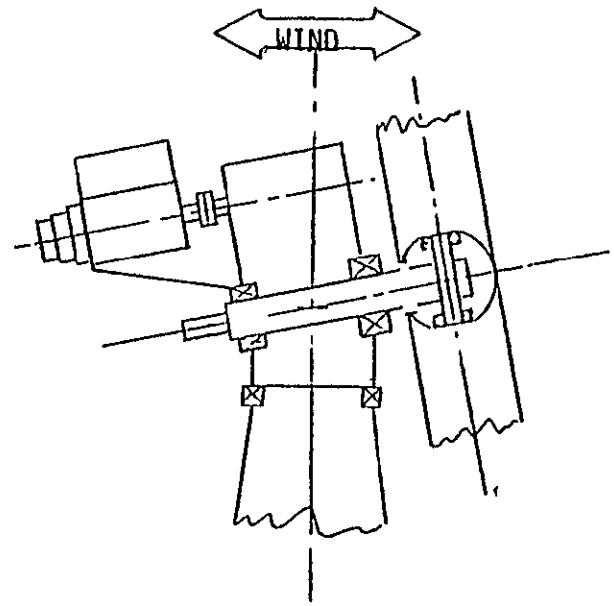
TOTAL WEIGHT
340,000 LBS



DESIGN CONCEPT #2
(EPICYCLIC GEAR)

- FIXED HUB
- 3 BLADES
- UPWIND ROTOR
- PARTIAL SPAN CONTROL
- EPICYCLIC GEARBOX
- MOD-1 ELEC. GEN.
- SHELL TOWER (SOFT)

TOTAL WEIGHT
355,000 LBS



DESIGN CONCEPT #3
(INTEGRAL GEARBOX)

- TEETERED HUB
- 2 BLADES
- DOWNWIND OR UPWIND
- PARTIAL SPAN CONTROL
- MOD-1 GEAR DRIVE
- MOD-1 ELEC. GEN.
- SHELL TOWER (SOFT)

TOTAL WEIGHT
320,000 LBS

FAVORABLE RESULTS COMPARED

TO MOD-1 WEIGHT OF 696,000

1.4 RESULTS

From the previous discussion, it was determined that all three concepts in various configurations can meet the weight objectives of the study program. Certain commonality of the MOD-1 hardware has been retained, such as the entire control system, power generation equipment and utility inter-connect, portions of all of the MOD-1 gearbox and the MOD-1 blade. Other innovations which include integral structure housing, partial span control, and conical shell tower at 1.2p do not require new technology development. Based on an early go-ahead utilizing the MOD-1 in-place team, a system of the characteristics of one of the three concepts can be rotating 19 1/2 months after contract go-ahead.

Because of major subsystems simplicity and loads alleviation, the concept selected is applicable to larger or smaller machines in wind regimes with low, moderate or high mean wind averages.

RESULTS

- SEVERAL ALTERNATE CONFIGURATIONS.
- UTILIZATION OF MOD-1 TECHNOLOGY & HARDWARE.
- ROTATING 19 1/2 MONTHS AFTER START.
- APPLICABLE TO LARGER OR SMALLER MACHINES.

SELECTED SYSTEM CAN DEMONSTRATE EARLY
ECONOMIC WIND ENERGY PRODUCTION

2.0 TECHNICAL DISCUSSION - Introduction

This technical discussion will cover the MOD-1 design drivers, the approaches to reaching the weight and cost goals, the major subsystem characteristics of the three candidate systems, the final loads analyses comparison and the selection summary.

TECHNICAL DISCUSSION

- MOD-1 DESIGN DRIVERS
- SYSTEM CONCEPT GOALS
- DESIGN REQUIREMENTS
- CONCEPT #1
- CONCEPT #2
- CONCEPT #3
- LOAD COMPARISONS
- CONCEPT #3 SELECTION
- SUMMARY

2.1 MOD-1 COST, WEIGHT AND DESIGN DRIVERS

The second unit MOD-1 costs, subsystem weight,specific costs in dollars per pound, and the summary of the design drivers for the major subsystems are shown here. It was determined early that fatigue loads, for which the entire MOD-1 system is designed, are the key and most influential cost and weight drivers of the MOD-1 system. In order to achieve the goals, it was necessary to reduce these fatigue loads so that the system would be stress-designed as much as possible, rather than fatigue designed, to make more efficient use of the structure.

In order to achieve the less than 400,000 pound goal, a weight at the top of the tower bogey was established at less than 200,000 pounds. Although the pitch change mechanism (torque control) was not a large weight driver in itself, the combination of weight and large over-hung moment contributed indirectly to a significant portion of the structural weight at the top of the tower. Similarly, the massive yaw control and structure contributed significantly to top-of-tower weight. Since tower weight is directly impacted by weight on top, then if these drivers could be reduced or eliminated, the potential for achieving the 400,000 machine was possible. Each subsystem, as well as assembly/ test and site preparation erection/and check-out contributes significant cost and weight. Each subsystem was independently investigated for potential simplicity and cost/weight reduction.

MOD-1

COST, WEIGHT & DESIGN DRIVERS

SUBSYSTEM	2ND UNIT COST K$	2ND UNIT COST %	WEIGHT K LB	WEIGHT %	SPEC.COST $/LB	DESIGN DRIVER
BLADES	280	7.6	36.0	4.8	7.75	FATIGUE LOADS, EMERGENCY FEATHER LOADS, TOWER SHADOW, WEIGHT
HUB	341	9.2	41.2	5.5	8.30	FATIGUE LOADS, BLADE ROOT MOMENTS, CONTROL MOMENTS
TORQUE CONTROL	161	4.4	42.6	5.7	3.77	GUST LOADS - MAX FORCE EMERGENCY SHUT DOWN - MAX. RATE STIFFNESS
BEARING & DRIVETRAIN	308	8.3	73.4	9.8	4.20	MAX. & CYCLIC TORQUE ROTOR LOADS ON BEARING
NACELLE/STRUCTURE	316	8.5	73.9	9.8	4.30	FATIGUE LOADS - LOAD PATH NO. OF COMPONENTS IN NACELLE GEARBOX MOUNTING BRG. MTG.
POWER GENERATION EQUIPMENT	290	7.8	70.1	9.4	4.15	POWER LEVEL, POWER QUALITY (ΔV), WTG/UTILITY PROTECTION
CONTROLS	173	4.7	8.1	1.1	21.35	UNATTENDED OPERATOR & UTILITY QUALITY POWER
YAW DRIVE SYSTEM	268	7.3	51.3	6.8	5.22	YAW TORQUE, OVERHUNG MOMENT, BEARING SUPPORT STRUCTURE
TOWER	360	9.7	352.7	47.1	1.00	LATERAL STIFFNESS/FATIGUE, TOWER SHADOW, LOW TEMP IMPACT REQUIREMENTS
ASS'Y & TEST	587	15.8	-	-	-	NO. OF PARTS, NO. OF JOINTS & CONNECTIONS, CRITICAL ALIGNMENTS, WEIGHT
SITE PREP. ERECT & CHECK-OUT	616	16.6	-	-	-	SITE CHARACT. & LOCATION, WEIGHT, SYSTEM COMPLEXITY
TOTALS	3700	100	749.3	100	4.95	

MAJOR DRIVERS
- FATIGUE LOADS
- FULL SPAN TORQUE CONTROL & RELATED COMPONENTS
- YAW SUPPORT/DRIVE & TOWER

2.2 CONCEPT GOALS

Certain judgment was used in establishing difficult and sometimes unrealistic goals for each major subsystem. Each subsystem design team was given the objective to produce a conceptual design meeting the overall MOD-1 operational requirements, but within cost and weight bogeys. Volumetric space was also allocated for those components on top of the tower in a manner similar to the way aircraft designers are constrained.

While the subsystem designers examined approaches to meet or exceed their allocated bogeys, the dynamic/stress team modified existing models to simulate modifications to MOD-1 that would achieve load alleviations through up-wind rotor, three-bladed rotor and/or teetered hub. The three system concepts evolved utilizing variations of two-blades, fixed or teetered hub, upwind, downwind, and soft-tower.

Emphasis was placed throughout on reducing labor during assembly and test as well as site preparation erection and checkout, both of which contribute significantly to today's second unit cost.

WHAT DOES IT TAKE TO MEET CONCEPT GOALS

SUBSYSTEM	2ND UNIT MOD 1-A BOGIES COST $K	WEIGHT K LB	2ND UNIT MOD-1 COST $K	WEIGHT K LB
BLADES	323	22	280	36.0
HUB	129	15	341	41.2
TORQUE CONTROL	65	8	161	42.6
NACELLE/STRUCTURE & DRIVE TRAIN	291	95	624	147.3
POWER GENERATION EQUIPMENT	178	46	290	70.1
CONTROLS	97	6	173	8.1
YAW DRIVE SYSTEM	112	33	268	51.3
TOWER	194	175	360	352.7
ASS'Y & TEST	258	-	587	-
SITE PREP, ERECT & CHECK-OUT	355	-	616	-
TOTAL	2002	400	3700	749.3

- REDUCE LOADS
- ELIMINATE FUNCTION, SIMPLIFY COMPONENTS
- INTEGRATE FUNCTIONS
- SIMPLIFY ASS'Y & ERECTION

2.3 CONCEPT DESIGN REQUIREMENTS

The system concept operational and design requirements are restated for the purpose of emphasizing that the selected configuration must meet the MOD-1 operational requirements.

CONCEPT DESIGN REQUIREMENTS

• RATED POWER	- 2000KW
• WIND SPEEDS @ 30' - MEAN - MIN. CUT IN - CUT OUT - MAX. SURVIVAL (NO SHEAR)	 - 18 MPH - 11 MPH - 35 MPH -150 MPH
• POWER GENERATION	- SYNCHRONOUS, UTILITY TYPE POWER QUALITY
• CONTROLS	- UNATTENDED OPERATION
• SYSTEM LIFE	- 30 YRS WITH OVERHAULS
• AVAILABILITY	- > 90%
• DESIGN LOAD CASES AS MOD 1	- CUMULATIVE FATIGUE FROM CUT IN TO CUT OUT - MAX UP & DOWN GUST - HURRICANE - NONOPERATING - EMERGENCY FEATHER

REQUIREMENTS SAME
AS PRESENT MOD-1

2.4 CONCEPT #1 (REDUCED MOD-1)

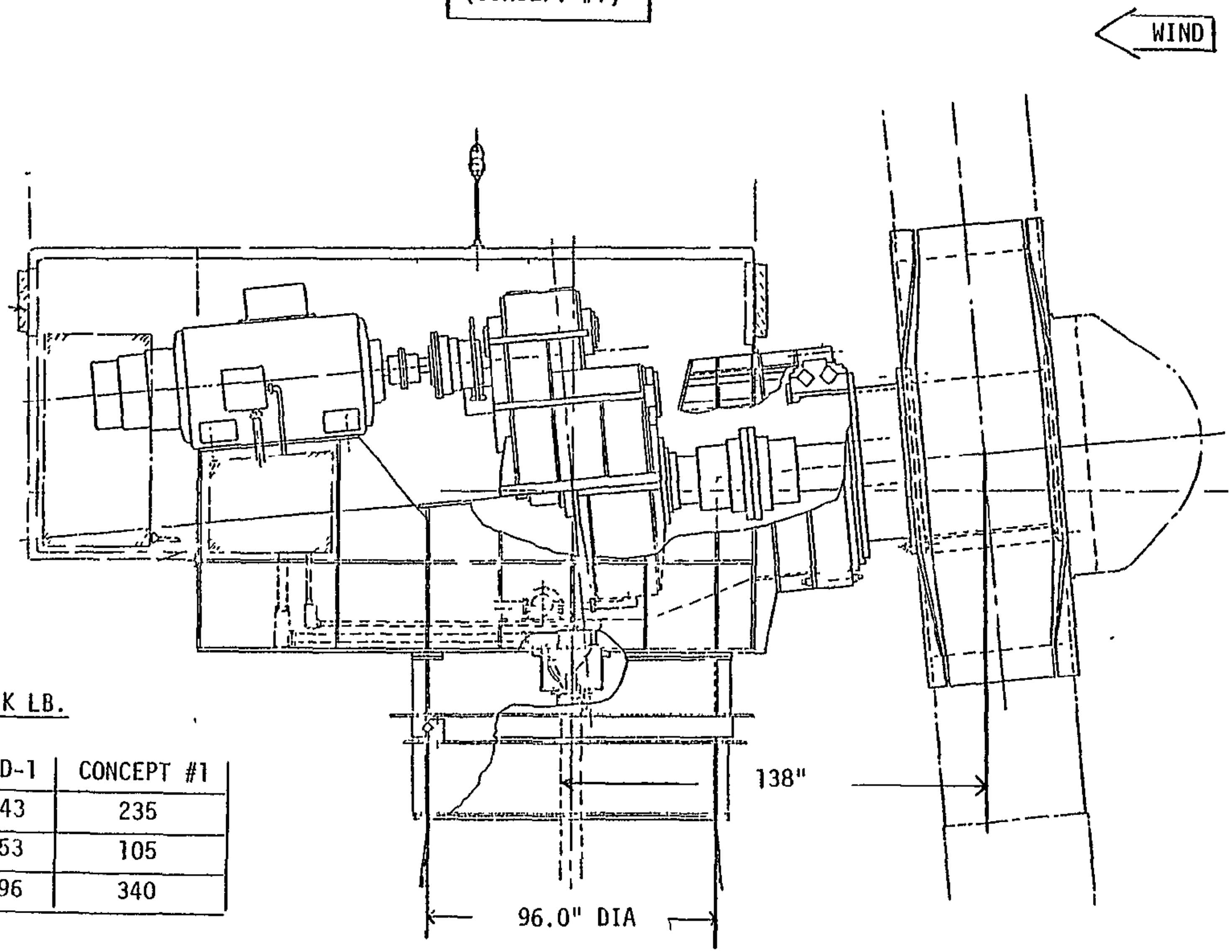
The first concept considered was a direct extrapolation from MOD-1 with minimum deviations from the MOD-1 Design. The similarities to MOD-1 are as follows:

- Two-blades (basic MOD-1 blade design)
- Gear box
- Electrical generation and control system
- Reduced (but similar fabrication) bed-plate and Nacelle structure
- Truss tower.

Deviations from MOD-1 System Design include the following:

- Fixed hub up-wind rotor
- Partial span torque control (in lieu of full-blade pitch change mechanism)
- Reduced main bearing diameter
- Modified hydraulic yaw actuation
- Soft tower

REDUCED MOD-1
(CONCEPT #1)



WEIGHTS K LB.

	MOD-1	CONCEPT #1
TOP OF TOWER	343	235
TOWER	353	105
TOTAL SYSTEM	696	340

The structural fatigue loads were alleviated by putting the rotor upwind. The elimination of the nacelle located pitch change mechanism and its rototating equipment allowed a significantly smaller bed-plate and support structure. The rotor center of gravity overhang was reduced 28 inches. Similarly, because of the lower weight on top of the tower and the simplified yaw support structure and actuator, the base diameter of the yaw system was reduced 48 inches. By reducing the weight at the top of the tower to approximately 235,000 pounds from 337,000 pounds and by selecting a soft truss tower, total weight at the base of the tower of 340,000 pounds is achieved.

Because of the significant commonality to the MOD-1 design, Concept #1 is in the class of being the lowest risk, earliest practicable implementation of a proposed next generation W.T.G.

CONCEPT #1
MAIN FEATURES

- FOLLOWS CLOSELY MOD-1 CONCEPT.
- PARTIAL SPAN TORQUE CONTROL.
- UPWIND 2 BLADE ROTOR AND REDUCED OVERHANG REDUCES LOADS.
- INCLINED AXIS.
- SIMPLIFIED YAW DRIVE - REDUCED YAW BEARING DIAMETER 8 FT.
- SAME HARDWARE FOR DRIVE TRAIN, POWER GENERATION EQUIPMENT & CONTROLS.
- TRUSS TOWER.

WEIGHTS		
TOP OF TOWER	235,000	LBS.
TOWER	105,000	LBS.
TOTAL WEIGHT	340,000	LBS.

2.5 CONCEPT #2 (EPICYCLIC GEAR)

From the standpoint of reduced fabrication complexity, Concept #2 has a significant advantage over Concept #1. Instead of the built-up steel fabricated bed-plate, it utilizes a 6-Ft. tubular steel 1/2 inch pipe that is cut at an approximate 45 degree angle and turned to house a smaller (in weight and size) epicyclic gear-box.

The features retained from the MOD-1 are as follows:

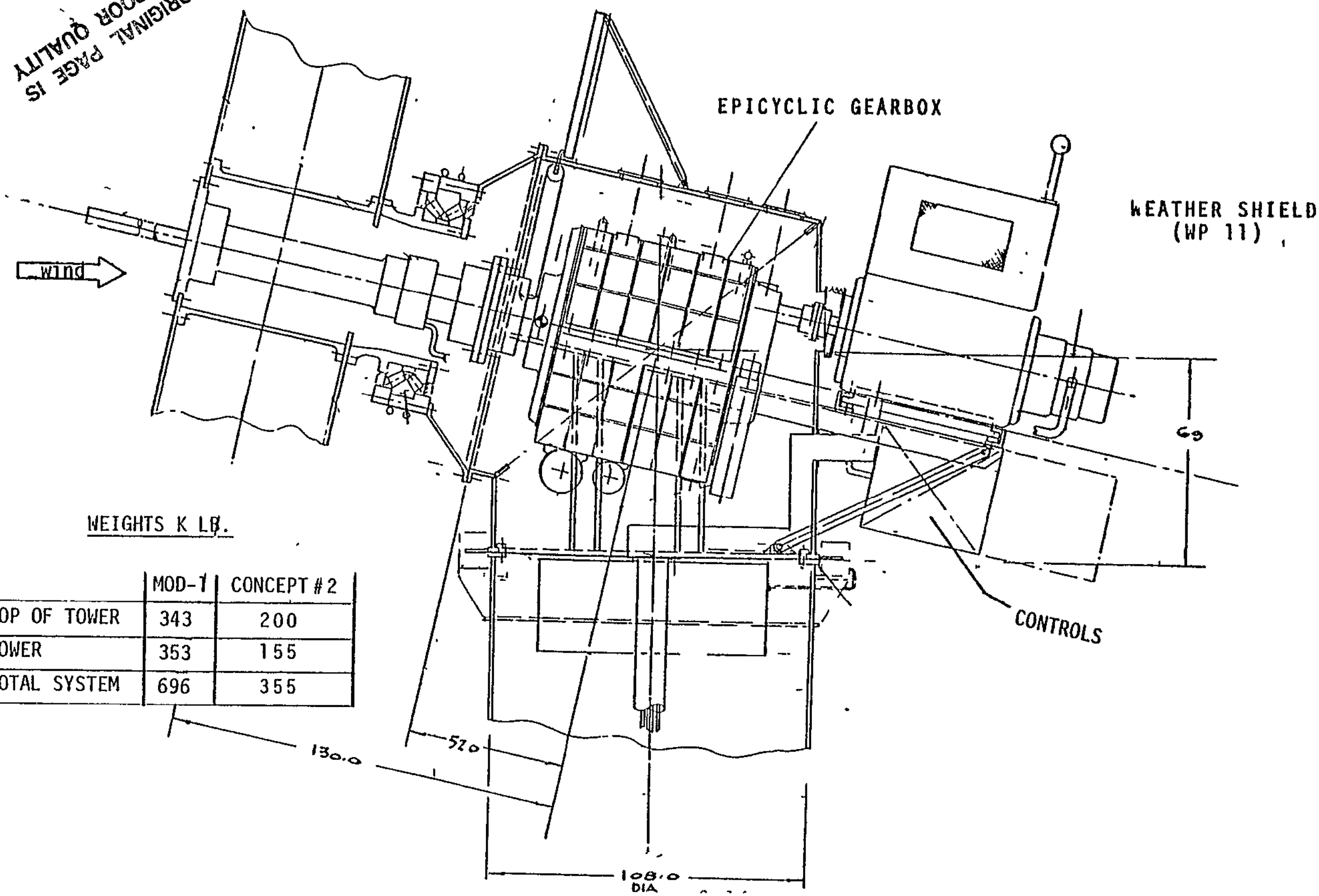
- Electrical generation and controls

Features which depart from the MOD-1 design or require entirely new designs include the following:

- Three-blade up-wind rotor
- Partial span torque control
- Epicyclic gear drive
- Cylindrical shell housing and structure
- Modified yaw actuator
- Conical shell tower (1.2p)

In addition to the 3-bladed rotor a 2-bladed hub was designed for this system that would accommodate up-wind rotor as well. With this two-bladed configuration, the MOD-1 blade can be used.

CYLINDRICAL SHELL STRUCTURE NACELLE
(CONCEPT #2)



WEIGHTS K LB.

	MOD-1	CONCEPT #2
TOP OF TOWER	343	200
TOWER	353	155
TOTAL SYSTEM	696	355

By reducing the weight at the top of the tower to 200,000 pounds and using a cylindrical soft (1.2p) tower the total weight of the system is 355,000 pounds, If, however, a truss-tower is used with the configuration, a total system weight of 305,000 pounds can be achieved. Although Concept #2 was the lowest weight, in this configuration with a Truss-Tower, it ended up being the highest in cost because of the cost penalty of the epicyclic gear and the labor associated with the erection.

CONCEPT #2
MAIN FEATURES

- 3 BLADED UPWIND ROTOR.
- PARTIAL SPAN TORQUE CONTROL.
- EPICYCLIC GEARBOX.
- TUBULAR BEDPLATE STRUCTURE SERVES AS FAIRING.
- JOURNAL BEARING FOR YAW SUPPORT/DRIVE.
- TUBULAR TOWER.

WEIGHTS		
TOP OF TOWER	200,000	LBS.
TOWER	155,000	LBS.
TOTAL	355,000	LBS.

2.6 CONCEPT #3 (INTEGRAL GEAR-BOX)

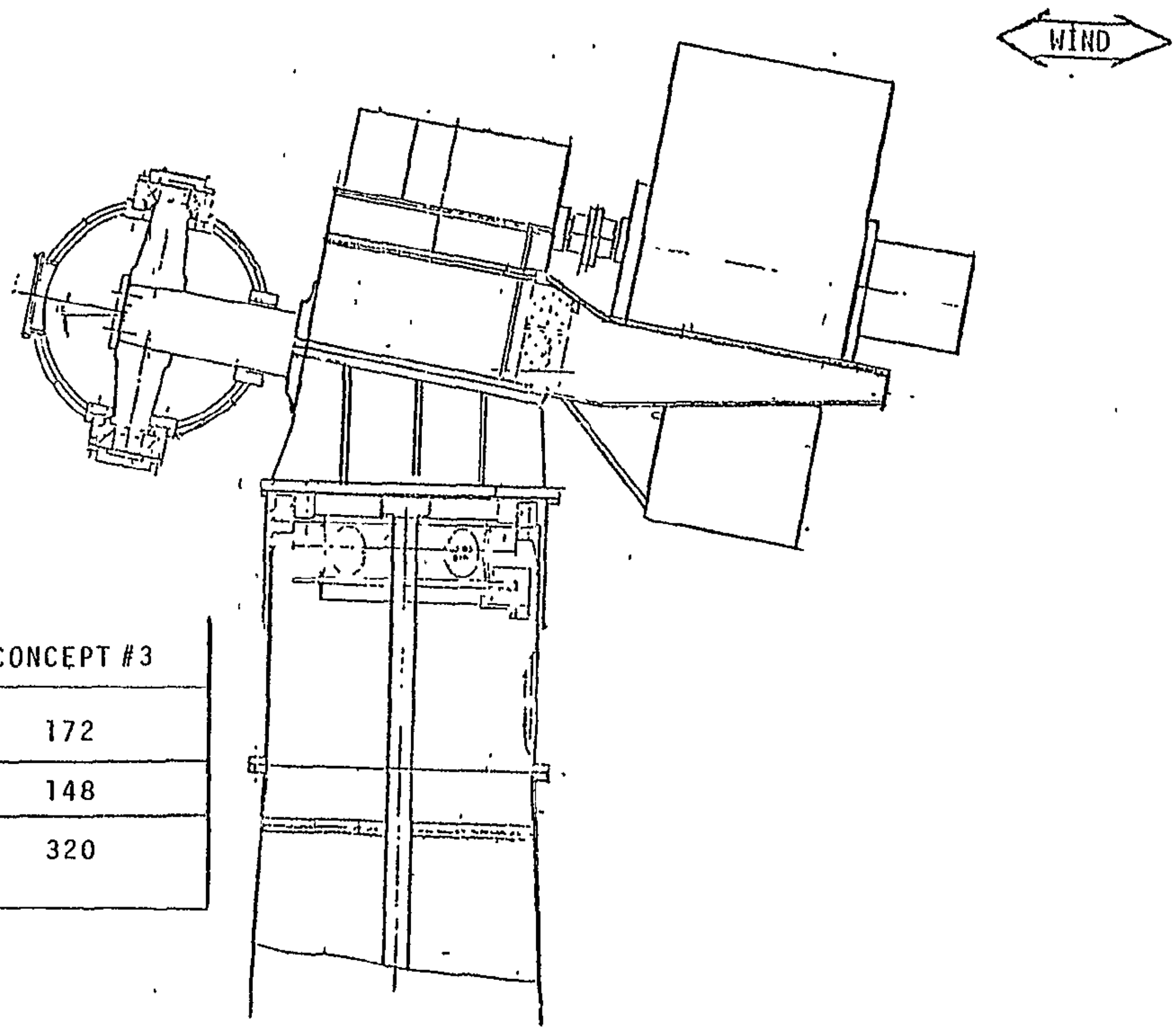
The main MOD-1 features retained in Concept #3, include the following:

- Two blades, (basic MOD-1 blade design)
- Gear-Drive
- Electrical Generation and Controls

The features which deviated from MOD-1 for our new design include the following:

- Teetered-Hub (Up-wind or Down-wind),
- Partial Span Torque Control
- Modified Yaw Support and Drive
- Bed-Plate and nacelle replaced by the gear-box integral structure (gear box housing has been designed to integrate directly with the top of the tower, thus eliminating the need for special conical housing.)
- Conical Shell Tower (1.2p)

INTEGRAL GEARBOX STRUCTURE NACELLE
(CONCEPT #3)



WEIGHTS K LB

	MOD-1	CONCEPT #3
TOP OF TOWER	343	172
TOWER	353	148
TOTAL SYSTEM	696	320

The most significant load alleviation features of Concept #3 are:

1) The simple teetered hub mounted on the gear train drive-shaft and,
2) the use of the integral gear-box housing as bed-plate and the main support for the yaw control and support.

By reducing the weight at the top of the tower to 172,000 pounds and providing a conical shell soft-tower of 148,000 pounds the total system weight at the base of the tower is 320,000 pounds.

Concept #3 is the selected system configuration and is described in more detail in 3.0 entitled, "Selected System Description."

CONCEPT #3
MAIN FEATURES

- 2 BLADED TEETERED HUB DOWNWIND
- PARTIAL SPAN TORQUE CONTROL.
- ROTOR SUPPORT ON L/S GEARBOX SHAFT - INCLINED AXIS.
- LOWER GEARBOX STRUCTURE USED AS YAW BEARING SUPPORT STRUCTURE.
- SIMPLIFIED YAW DRIVE - 6 FT. YAW BEARING DIA.
- TUBULAR TOWER.
- GEAR DRIVE, POWER GENERATION EQUIPMENT & CONTROLS SAME HARDWARE AS MOD-1.

WEIGHTS	
TOP OF TOWER	172,000 LBS.
TOWER	148,000 LBS.
TOTAL WEIGHT	320,000 LBS.

2.7 TRADE STUDY LOADS ANALYSIS

Several steps are required using GE's analytical codes to develop comparative dynamics and stress analysis to validate the assumption of a minimum 25% loads reduction. Since the MOD-1 system was modeled in detail as a down-wind two-bladed rotor with the existing geometry, it was determined that the most expeditious approach to establish the impact of various configurations such as up-wind, three-blades, and teetered rotor would be to retain basic geometries (to establish the load changes on a one-for-one basis using the MOD-1 as the baseline). Minor code modifications were therefore required to simulate the conditions of an up-wind rotor and a teetered rotor. Creation of a new code for three blades was required to understand the impact of three versus two blades. At the completion of the code modifications and their verification for authenticity, the MOD-1 (Baseline) was modeled individually as a 2-blade up-wind fixed rotor, 2-blade down-wind teetered rotor, and a 3-blade up-wind fixed rotor. Various combinations were run to determine the impact of soft-tower and reduced weight at the top of the tower. When these loads were finally completed and analyzed, hand calculations were performed on the selected system configuration (Case #6) to account for geometry differences which reduced moments at the yaw, base of the tower and hub/shaft. The following discussions summarize the results of this final loads analysis and comparison.

LOAD CASES AND DESCRIPTIONS

CASE #	SYSTEM DESIGN	# BLADES	WEIGHT (K-#)	ROTOR FIXED/TEETERED	WIND UPWIND/ DOWNWIND	TOWER SHELL/ TRUSS	TOWER FREQ.
BASELINE							
1	MOD-1	2	340	F	D	T	3.2
2	MOD-1	2	340	F	U	T	3.2
3	MOD-1	2	340	T	D	T	3.2
4	MOD-1	3	340	F	D	T	3.2
COMBINATIONS							
5	MOD-1	3	340	F	U	S	1.4
6	MOD-1A	2	200	T	D	S	1.2
7	MOD-1A	2	200	T	U	S	1.4

LOAD COMPARISONS

The load calculations for all cases investigated are shown as a fraction of the MOD-1 loads (Case -1); since the bending moments are the dominant design drivers, only these are shown. Except for the blade flapwise (My), blade cordwise (Mz) and yaw torque (Mx), shown separately, remaining moments are given as combined bending moments not including torsion.

All loads are calculated for the MOD-1 geometric configuration, i.e. same offsets for rotor to the hub bearing and tower centerline. Partial span control was not considered in the loads calculation. Blade coning is 9° for MOD-1 downwind baseline (Case 1); for all other cases the rotor has zero coning.

The comparison was made for the 35 mph steady state case which is the design driver for the fatigue loads. Peak loads for gust conditions were only calculated for the selected configuration (Case 6).

Based on these load calculations and the comparisons shown on the attached charts the following conclusion can be drawn:

(1) The most significant system load reductions result from a teetered rotor or a 3-bladed upwind rotor

(2) A most significant reduction results from teetering (note that the reduced loads for the 3-bladed rotor mostly stem from the fact that the weight of a single blade was reduced to 2/3 of the present blade in order to stay consistent; considering the reduced cross-sectional properties, blade stresses in this configuration are essentially the same as for the fixed 2-blade upwind rotor).

LOAD CALCULATIONS

35MPH FATIGUE LOADS

CASE #	DESCRIPTION	BLADE RETEN'N FLAP MY	BLADE RETEN'N EDGE MZ	HUB/SHAFT RESULTANT	YAW BRG. RESULTANT	TOWER BASE RESULTANT	YAW TORQUE MX	COMMENTS
BASELINE								
1	MOD-1* DOWN-WIND FT-LBS(M)	1.07	1.08	2.08	.46	6.30	.72	MOD-1 ACTUALS
	% MOD-1	1.0	1.0	1.0	1.0	1.0	1.0	
2	MOD-1 - UPWIND	.77	.94	.28	1.22	.41	.54	INDIVIDUAL LOAD CASES WITH MOD-1 WT. AND TOWER FOR COMPARATIVE PURPOSES.
3	MOD-1 - TEETERED	.40	.87	.20	.82	.56	.21	
4	MOD-1 - 3 BLADE	.48	.61**	.54	.75	.32	.52	
COMBINATIONS								
5	MOD-1 - 3 BLADES, UPWIND, 1.4P TOWER	.47	.60**	.12	.18	.04	.18	BEST LOAD COMBINATION RESULTS
6	MOD-1A - TEETERED, DOWN-WIND, 1.2P TOWER	.59	.88	.29	39	.20	.15	SELECTED MOD-1A
7	MOD-1A - TEETERED, UPWIND, 1.4P TOWER	.37	.90	.26	.71	.30	.18	

* ALL CASES EXCEPT #1 ARE ZERO CONING: CASE #1 IS 9^{0} CONING.
** FOR 3-BLADE VERSION BLADE WEIGHT & CHORD THICKNESS ARE ASSUMED 1/3 REDUCED

(3) For a teetered hub, the cyclic tower base loads are so significantly reduced (more than a factor five) that peak bending moments from gusting becomes the driver. (For structural shells peak loads become dominating when they are higher than $\approx$ 3 to 5 times cyclic loads depending on type of weld.)

(4) For a teetered hub, up and down wind rotor position result in similar load reductions.

(5) A soft tower (1st lateral bending frequency $\cong$ 1.2) has no significant impact on loads as long as coincidence of higher tower bending modes with blade flapwise collective modes are avoided.

(6) The main drawback of a fixed upwind compared to a teetered rotor are (Case 2 vs. Case 6).

a) Cyclic flapwise blade bending moment more than 1 1/2 times higher

b) Cyclic bending moments on top of tower (yaw bearing) more than 3 times higher, peak loads 2 times higher

c) Cyclic bending moments on tower base more than 2 times higher

d) Cyclic and Maximum yaw torques about twice as high.

(7) The major reduction in flapwise gust loads (more than a factor 10) for the down gusts (35 ---20 mph) stems in part from the zero-coning; this benefits the blade bending loads which is a design driver for MOD-1. For the same reason the flapwise bending for the upgust remained as high as for MOD-1. Note the significant reduction in gust-peak loads for <u>shaft and yaw drive torque</u> due to teetering.

LOAD CALCULATIONS

35MPH PEAK LOADS

CASE #	DESCRIPTION	BLADE RETEN'N FLAP MY	BLADE RETEN'N EDGE MZ	HUB/SHAFT RESULTANT	YAW BRG. RESULTANT	TOWER BASE RESULTANT	YAW TORQUE MX	COMMENTS
BASELINE								
	FT-LBS. M	3.34	1.29	2.08	1.22	10.80	.75	MOD-1 ACTUALS
1	MOD-1 % MOD-1	1.0	1.0	1.0	1.0	1.0	1.0	
2	MOD-1 - UPWIND	.47	.93	.28	.95	.63	.54	
3	MOD-1 - TEETERED	.44	.89	.20	.70	.72	.20	
4	MOD-1 - 3 BLADE	.33	.62	.60	.66	.57	.52	
COMBINATIONS								
5	MOD-1 - 3 BLADES, UPWIND, SOFT TOWER	.34	.61	.13	.63	.43	.19	
6	MOD-1A - TEETERED, DOWN-WIND, 1.2P TOWER	.50	.88	.42	.52	.48	.17	
7	MOD-1A - TEETERED, UPWIND, 1.4P TOWER	.43	.91	.26	.53	.47	.20	

LOAD CALCULATIONS

PEAK GUST LOAD

CASE #	DESCRIPTION		BLADE RETEN'N FLAP MY	BLADE RETEN'N EDGE MZ	HUB/SHAFT RESULTANT	YAW BRG. RESULTANT	TOWER BASE RESULTANT	YAW TORQUE MX	COMMENTS
BASELINE									
1	MOD-1	UPGUST -FT-#(M)	2.29	1.43	2.23	2.11	12.70	1.48	
		DOWNGUST-FT-#(M)	3.48	.84	1.42	.96	5.51	1.12	
		% MOD 1	1.0	1.0	1.0	1.0	1.0	1.0	
2	MOD-1 - UPWIND								
3	MOD-1 - TEETERED								
4	MOD-1 - 3 BLADE								
COMBINATIONS									
5	MOD-1 - 3 BLADES, UPWIND, SOFT TOWER								
6	MOD-1A - TEETERED, DOWNWIND, 1.2P	UG	1.02	.80	.23	.82	.81	.16	
		DG	.10	.78	.23	.25	.20	.08	
7	MOD-1A - TEETERED, UPWIND, 1.4P TOWER								

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2.8 Configuration Selection Summary

Several factors contributed to the final selection. The most significant are summarized here.

Two Blade vs. Three Blade

The preliminary loads alleviation analyses indicated (late in the study) that a three blade up-wind rotor provides a marginal advantage, load-wise, to a two blade (teetered) system. A three blade rotor, however would require a complete blade redeisgn of lighter weight and approximate equal solidity factor. Since the MOD-1 blade design and tooling are complete, it was determined early in the study that the cost of a new blade or blade re-design must be offset by other significant cost reductions. Because of the short duration and funds limitation, complete cost/performance trade-offs were considered out of the study scope. As future blade costs are reduced, however, a 3-blade configuration may be more cost effective. Additional analyses and cost trades incorporating the full third blade cost and cost savings resulting from further structural weight reduction (taking advantage of the reduced loads) may provide a cost advantage for future generation systems. For the purpose of this study, utilization of the MOD-1 blade design and in-place tooling clearly provides cost and schedule advantages for the next generation WTG.

Fixed vs. Teetered Rotor

In order to alleviate the loads to achieve a stress limited rather than fatigue limited system, both teetered down-wind and fixed up-wind rotor systems are serious candidates. Concept #3 can accommodate either up-wind or down-wind, teetered or fixed rotor. Until the final peak loads and in-depth cost trade-offs between fixed and teetered rotors are in-hand, Concept #3 with a teetered down-wind rotor was selected as the tentative candidate. In addition to the superior preliminary loads alleviation provided by the teetered system, a smaller tilt angle or shorter shaft overhang provide cleaner geometry and lighter weight.

Upwind vs. Downwind Rotor

By comparing the preliminary load alleviation data for the up-wind vs. down-wind fixed rotor cases, the preferred configuration for both two and three blades is up-wind. However, since teetering significantly reduces yaw moment and flapwise blade bending a teetered rotor is not sensitive to either the up-wind or down-wind configurations.

CONFIGURATION SELECTION SUMMARY

- USE MOD-1 BLADES WITH MINIMUM MODIFICATIONS ⟶ 2 BLADE ROTOR
- ALLEVIATE LOADS ⟶ TEETERED ROTOR
- SIMPLIFY COMPONENTS ⟶ PARTIAL SPAN TORQUE CONTROL
  ⟶ SIMPLIFIED YAW DRIVE/REDUCED BEARING DIAMETER
- INTEGRATE FUNCTIONS ⟶ ROTOR SHAFT = L/S GEARBOX SHAFT
  ⟶ BEDPLATE = GEARBOX STRUCTURE
- MAKE EXTENSIVE USE OF MOD-1 HARDWARE ⟶ SAME GEAR DRIVE
  ⟶ SAME POWER GENERATION EQUIPMENT
  ⟶ SAME CONTROL EQUIPMENT
- SIMPLIFY ASS'Y. & ERECTION ⟶ SHELL TOWER
  ⟶ SINGLE LIFT

Stiff vs Soft Tower

Each concept was configured with both a soft and a rigid tower, which had natural frequencies at 1.2, 1.4, 1.7, 2.2, and 3.4P. By avoiding critical frequency harmonics of other subsystems it became clear that the lowest cost, lowest weight tower should be a soft-tower in the 1.2P range. In order to assure complete system stability it is necessary that the final configuration be tuned in the field with appropriately placed weights after erection during check-out of the system.

Use of existing or designed MOD-1 hardware is a benefit that resulted from the final analysis and trade between the three system concepts and allowed the economic use of major hardware components. These include:

- Existing gears from MOD-1 #2 unit gear box
- Existing MOD-1 power generation equipment
- Existing MOD-1 control equipment
- Same blade design

The selection of Concept #3 incorporating these MOD-1 components, further confirms credibility of design without major redevelopment and system design. Features such as partial span torque control utilizing 15% of the blade; the simplified yaw drive, reducing the yaw bearing diameter; the integral rotor shaft and low-speed gear-box shaft; and integral bed-plate/gear box structures contributed to simplifying components and the integration of functions within the structures. These features were considered to be most cost effective and most-weight economic. Again, Concept #3 could accommodate all of these features. Several fabrication steps were either simplified or eliminated entirely. Similarly, the conical, cylindrical tower was selected because of low-cost of its single lift procedure.

3.0 SELECTED SYSTEM DESCRIPTION

Based on the findings summarized in Section 2.8 Concept #3 was selected.

- Rotor - two blades 200 ft. diameter, 35 RPM, tilted axis, air foil 44xx.
- Hub - teetered with fixed blade root
- Rotor Torque Control - 15% tip control hydraulicly actuated, servo controlled
- Nacelle Structure - lower part of gearbox housing carries rotor loads.
- Yaw Drive - Single row cross roller bearing with hydraulic cylinder/disc brake drive
- Electrical Connection - cable twist
- Tower - tubular single member 1.2P bending frequency.

SELECTED SYSTEM DESCRIPTION

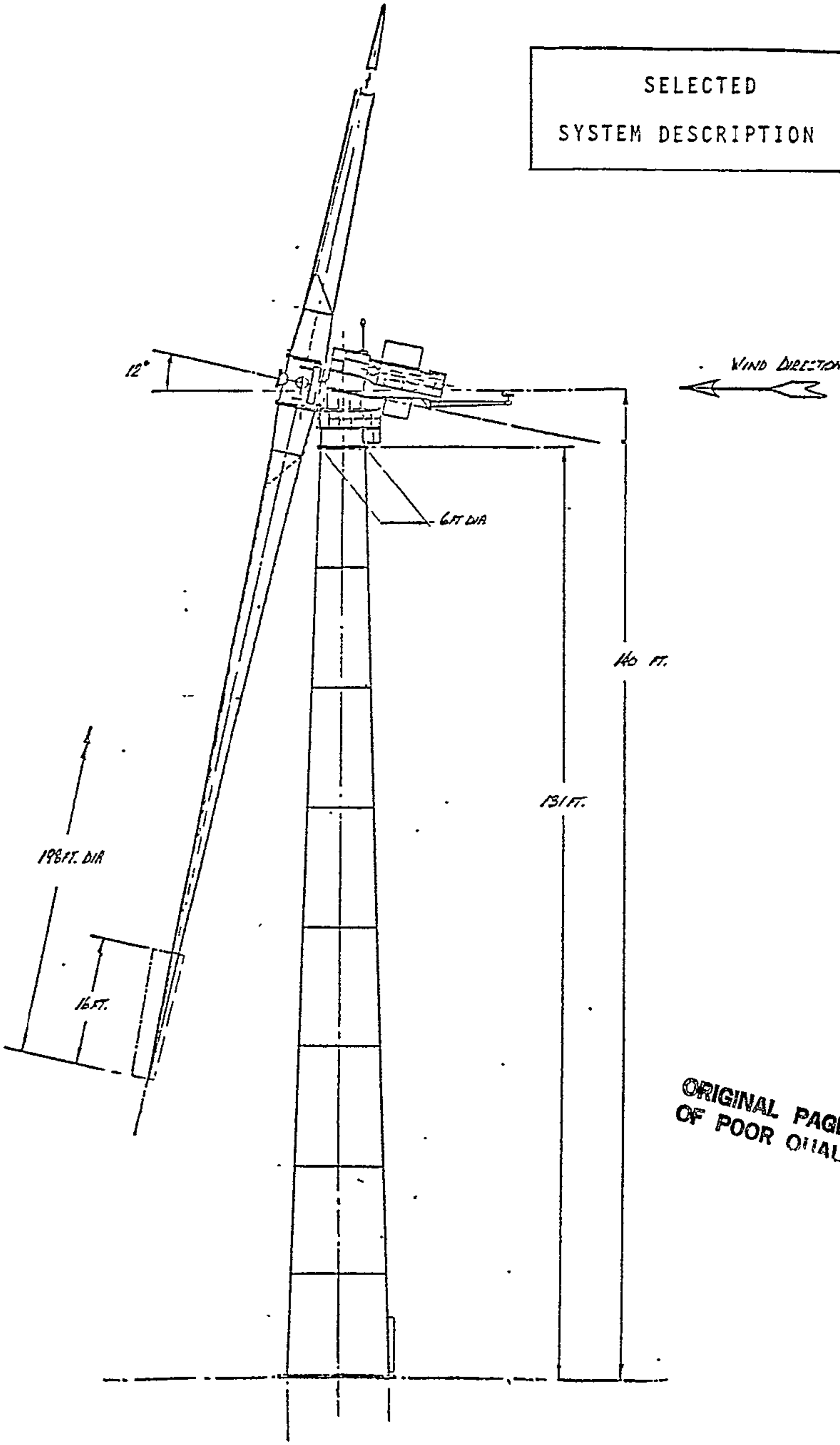
- BLADE ASSEMBLY
- PARTIAL SPAN TORQUE CONTROL
- TEETERED HUB
- GEARBOX
- YAW DRIVE
- POWER GENERATION & CONTROLS
- TOWER
- TECHNICAL RISK ASSESSMENT
- WEIGHT COMPARISON

Design Description

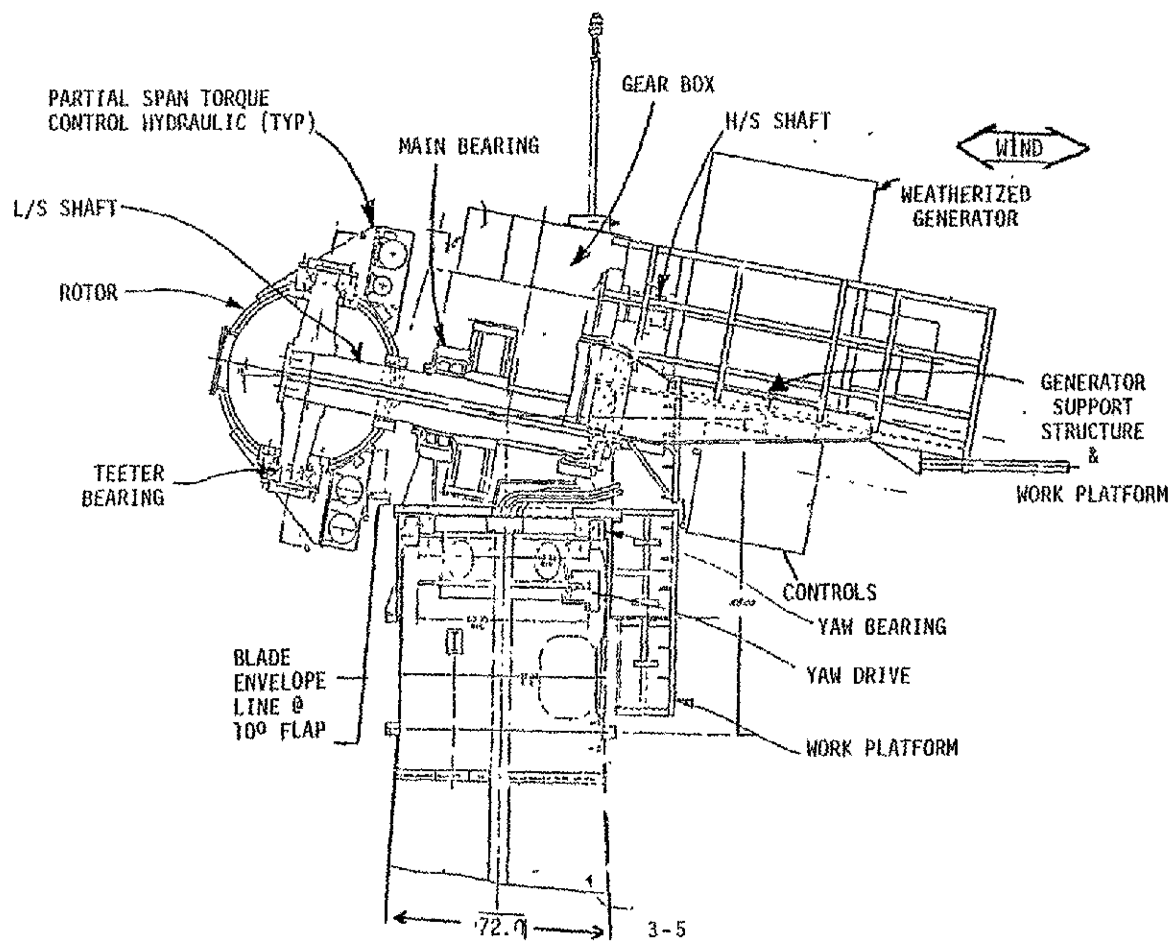
In order to maximize the benefit from lessons learned from the MOD-1 design, MOD-1 components and techniques were used as far as economically prudent. The aim was to reduce total cost, minimizing recurring costs such as hardware and erection, as well as keeping engineering and other non-recurring costs compatible with 2nd generation system design. In the following description, therefore, reference to the MOD-1 design will be made in order to point out the similarities in design and the natural evolution of this concept from the MOD-1 design and experience.

SELECTED

SYSTEM DESCRIPTION



INTEGRAL GEARBOX STRUCTURE NACELLE
(CONCEPT #3)

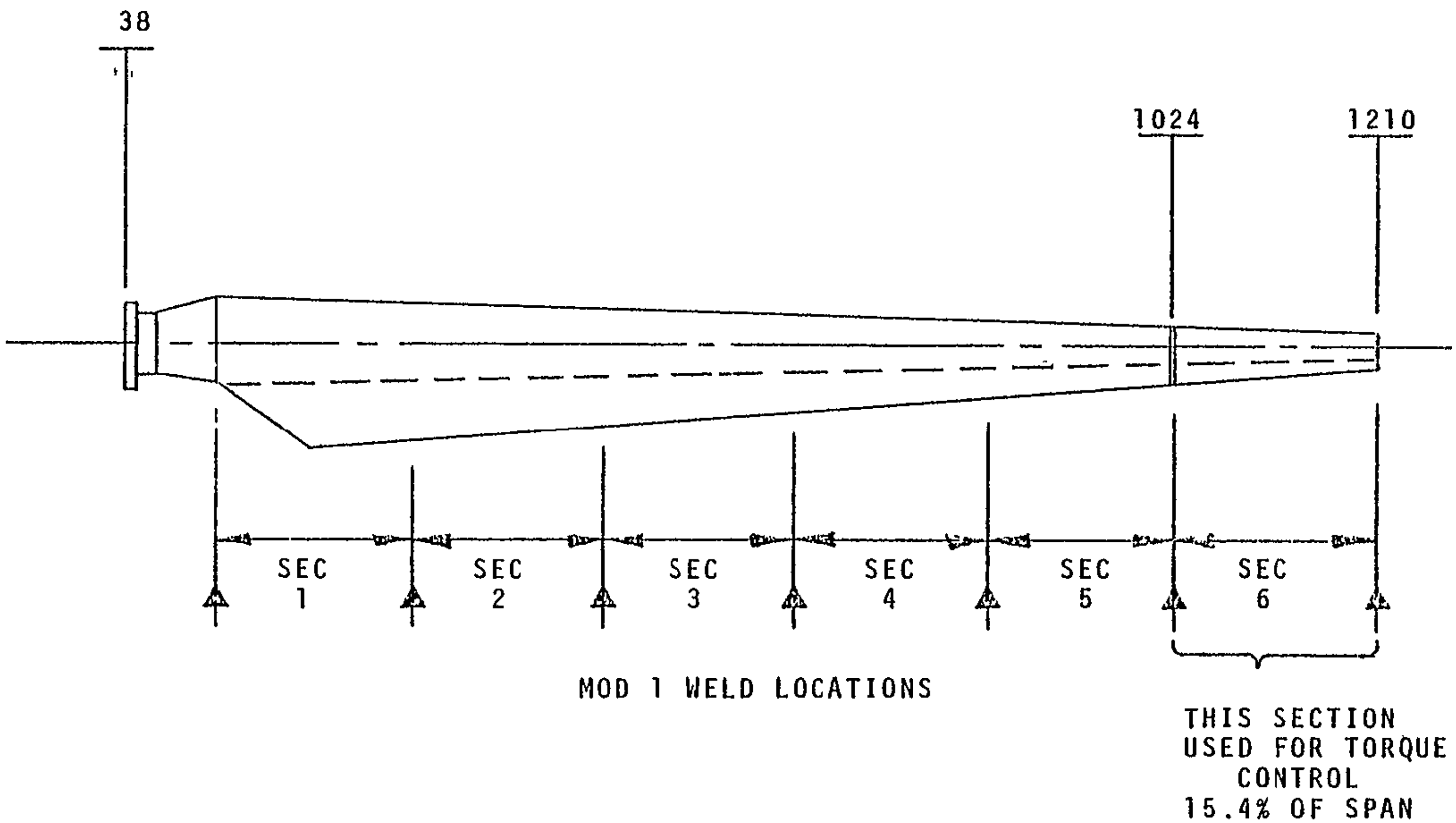


3.1 BLADE ASSEMBLY

The selected blade is identical in size, weight, exterior geometry, materials, fabrication and assembly techniques as the MOD-1 blade. It utilizes the existing tooling already purchased for MOD-1 Program. No changes are required to the exterior geometry, including the blade root section, leading and trailing edges, cord and twist. Station 1024 was selected for incorporating the partial span control because it exists at one of the weld stations, between section 5 and 6. A careful analysis indicates that approximately 15% of the span provides sufficient torque control to satisfy the operational requirements.

Since the selected system was zero coning (compared to 9 degree on the MOD-1) a slight modification to the spar could be required. The preliminary load alleviation analysis indicates that the MOD-1 blade will be slightly over-designed which means that skin thickness and weight can be reduced. As a matter of expediency in economics, however, a slightly heavier blade using the exact replica of MOD-1 is considered a reasonable approach for the selected system. It is necessary to note, however, that any blade within the 20,000 lb weight range can be used as a blade for this design.

BLADE ASSEMBLY

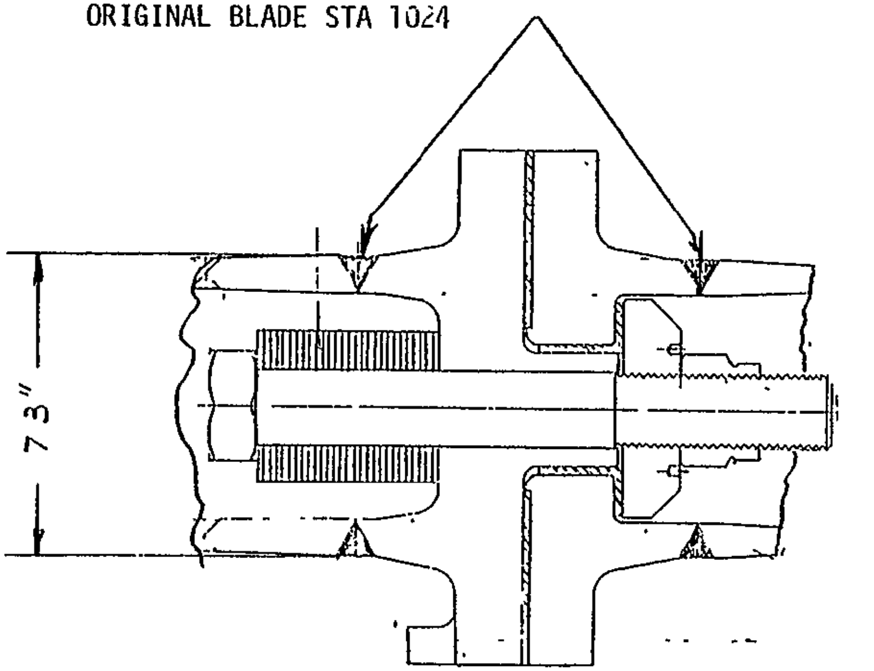


PRESENT MOD-1 BLADE USABLE

WITH MINOR MODIFICATIONS FOR TORQUE CONTROL

Because of the simplicity of the torque control design the structural integrity of the spar of the MOD-1 blade is retained. The additional weight for each partial span is offset by removal of existing MOD-1 frequency trim weights at the blade tip. The only additional testing required will be incident to the partial span torque since fatigue sample tests have been conducted on the MOD-1 program.

BLADE ASS'Y. FEATURES



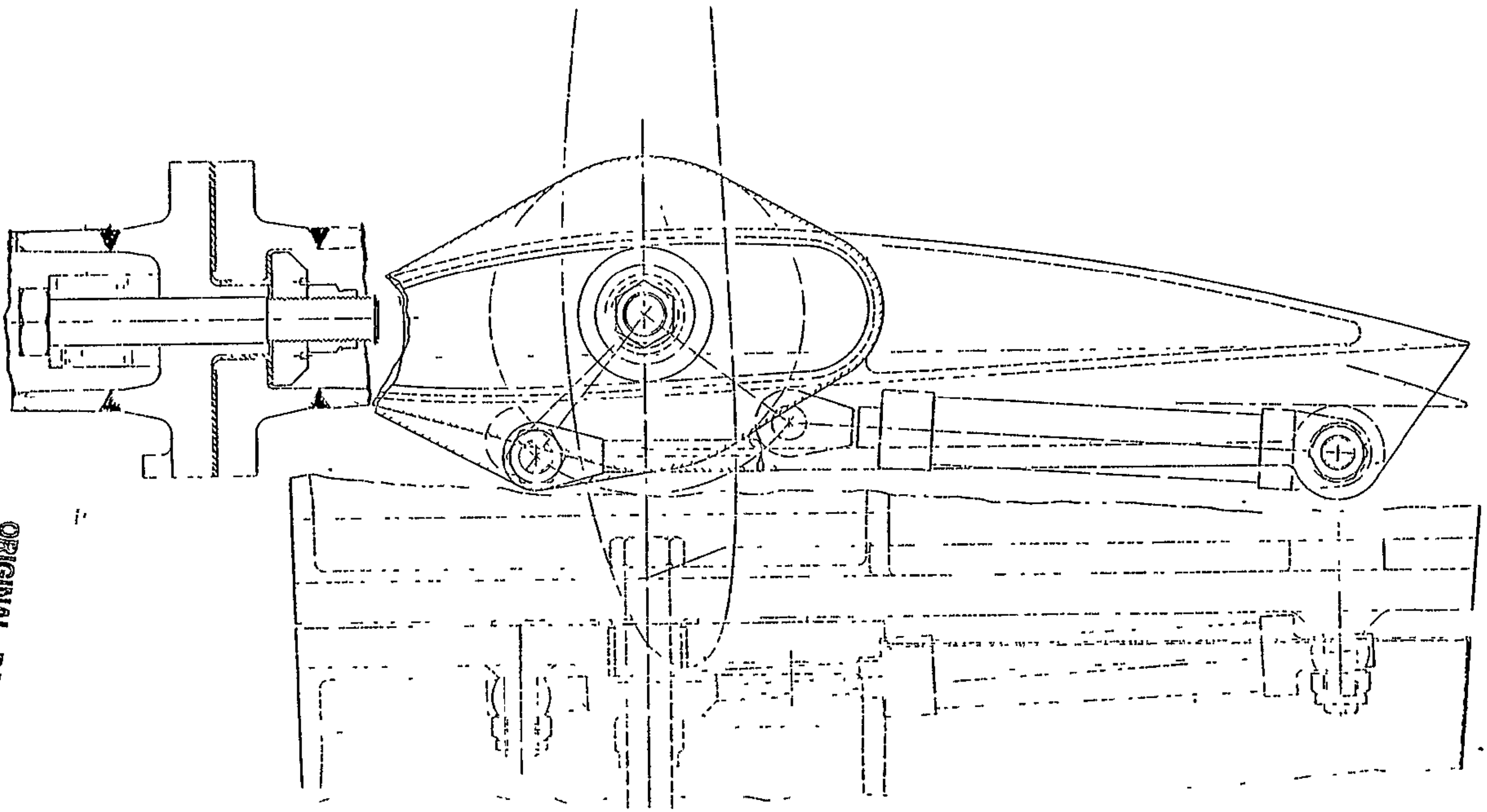
- PRELIMINARY STRESS CALCULATION INDICATE MINOR BLADE CHANGES REQUIRED.
- ADDITIONAL WEIGHT FOR TORQUE CONTROL COMPENSATED BY REDUCTION OF TIP WEIGHTS
- ROOT FLANGE INTERFACE IDENTICAL
- NO INTEGRATION PROBLEMS WITH PARTIAL SPAN TORQUE CONTROL EXPECTED

USE OF IDENTICAL BLADE CONFIGURATION MINIMIZES COST & SCHEDULE RISK

3.2 PARTIAL SPAN TORQUE CONTROL

Four different concepts were investigated in enough detail to determine a preferred approach from the standpoint of both cost and weight. These included electric motor/gear-drive, hydraulic motor with gear-drive, hydraulic actuator with a long shaft and the hydraulic actuator with the bulk-head bearing concept that was finally selected. Prior to the final selection of the partial span approach to provide for blade pitch, additional concepts were investigated such as flap (partial trailing edge movement), boundary layer control, and spoilers, (leading edge partial surface deflection). Preliminary investigations indicated that the partial span torque control utilizing the hydraulic motor was the most effective from the standpoint of simplicity, power dissipation, and maturity of components and design concept. It consists of a hydraulic actuator mounted on a steel chord plate. A bearing separates and maintains structural rigidity between the two blade cord surfaces. A stainless steel bolt connects the two cords while also acting as the pivot axle.

SELECTED CONFIGURATION OF
PARTIAL SPAN TORQUE CONTROL



This version was configured so that it was compatible with the available space and the structure capabilities of the present MOD-1 blade. As in MOD-1 a hydraulic pitch change mechanism was chosen. The selection was governed mainly by several considerations:

1) The space requirements on the outer 15% of the blade was such that a hydraulic actuator is easily accommodated.
2) The hydraulic actuator reduces the blade outboard weight.
3) For emergency conditions, a hydraulic system with reservoirs placed in the blade can be activated in case of power failure or a failure of the control slip ring.

Controllability is the same or better than provided on MOD-1. The tip can be moved at a rate that more than compensates for the smaller control surface in order to produce comparable time rate of change of aerodynamic torque.

The hydraulic system layout is quite similar to MOD-1, except for a simplification due to the lower force requirement and mechanical trigger for emergency feather.

The inherent disadvantage of maintenance and repair of a hub mounted hydraulics was overcome by mounting all hydraulic components (except the tip located actuator) in a compact arrangement on the outer diameter of the hub so that it can be reached from the maintenance/assembly platform. The actuator is mounted on the outside (pressure side) of the blade and can be reached with a "cherry picker."

The direct weight savings of 38,000 pounds is attributed to the partial span Torque control.

PARTIAL SPAN TORQUE CONTROL

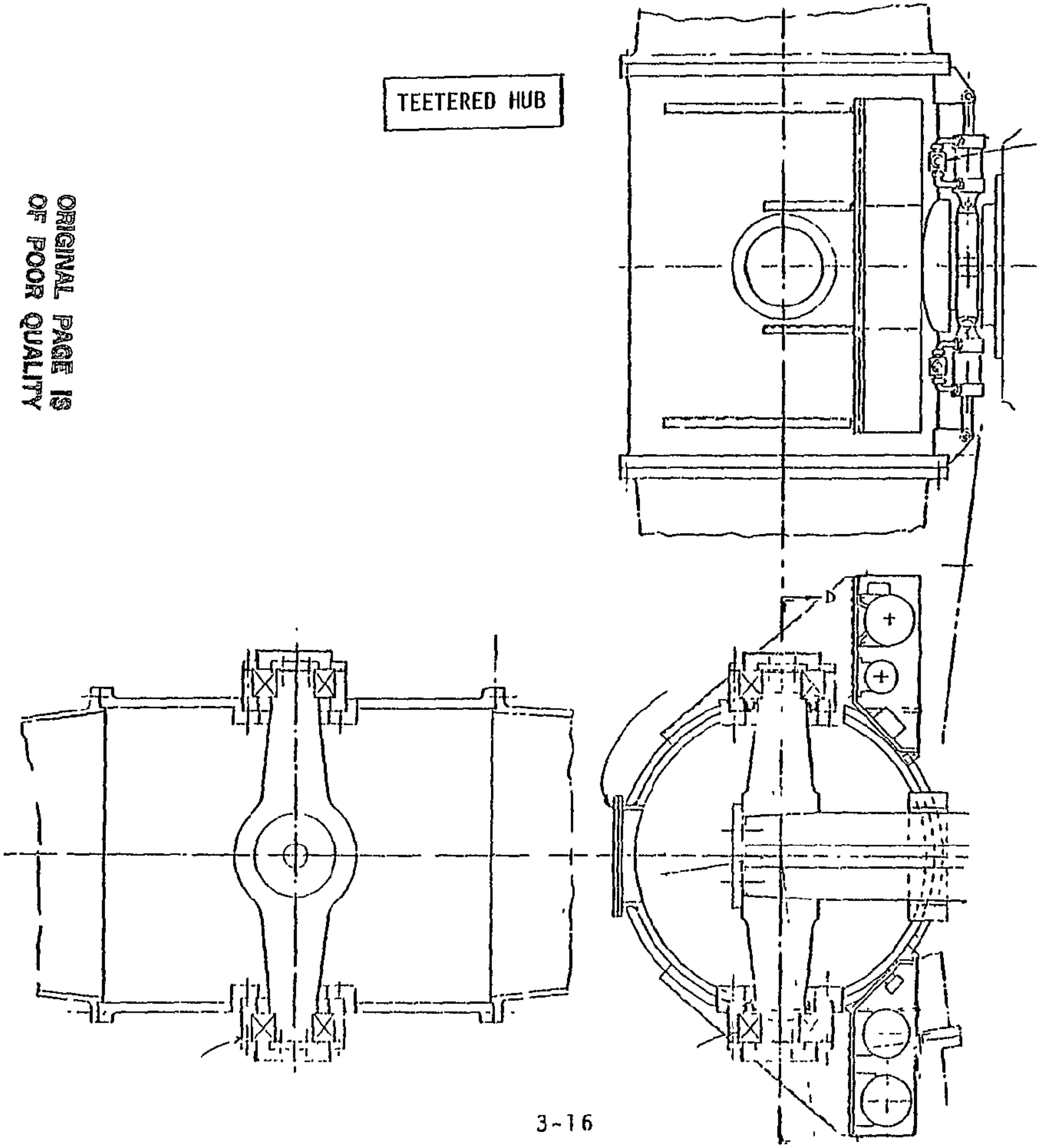
- 4 CONCEPTS INVESTIGATED.
- CHOSEN APPROACH IS READILY ADAPTABLE TO PRESENT MOD-1 BLADE.
- 15% OUTBOARD SPAN AND PITCH RATE PROVIDES SAME TORQUE CONTROL AS MOD-1 DESIGN.
- PRELIMINARY LOAD CALCULATIONS INDICATE EXISTING BLADE STRUCTURE ADEQUATE.
- HYDRAULIC ACTUATOR SYSTEM - SIMILAR TO MOD-1 CONCEPT.
  - MINIMIZES BLADE OUTBOARD WEIGHT
  - PROVIDES HIGH PITCH RATES FOR GUST CONTROL & EMERGENCY FEATHER
  - SAFE SHUTDOWN FOR POWER AND SLIPRING FAILURES
- PIVOT POINT POSITION SELECTED FOR SAFE SHUTDOWN.
- UNSYMMETRIC FAILURE NO PROBLEM.
- EASY ACCESS TO ACTUATOR & HYDRAULIC SYSTFM.
- WEIGHT REDUCED FROM 42,000 TO 4,000 LBS.

PARTIAL SPAN CONTROL CONTRIBUTES
TO MAJOR SYSTEM COST & WEIGHT REDUCTION

3.3 TEETERED HUB

The hub is a significant departure from the MOD-1 design as it incorporates a teetered blade attachment. In weight and manufacturing cost, however, it is largely simplified compared to the MOD-1 design. This stems from the fact that the large root flap moments from the blade do not have to be carried into the shaft and therefore the horizontal part of the hub, or tail shaft is largely reduced in size and cost. The teetering is provided by one cylindrical and one tapered (fixed) roller bearing. This type of bearing is used in the MOD-1 pitch change mechanism and has the advantage of being essentially maintenance free.

TEETERED HUB



In order to avoid a tower strike by the blades through extreme gusts or unsymmetric failure of the partial span pitch control, an end stop is provided at the hub limiting teeter travel. The tail shaft is designed so that additional moments during these extreme conditions can be tolerated without damage to the shaft. The hub barrel is a simple cylindrical weldment. The connection between teeter axis and main shaft is a conical shrink fit with provisions for hydraulic disassembly. This type of connection is extensively used for heavy machinery and was also used for the MOD-1 main shaft and coupling connections.

TEETERED HUB

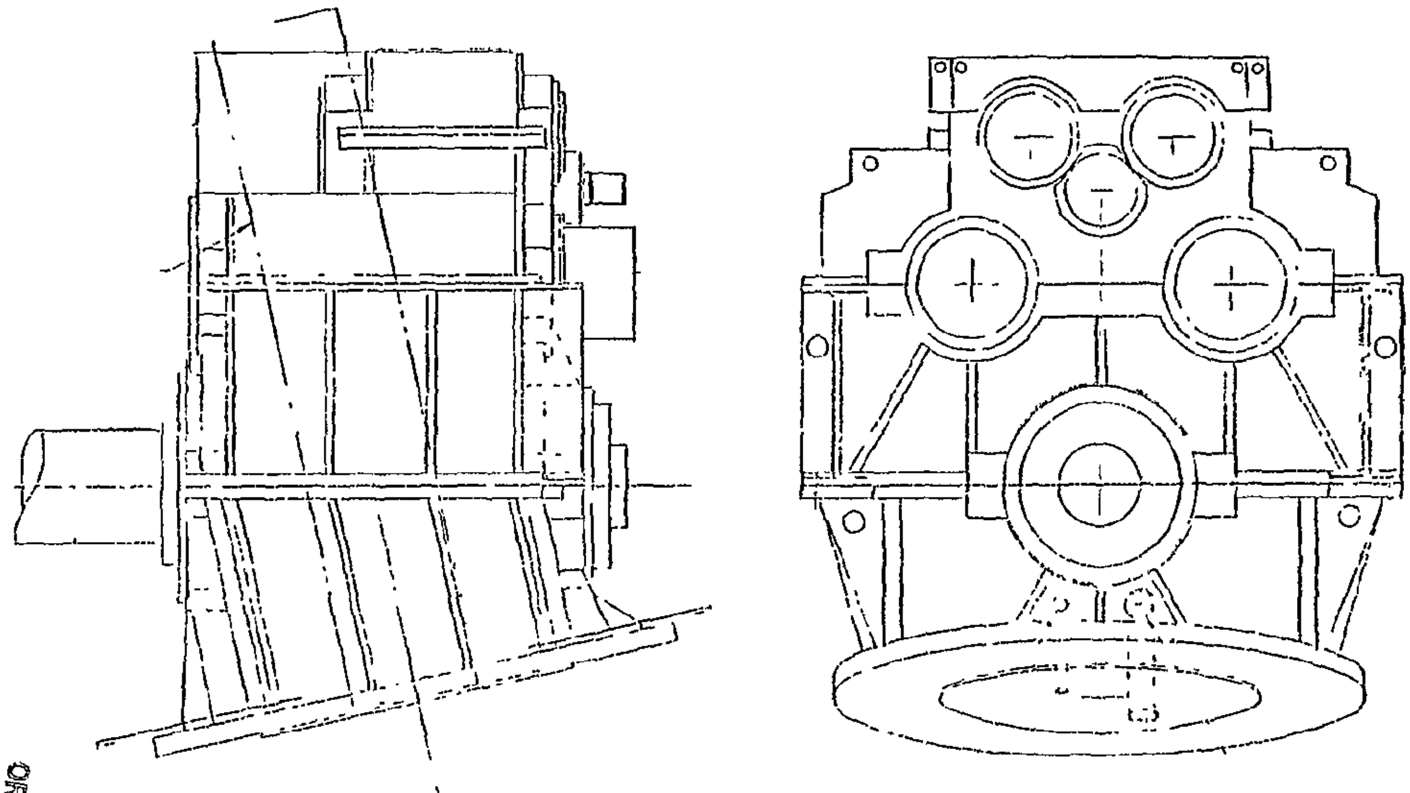
- MINIMIZES BLADE STRESSES - LOWEST LOADS FOR 2 BLADE SYSTEM.
- $\approx$ 2 DEGREES - NORMAL OPERATING RANGE
- HUB BARREL ROLLED STEEL WELDMENT WITH REINFORCEMENTS AT SHAFT PENETRATION AND BEARING MOUNTING.
- TAPERED AND CYLINDRICAL ROLLER BEARING AS TEETER SUPPORT.
- CONICAL SHRINK FIT AS INTERFACE TO MAIN SHAFT.
- WEIGHT REDUCED FROM 41,200 LBS. TO 17,300 LBS.

TEETERED HUB PROVIDES BEST
LOAD ALLEVIATION FOR
TWO BLADED ROTOR

3.4 GEARBOX/BEDPLATE

Tradeoff studies involving epicyclic and standard three stage parallel shaft gear systems were performed on the selection of the type of gearbox. Even though an epicyclic gearbox reduces weight by approximately 20,000 lbs, the cost based on production of 100 units is approximately $80,000 higher. Furthermore, the conventional parallel shaft gear-box lends itself better to a rotor shaft integration. Therefore, economic reasons and overall system weight prompted the selection of a MOD-1 type gearbox.

GEARBOX/BEDPLATE



Basic design of the gearbox is identical with existing MOD-1 hardware, except for the minor modifications:

- Increased length and diameter of 1st stage shaft.
- The lower part of the first stage housing accommodates the yaw bearing where the rotor forces are reacted into the tower structure.
- The gearbox lube pump is integral with the gearbox.

GEAR BOX

- 3 STAGE PARALLEL SHAFT GEAR BOX NOT LIGHTEST (+20K LB)
  BUT LOWEST COST (- 80K $)
- INTEGRATION OF INPUT SHAFT WITH ROTOR SUPPORT SHAFT ELIMINATES
  SEPARATE ROTOR BEARINGS, LOW SPEED SHAFT & COUPLINGS
- LOWER PORTION OF HOUSING SERVES AS YAW BEARING SUPPORT STRUCTURE
- WEIGHT OF GEAR BOX, BEDPLATE & NACELLE STRUCTURE REDUCED FROM
  147,300LBS TO 82,600 LBS

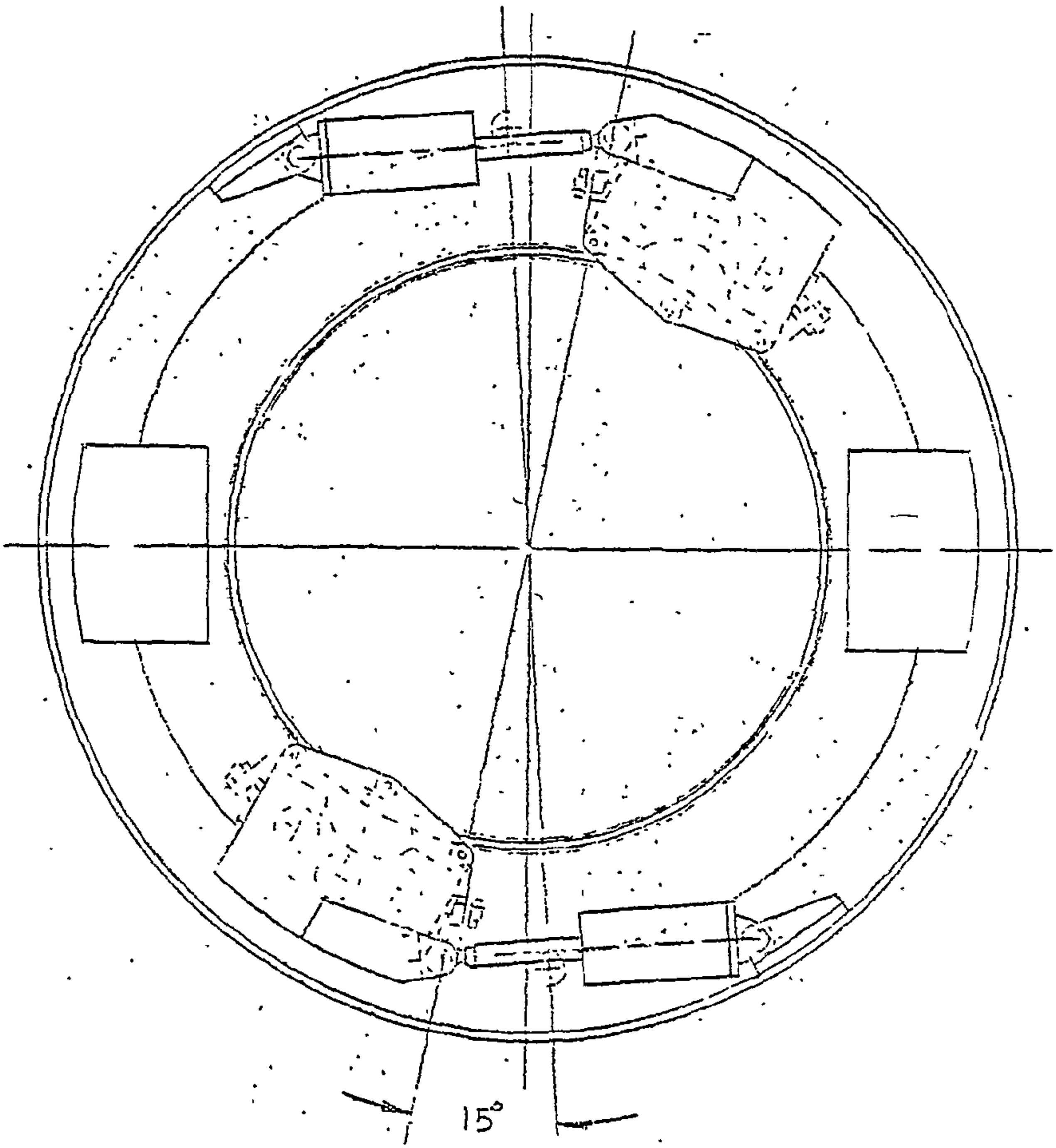
INTEGRATION OF ROTOR & YAW
SUPPORT INTO GEAR BOX
ELIMINATES NACELLE STRUCTURE

3.5 YAW SUPPORT AND DRIVE

The reduced yaw moments due to teetering and the reduced distance between hub and tower centerline allows a significantly simpler and more cost effective yaw drive system.

Two hydraulic cylinders, attached on one end to the stationary tower structure, on the other end to a brake, are used as the yaw drive motor.

YAW SUPPORT AND DRIVE



The actuators are parked to provide a yaw capability for one stroke which will be sufficient for almost all normal operating yaw maneuvers. After the execution of a yaw maneuver, holding brakes will be locked. The yaw actuators are placed again in position for later yaw corrections when required.

The hydraulic actuators and holding brakes are located inside the tower shell for environmental protection. The hydraulic system is also attached to the stationary part of the tower either at the tower base or on the yaw drive service platform on the top inside the shell. The reduced loads result in a significantly smaller yaw bearing diameter (6 ft. vs. 12 ft. for MOD-1). Even though the bearing loads would allow a further reduction in bearing diameter (6 ft. vs. 12 ft. for MOD-1). Even though the bearing loads would allow a further reduction in bearing diameter, the brake disc for the yaw drive system and the tower bending and torsional stresses as well as access for maintenance, dictate a minimum tower diameter of 6 feet at the yaw bearing interface.

YAW SUPPORT & DRIVE SYSTEM

- GEARBOX MOUNTED YAW BEARING AND SHELL TOWER ELIMINATE SEPARATE SUPPORT STRUCTURE
- HYDRAULIC ACTUATOR COMBINED WITH BRAKE REPLACES HYDRAULIC MOTOR/GEAR DRIVE
- REDUCED LOADS ALLOWS BEARING DIAMETER REDUCTION TO 6 FT.
- YAW BEARING SIMILAR TO MOD-1.
- SHELL TOWER PROVIDES ENVIRONMENTAL PROTECTION
- WEIGHT REDUCED FROM 51,300 LBS TO 7,800LBS.

SUBSTANTIAL SAVINGS THROUGH
ELIMINATION OF DEDICATED
YAW BEARING SUPPORT STRUCTURES

3.6 POWER GENERATION, INTERCONNECTION EQUIPMENT AND CONTROLS

The electrical system is substantially the same as for MOD-1 with some changes as follows:

- The generator is provided with a WPII frame suitable for outdoor application and resistance grounding is eliminated.
- A cable twist configuration instead of a slip ring will accommodate the yaw rotation.
- A single auxiliary power supply transformer at 208Y120 is sufficient to handle the reduced auxiliary power requirements.
- The control enclosure size will be reduced, and the base of the shell tower will be considered as control enclosure
- Reduction in mechanical complexity permits elimination of several motor starters and feeders.

Elimination of the engineering data system from production units is presumed, although the first unit of a new design would probably have an engineering data system. The operational control system is the same as on MOD-1 but with reduced hardware costs and less human interface data availability. For production systems, elimination of some sensors, utilization of a micro-processor, reduced memory requirements for the executive system and operator interface programs permit reduction in recurring costs.

POWER GENERATION & CONTROLS

- FOLLOWS IDENTICAL APPROACH AS MOD-1.
- GENERATOR HOUSING WITH WP II FRAME FOR ENVIRONMENTAL PROTECTION.
- CABLE TWIST INSTEAD OF YAW SLIPRING.
- REDUCED AUXILIARY POWER REQUIREMENT ALLOWS ELIMINATION OF ONE AUXILIARY POWER TRANSFORMER AND SEVERAL MOTOR STARTERS.
- SMALLER CONTROL ENCLOSURE - SHELL TOWER AS CONTROL ENCLOSURE CONSIDERED
- REDUCED SUPPORT EQUIPMENT ALOFT.
- REDUCED NUMBER OF OPERATIONAL CONTROL SENSORS.
- ALL EXISTING MOD-1 HARDWARE USABLE.

SAME CONTROL SYSTEM ELIMINATES MAJOR PORTION OF SOFTWARE DEVELOPMENT

3.7 TOWER

Three types of towers were investigated:

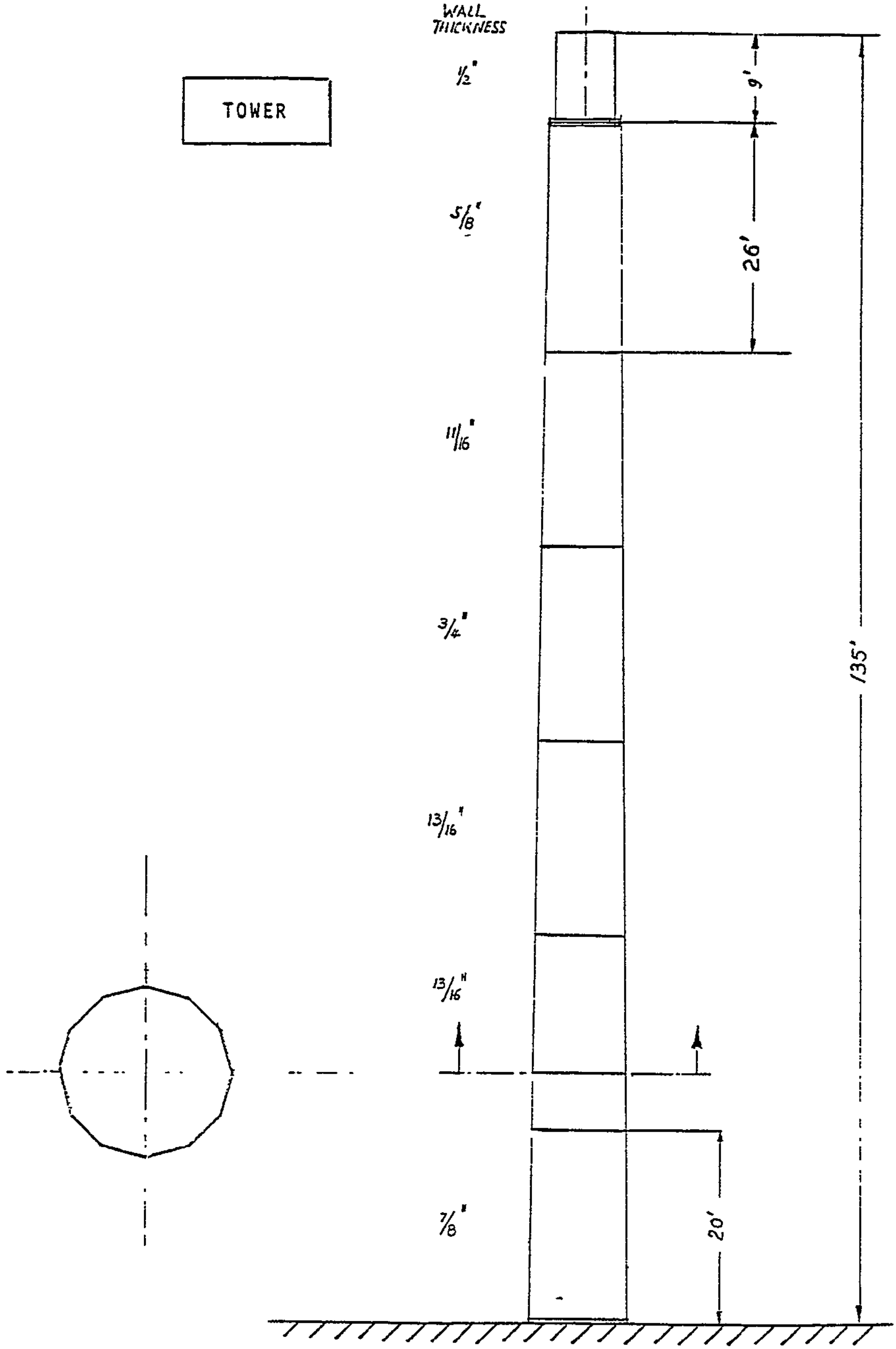
- Truss type -- similar to MOD-1
- Cylindrical shell with conical base
- Conical shell

Considering overall cost (including erection) the conical shell was selected for the following reasons:

- Reduced yaw bearing diameters without significant weight penalties in the tower.
- Simplified erection
- Improved aesthetic appearance.

The selected configuration has a top diameter of 72 inches and a base diameter of 100 inches.

The top section (approximately 10 ft.) is connected to the remaining portion and provides a convenient mounting and assembly base. This section also houses the interior work and hydraulic equipment platform and the access door to the nacelle platform.


TOWER
WALL THICKNESS
1/2"
5/8"
11/16"
3/4"
13/16"
13/16"
7/8"
9'
26'
135'
20'

Because of the lower cyclic stresses, a considerable lower bending stiffness was selected, resulting in a lateral bending frequency of approximately 1.2 cy/rev (P). For the preliminary design three materials were considered:

- A-588 (Cor-ten)
- A-36 with paint finish
- A-J72 with paint finish

Final selection will be made considering low temperature notch toughness and FAA painting requirements. Unlike MOD-1, where stiffness and fatigue requirements were structural design drives, the maximum base bending moment caused by the rotor thrust is pacing the tower design. Since rotor thrust for equal power level is almost independent of rotor configuration (teetered, 3 blades, etc.). This will become a basic design drive for horizontal axis wind turbine generators.

TOWER

- CONICAL SHELL - LATERAL BENDING FREQUENCY $\approx$ 1.2P
- FULL PENETRATION CIRCUMFERENTIAL WELDS (C or B)
- BASE CONSIDERED AS CONTROL ENCLOSURE
- TOP SECTION SERVES AS GROUND BASE FOR ASSEMBLY & OVERHAUL
- SHELL WEIGHT INCLUDING FLANGES 148,000 LB
- MAX. BENDING MOMENT FROM ROTOR THRUST BECOMES DESIGN DRIVER

ELIMINATION OF E-WELDS PROVIDES
INCREASE FATIGUE ALLOWABLES.
SHELL TOWER MINIMIZES ERECTION
COST & ENHANCES AESTHETIC APPEARANCE

3.8 TECHNICAL RISK ASSESEMENT

The overall program risk has been reduced by maintaining a significant portion of MOD-1 commonality throughout the program development. Those new systems requiring new designs/developments are identified as follows:

- Partial span torque control
- Teetered hub
- Yaw drive support
- Cylindrical conical tower (soft tower)

Partial Span Torque Control - The size and power required to drive the partial span torque control will fully provide for the necessary control of the wind turbine generator. Simulation has been performed which indicates that this is a practical approach in controlling the WTG within the same spectrum as the MOD-1. Design and development of the hardware, however, has not been proven, except by inference on aircraft structures.

Teetered Hub - Altough the teetered hub is a deviation from the fixed hub approach utilized on MOD-1, it represents a straightforward mechanical design. The teeter shaft (or axle) will utilize common steel fabrication practices such as shrink fit over the gear-box shaft. The teeter shaft bearings are off-the-shelf components and employ a standard interferface into the hub barrel/casement. Both the teeter hub and the hub/shaft interface processes are similar to the MOD-1.

Drive Train Dynamics with Short Low Speed Shaft - the Main drive train has been shortened significantly from the MOD-1 design reducing shaft flexibility to alleviate the transfer of the impact of wind gusts on the rotor.

Main Shaft Deflection/Gear Interference - Concept #3 incorporates the use of the gearbox to house the fore and aft main shaft bearings. Concern has been raised regarding the possibility of mainshaft bending causing off-axis rotation of the gears and the gear interface problems. Preliminary stress and bending analysis indicates that the selected gear-shaft diameter is adequate to overcome any potential problems.

Yaw Support and Drive System - The proposed hydraulic actuator and brake system is a significant departure from MOD-1. It is a straightforward mechanical design in the use of hydraulic actuator and brakes.

Soft Tower - The stiff tower design for MOD-1 was selected on the basis of confidence and demonstration of the MOD-0 approach. With more confidence in the ability to establish natural frequency parameters to major subsystems, and be able to "tune" major components, it has now been determined that a soft tower is a practical solution for a low-weight economic system. It is expected that once placed in the field during the checkout final frequency placement will be required by tuning.

TECHNICAL RISK ASSESSMENT

- PARTIAL SPAN TORQUE CONTROL DESIGN LARGEST SINGLE RISK
  - MINIMIZED BY TEST
  - GOOD ANALYTICAL SIMULATION REQUIRED
- TEETERED HUB - STRAIGHT FORWARD DESIGN
- HUB/SHAFT INTERFACE - STANDARD PRACTICE USED ON MOD-1
- DRIVE TRAIN DYNAMICS WITH SHORT LOW SPEED SHAFT
  - NO PROBLEM EXPECTED BUT CAREFUL SIMULATION REQUIRED
- MAIN SHAFT DEFLECTION/GEAR INTERFERANCE
  - PRELIMINARY NUMBERS INDICATE NO PROBLEM
  - INCREASE OF SHAFT DIAMETER INCONSEQUENTIAL
- NEW YAW DRIVE CONCEPT
  - FALL BACK TO GEAR DRIVE ON BRAKE DISC
  - 23% OF MOD-1 TORQUE
- SOFT TOWER
  - MAY REQUIRE TUNING
  - CAREFUL FREQUENCY PLACEMENT REQUIRED

NO MAJOR TECHNICAL
RISK FORSEEN

3.9 SECOND UNIT CONCEPT #3 WEIGHT AND COST COMPARISONS TO STUDY GOALS AND MOD-1

The final weights and Second unit costs of Concept #3 are shown in the cross-hatched blocks and are compared to the study goals and MOD-1 system, broken down by subsystem, assembly/test and erection. These original goals, shown in Section 2.2, were not changed throughout the conduct of the study. The early goals were derived somewhat arbitrarily, with the exception of the hub, torque control and total weight at the top of the tower as well as the tower weight. Significant improvements in yaw drive system, tower weights and site preparation, erection and check-out contribute to off-set the early optimistic power generation weights and cost goals, which represent a mature subsystem.

2ND UNIT WEIGHT & COST COMPARISONS

SUBSYSTEM	2ND UNIT MOD 1-A BOGIES COST $K	2ND UNIT MOD 1-A BOGIES WEIGHT K LB	CONCEPT 3 COST $K	CONCEPT 3 WEIGHT K LB	2ND UNIT MOD-1 COST $K	2ND UNIT MOD-1 WEIGHT K LB
BLADES	323	22	308	38.5	280	36.0
HUB	129	15	150	17.3	341	41.2
TORQUE CONTROL	65	8	19	4.6	161	42.6
NACELLE/STRUCTURE & DRIVE TRAIN	291	95	287	82.6	624	147.3
POWER GENERATION EQUIPMENT	178	46	270	68.7	290	70.1
CONTROLS	97	6	152	7.5	173	8.1
YAW DRIVE SYSTEM	112	33	71	7.8	268	51.2
TOWER	194	175	123	148	360	352.6
ASS'Y & TEST	258	-	456		587	-
SITE PREP, ERECT & CHECK-OUT	355	-	242		616	-
TOTAL	2,002	400	2,078	320	3,700	749

- REDUCE LOADS
- ELIMINATE FUNCTION, SIMPLIFY COMPONENTS
- INTEGRATE FUNCTIONS
- SIMPLIFY ASS'Y & ERECTION

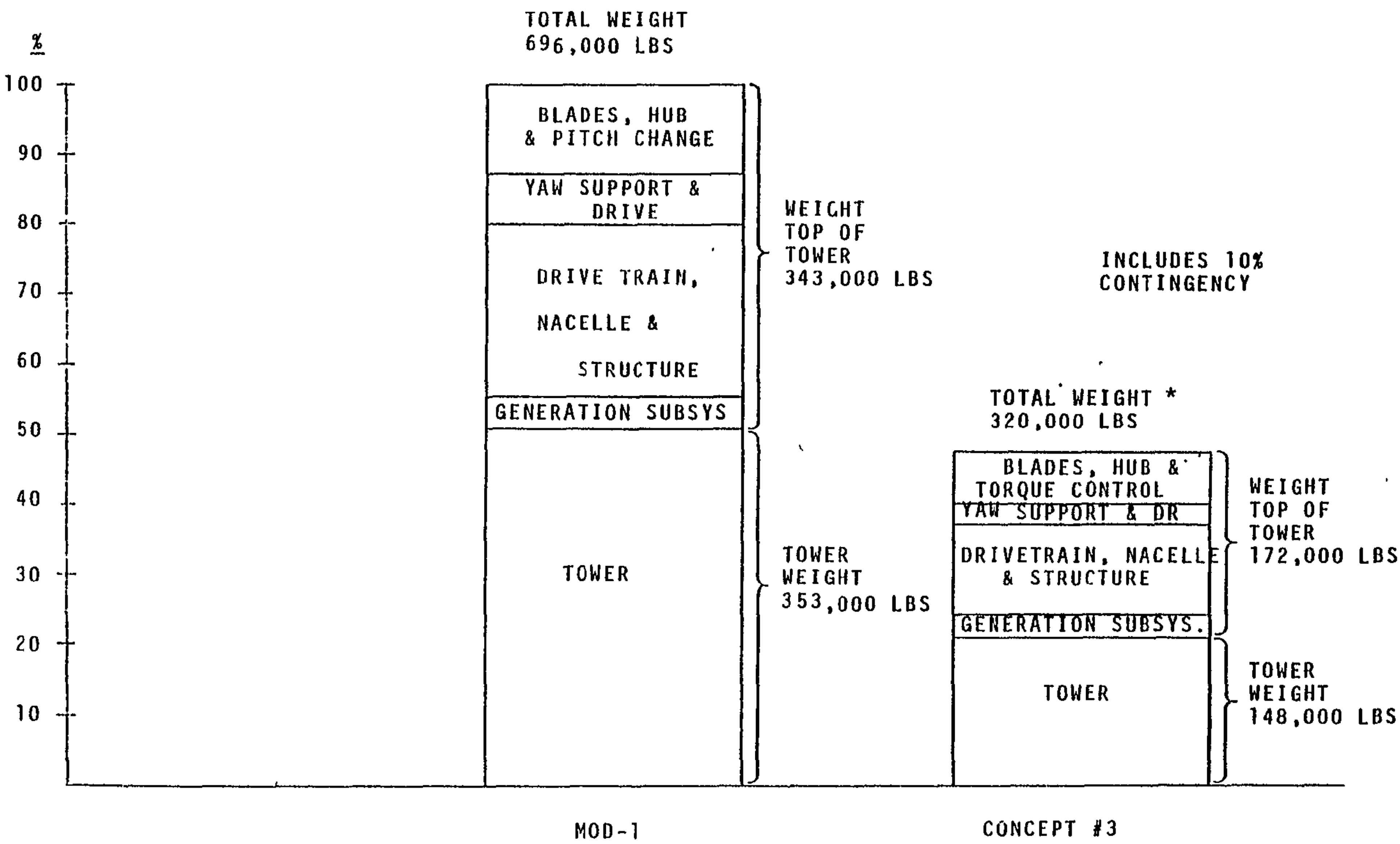
3.10 MOD-1 CONCEPT #3 WEIGHT COMPARISON

The table below provides weights of both MOD-1 and Concept #3 and an indication of where significant savings have been achieved.

SUBSYSTEM	Weight - LB(K)		Concept #3 % Of MOD-1
	MOD-1	Concept #3	
Blades, Hub and Torque Control	119.8	60.4	50
Yaw Support and Drive	51.3	7.8	15
Drive Train, Nacelle and Structure	147.3	82.6	56
Power Generation (and Controls)	24.6	21.6	87
Subtotal (Weight-top of Tower)	343.0	172.0	50
Tower	353.0	148.0	42
Total (Weight - Base of Tower)	696.0	320.0	46

Each major subsystem, with the exception of power generation and controls contributes significantly to the overall weight reduction. Electrical power generation equipment is well within the state-of-the-art and is represented by a high degree of off-the-shelf, proven hardware. It is understandable then that little impact on overall weight can be made in power generation and controls. The major contributions in weight reduction are attributable to the reduction in loads, reduction in sizes of torque control yaw support and drive, the elimination of the bed-plate, the simplification of the support structure, and the soft tower. These all combine to provide an overall total weight that is only 46% of the entire MOD-1 system.

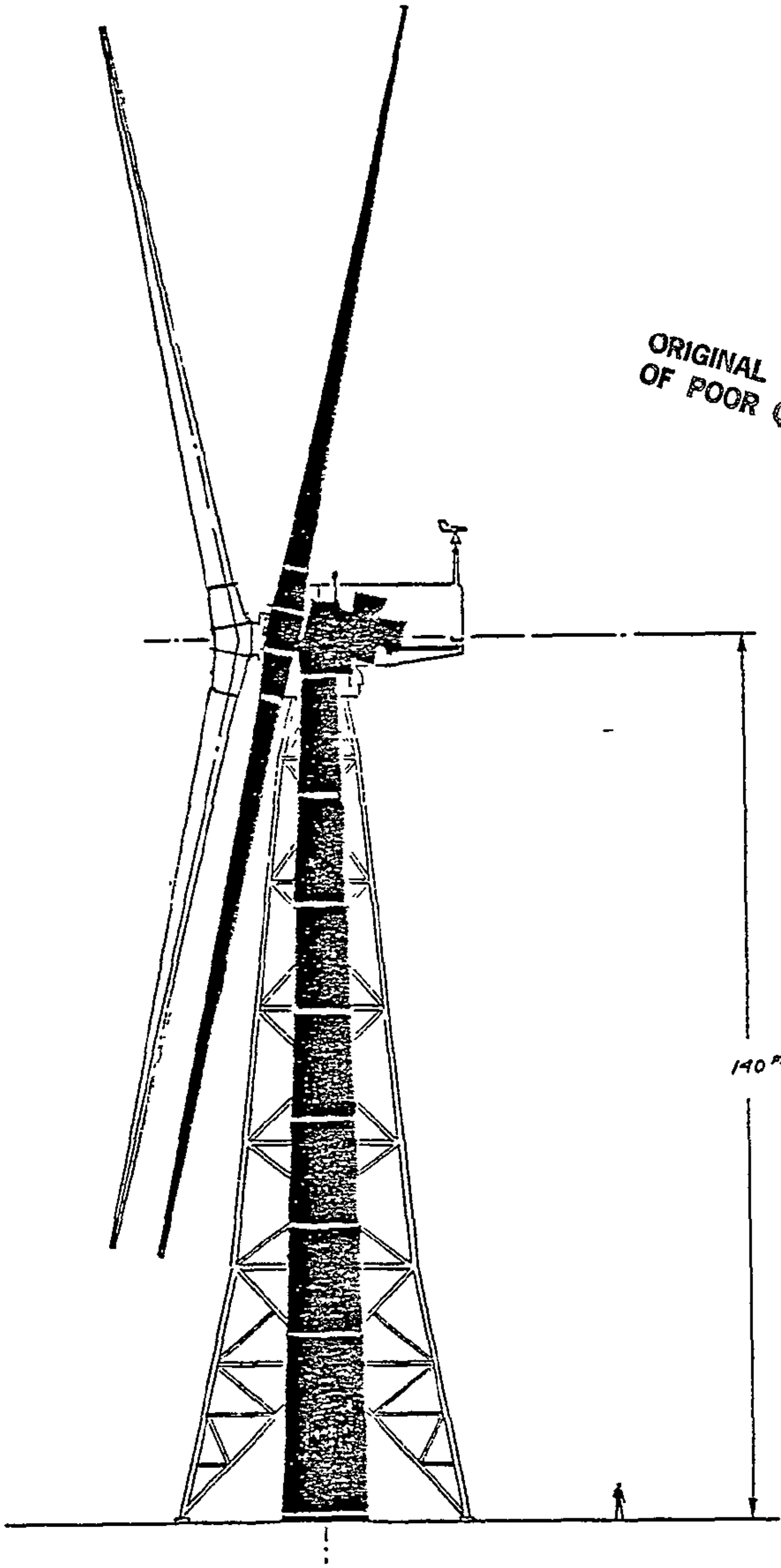
MOD-1/CONCEPT #3 - WEIGHT COMPARISON



3.11 MOD-1 CONCEPT #3 COMPARISON

This sketch provides a silhoutte of Concept #3 superimposed over the MOD-1 and shows the relative size advantages. It provdes a striking comparison of physical sizes.

MOD-1/CONCEPT #3 COMPARISON



4.0 SYSTEM ECONOMICS AND PERFORMANCE

Second unit system costs were developed in (1977 dollars) for comparison to MOD-1 and to establish the base capital cost to compute the cost of energy. This data is provided in this section.

SYSTEM ECONOMICS & PERFORMANCE

2ND UNIT COSTS
POWER OUTPUT
ANNUAL ENERGY CAPTURE
COST OF ENERGY

4.1 SECOND UNIT COST AND COMPARISONS

This table provides weighting and costing data for both MOD-1 and Concept #3 on a subsystem by subsystem comparative basis. Although we achieved our goal of less than 400,000 pounds we did not quite achieve the goal of $1000 per kilowatt. It is likely, however, that future improvements from this generation system could easily achieve that goal.

MOD-1 & CONCEPT #3 –
2ND UNIT WEIGHT & COST COMPARISONS
(CONSTANT 1977 $)

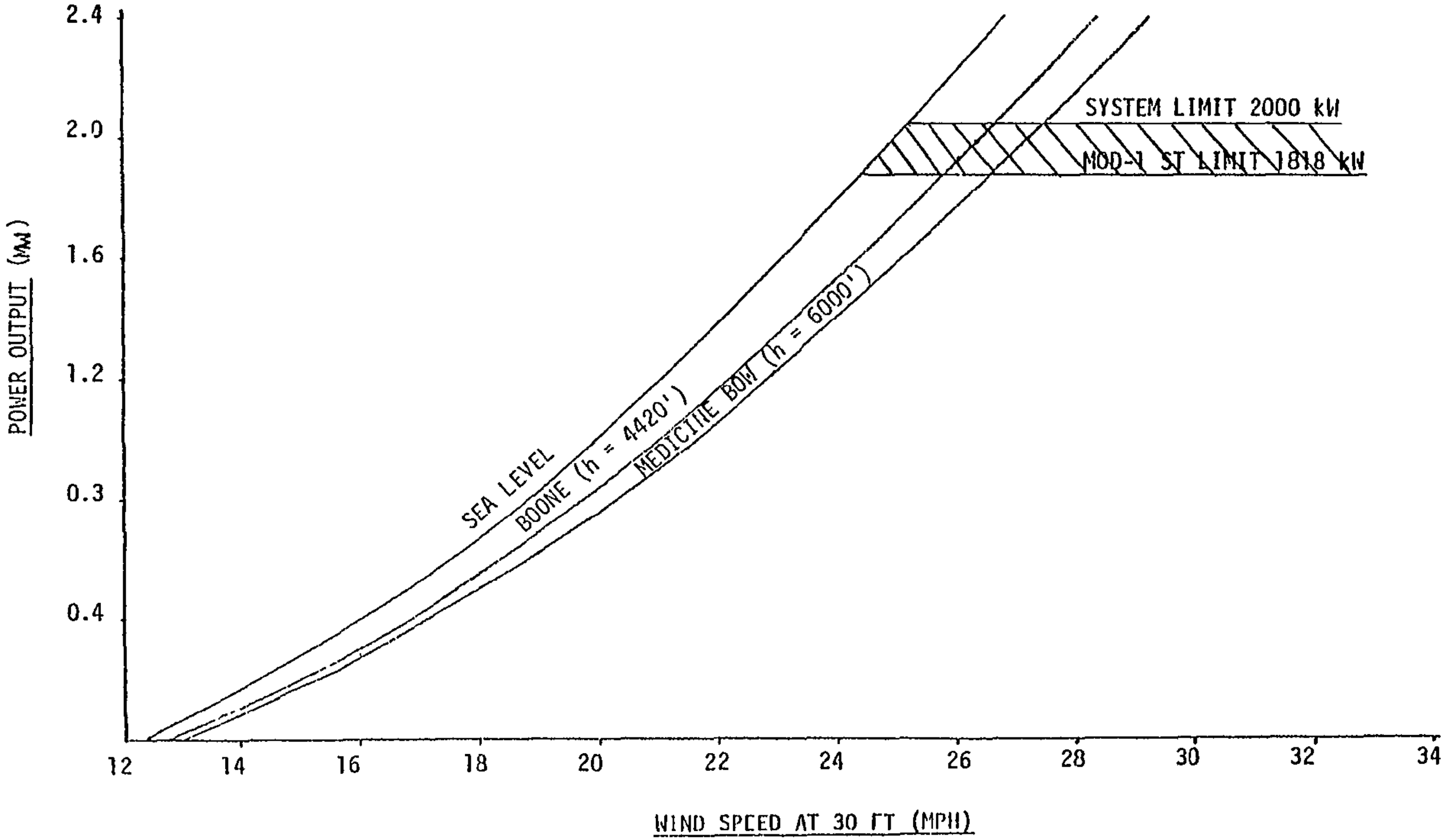
	MOD-1			CONCEPT #3		
	(K)		$/LB	(K)		$/LB
	LBS	$		LBS*	$	
BLADES	36.0	250	6.95	38.5	274	7.10
HUB	41.2	264	6.40	17.3	118	6.80
TORQUE CONTROL	42.6	126	2.95	4.6	15	3.25
BEARING & DRIVETRAIN	73.4	240	3.25	76.5	190	2.48
NACELLE/STRUCTURE	73.9	246	3.45	6.1	33	5.40
POWER GENERATION EQPT.	70.1	230	3.30	68.7	214	3.11
CONTROLS	8.1	135	16.65	7.5	120	16.00
YAW DRIVE SYSTEM	51.3	209	4.10	7.8	56	7.20
TOWER	352.7	321	.91	148.0	110	.75
SUBTOTAL	749.3	2022	2.70	375.0	1130	3.00
SITE PREPARATION/WTG INSTALL.	-	547	-	-	215	-
VF LABOR/OVERHEAD	-	437	-	-	340	-
T&L/COMPUTER	-	20	-	-	20	-
SUBTOTAL		3026	-		1705	-
G&A		432	-		237	-
FEE		242	-		136	-
TOTAL WTG - INSTALLED	749.3	3700	4.95		2078	5.55

* INCLUDES 10% CONTINGENCY

4.2 POWER OUTPUT

Because of the load reductions on the blades, there is no limitation in rated capacity as is the case with the MOD-1 system. Therefore, the hatched area on the attached curve represents the increased power output that can be achieved at the various wind speeds at the three different locations: MOD-1 specifications (sea level), MOD-1 installation, (Boone, N.C.) and a high wind regime.

POWER OUTPUT

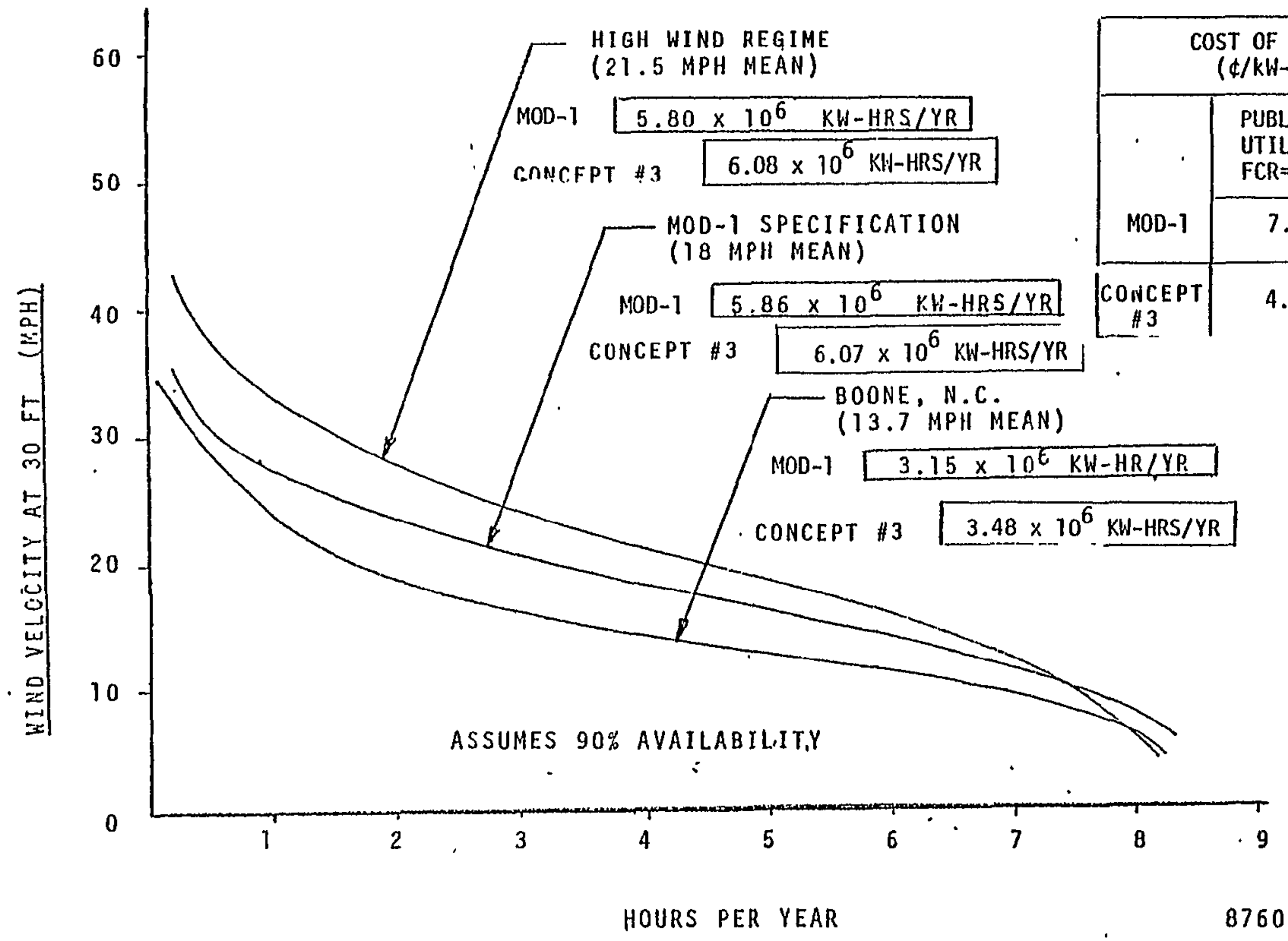


4.3 ANNUAL ENERGY CAPTURE COMPARISON

Using the wind duration curves of the sites previously mentioned, both MOD-1 and Concept #3 energy capture were computed and are compared. Depending upon wind availability total annual energy output increase ranges between 3 - 10%. This is due to the Concept #3 rated capacity of 2000 KW versus 1818 KW, which is a blade structural limitation on the MOD-1 system.

The cost of energy of Concept #3, depending upon investor owned or public owned category, varies between 4.2¢ and 6.3¢ per kilowatt hour as compared to just under 7.5¢ and 11.3¢ for the MOD-1. The cost of energy calculation is very encouraging and represents a potential system that can be marketed competitively with either investor owned or public owned utilities, in selective geographical areas in the next 3 - 5 years.

ANNUAL ENER CAPTURE COMPARISON



COST OF ENERGY (¢/kW-HR)	PUBLIC UTILITY FCR=.12	INVESTOR UTILITY FCR=.18
MOD-1	7.6	11.3
CONCEPT #3	4.2	6.3

APPENDIX G

ELECTRICAL STABILITY ANALYSIS

Abstract

This appendix is a report on the final phase of a series of analyses performed by G. E. - Electric Utility Systems Engineering. It includes the results of analysis conducted to determine the compatibility of the Mod 1 WTG with its selected installation site in the Blue Ridge Electrical Membership Co-operative.

INTRODUCTION

The first phase of this study provided a preliminary performance appraisal of a single wind generator connected to an infinite bus. During Phase II, dynamic performance of single and multiple wind generator units in an electric utility environment was investigated. This report of the third and final phase of the performance study presents the results of analysis conducted to determine the compatibility of the MOD-1 Wind Turbine-Generator with its selected installation site in the Blue Ridge Electric System. In analyzing this compatibility the Phase III study served a threefold purpose:

1. to examine dynamic stability for a variety of possible system conditions,

2. to assess transient response to severe wind gusts, and

3. to identify and pre-test analytically parameter adjustments which could improve performance.

The analysis discussed herein was performed using models developed previously and described in detail in the Phase I and Phase II reports. Hence dynamic models are not discussed in this report. However, a complete data set for the models used in the Phase III analysis has been included in the Appendix.

SUMMARY

This study was undertaken to assess the wind turbine-generator performance on the Blue Ridge Electric System. The study was conducted using digital dynamic simulation techniques to represent the wind generator and the 12 kV distribution system. Dynamic stability and transient response to wind gusts were investigated utilizing this approach.

Study results indicated that the system was dynamically unstable when operated at very low power output. It was found that this condition could be rectified by adjusting the tuned frequency of the notch filter in the power regulator. The results also indicated that transient response to wind gusts could be substantially improved by slightly increasing the transient gain of the power regulator.

1. Distribution System Representation

The installation site for the MOD-1 Wind Generator was chosen to be Boone, North Carolina, where it will be connected to a 12 kV distribution circuit of the Blue Ridge Electric Membership Corporation. Data describing the transmission system and significant load characteristics was provided by Mr. William Terry of Blue Ridge Electric. This data has been included in its original form in the appendix. From this data, a reduced order model of the distribution system was obtained for use in the analysis.

A one line diagram of the reduced order model is shown in Figure 1. Howard's Knob Circuit No. 211 was modeled in detail while the Sands Circuit No. 213 and Bamboo Circuit No. 212 were lumped into equivalent lines and loads at buses 4 and 5. Bus 3 corresponds to the Boone 12 kV bus. Bus 2 was modeled as an infinite bus representing the higher voltage transmission system. The Wind Generator was located at bus 1.

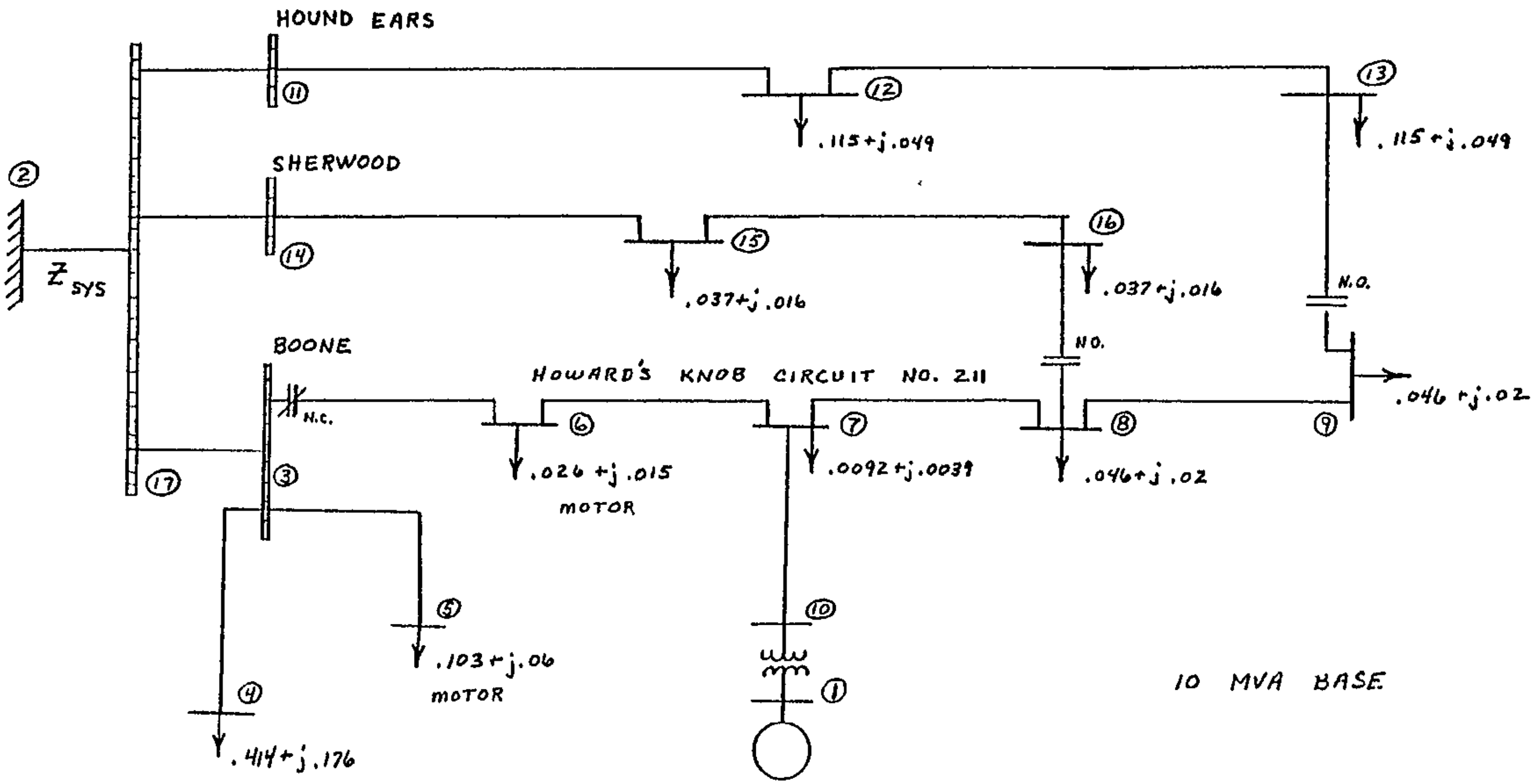
Load dynamics play an important role in the overall performance of the Wind Generator. The load models used to characterize this portion of the Blue Ridge system are summarized below:

Bus 5 — 1200 KVA induction motor load representing the water pumping station, hospital, sewage treatment plant, and lumber company.

Bus 6 — 300 KVA induction motor load representing the water filtration plant.

Buses 4,7,8,9	Constant current real power load and constant impedance reactive power load.
Buses 12,13,15,16	Constant impedance loads.

Howard's Knob Circuit No. 211 is normally connected to the Boone 12 kV Bus. Alternate modes of operation include connecting it through feeders to Sherwood or Hound Ears.



One line diagram of distribution system

Figure 1

2. Dynamic Stability Analysis

Dynamic stability of the Wind Generator System was assessed by calculating eigenvalues and plotting root migration as a function of four system parameters:

1. voltage regulator gain, K_A,
2. power regulator gain, K_P,
3. power system stabilizer gain, K_{STAB}, and
4. generated power, P.

Wind speed was assumed to be constant at 37 ft/sec.

With the power regulator and stabilizer gains set to zero and the generator operating at full output (1875 KW), root migration was plotted as a function of voltage regulator gain. Saturation effects were included in the excitation system representation. The gain was varied from zero to 975 and the resulting plot is shown in Figure 2A.

With the voltage regulator gain at 975 and stabilizer gain remaining at zero, the power regulator gain was varied from zero to its specified setpoint of 75. The root locus plot for this case is shown in Figure 2B. It can be seen that the power regulator gain had no effect on most system modes of oscillation. However, the only mode significantly affected was the rigid body mode at 2.6 rad/sec., and its damping ratio was decreased with increased power regulator gain.

Root migration as a function of power system stabilizer gain was also investigated. For this case, the voltage and power regulator gains were held at their specified setpoints and the stabilizer gain was varied from zero to 500. The plot, Figure 2C, shows that the damping ratios of the two most lightly damped modes, 2.6 and 76 rad/sec., were substantially improved with increased stabilizer gain.

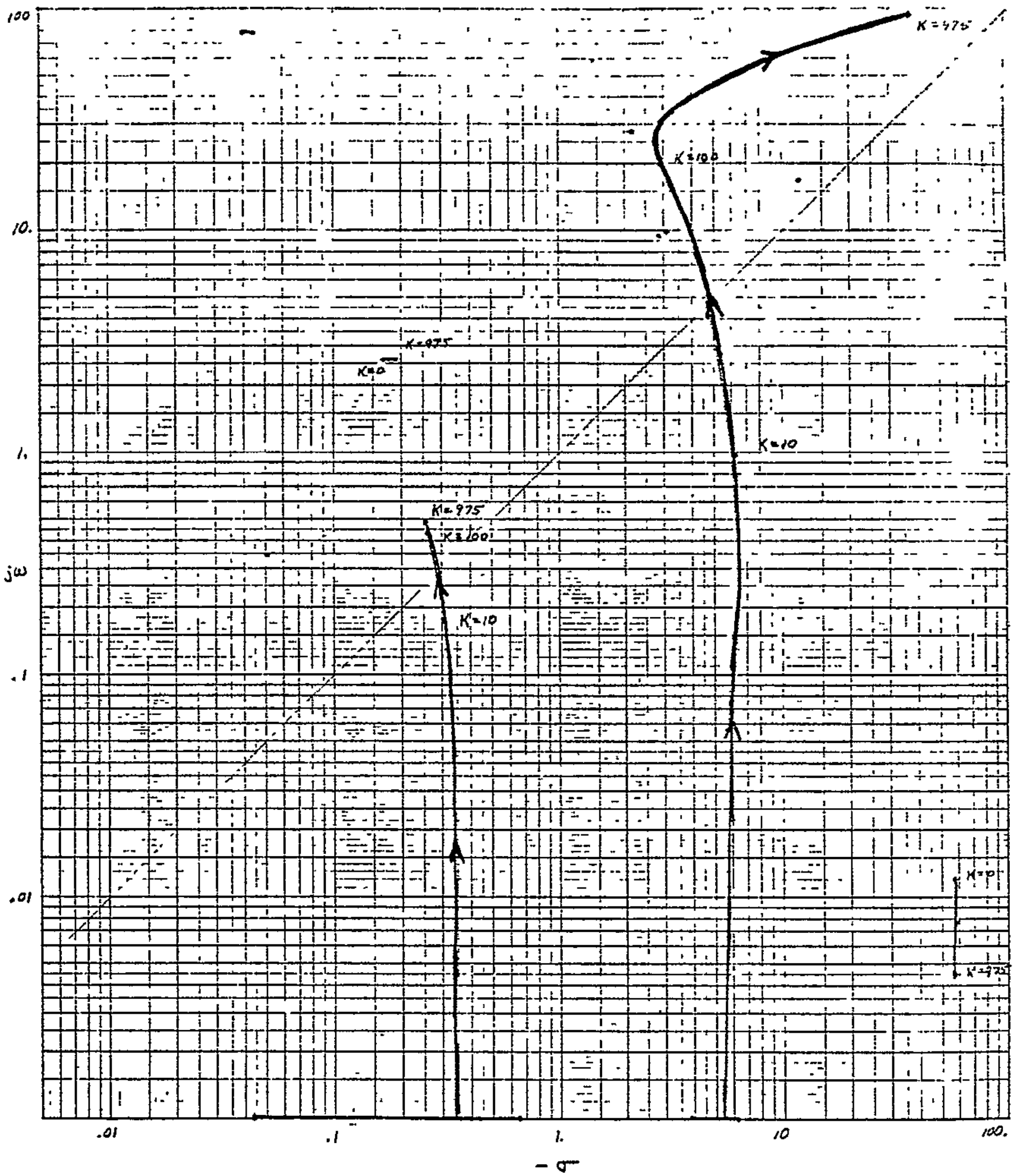
A root locus was also plotted as a function of generator power output. Power was varied from zero to 1875 KW with regulator and stabilizer gains set at their recommended values. The plot, shown in Figure 2D, indicates dynamic instability for the rigid body mode at zero power output. Other system roots were well behaved as power output was varied.

In order to gain an understanding of the mechanism of the instability, the transfer function from the blade pitch angle, β, to electrical power P_E, was calculated for operating points of zero KW and 1975 KW. The magnitude of this transfer function is plotted in Figure 2E and the phase angle is plotted in Figure 2F. At full power output, the rigid body mode natural frequency is at 2.6 r/s, as indicated by the resonant peak in magnitude at that frequency. At zero power output, the resonance occurs at 2.1 r/s, and the magnitude of the peak is greater than that at 2.6 r/s.

A Bode plot of the power regulator transfer function is shown in Figure 2G. At 2.6 r/s, the regulator operates with a net phase lag of 38 degrees. But at 2.1 r/s, the phase lag is 78 degrees suggesting that the power regulator is contributing to the instability at low power. In order to compensate for this, the notch filter in the power regulator was adjusted from 2.6 r/s to 2.1 r/s. Gain and phase plots of the adjusted regulator are shown by the dashed lines in Figures 2H and 2I. With the adjusted notch, the regulator operates with 38 degrees of phase lag at 2.1 r/s. Transient gain has also been increased, but gust response indicated that this was also a desirable effect. (The transient response studies are discussed in detail in Section 3 of this report.) The effectiveness of this method of compensation was tested by plotting a root locus as a function of generator power output, similar to a previous plot wherein the problem was identified. The root locus plot is shown in Figure 2J. Although the damping ratio of the rigid body mode decreases somewhat at low power, the root does remain stable for all power levels.

This analysis has shown that rigid body mode frequency varies as a function of power output and that appropriate filtering in the power regulator can eliminate instability caused by that effect. However, rigid body mode frequency is typically very strongly influenced by the stiffness or short circuit capacity of the ac transmission system.

In order to investigate WTG stability as a function of changes in the host distribution system, seven different system configurations were simulated, including conditions with weakened higher voltage transmission. Eigenvalues for the rigid body mode and the 12 Hz torsional mode are tabulated in Figure 2K for all these conditions. Z_{EQUIV} is the equivalent system impedance seen from the WTG terminals. Z_{SYS} is the equivalent impedance of the higher voltage transmission system and is identified in Figure 1. The results show that rigid body mode frequency changes very little with ac system condition. In all cases simulated, the frequency remained between 2.2 r/s and 2.75 r/s. Also, dynamic stability was achieved in all cases, even with a severely weakened ac system.



$K_p = 0$
$K_{STAB} = 0$
$P = 1875KW$

Root Locus Plot:
K_A varied from 0 to 975

Figure 2A

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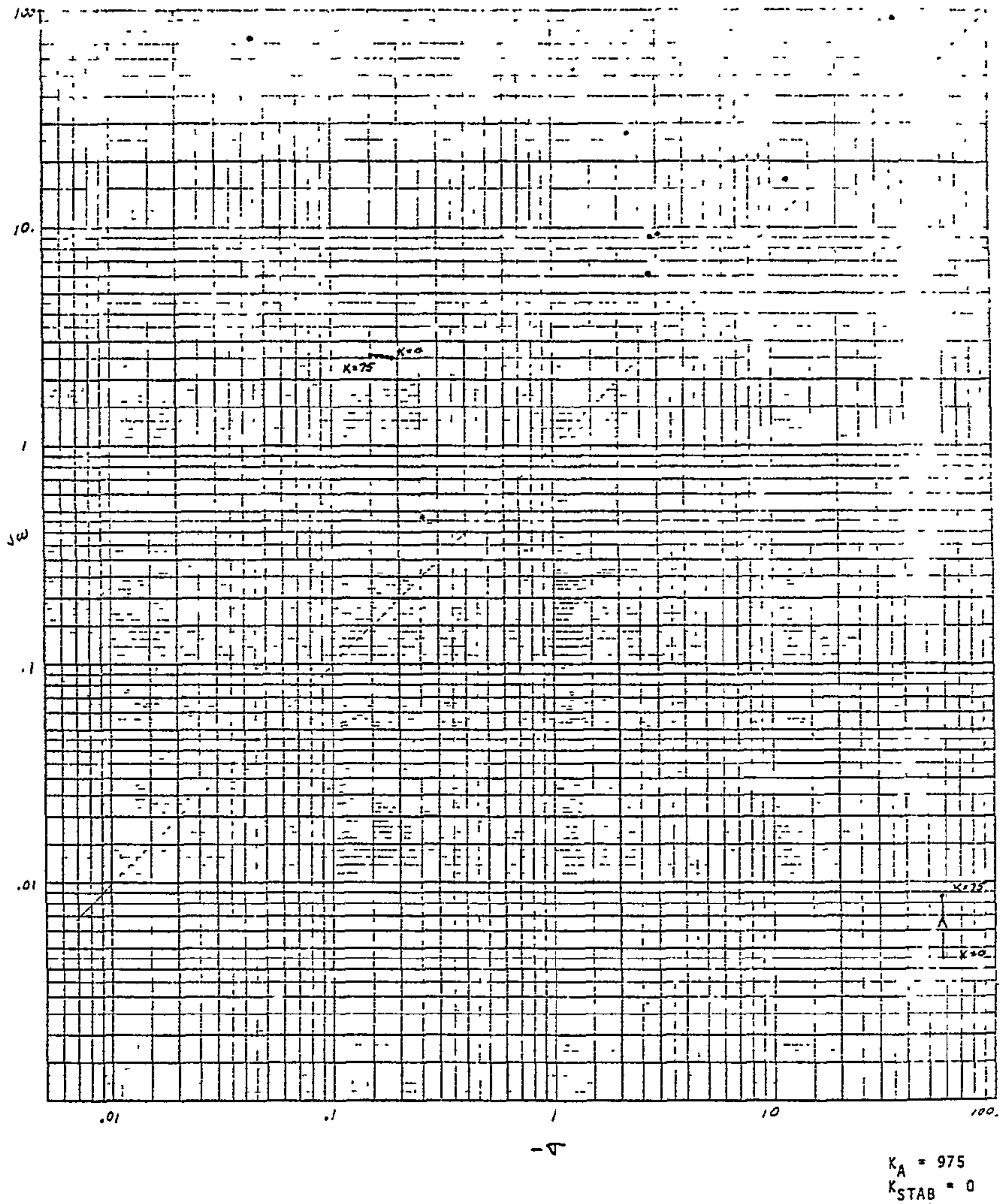
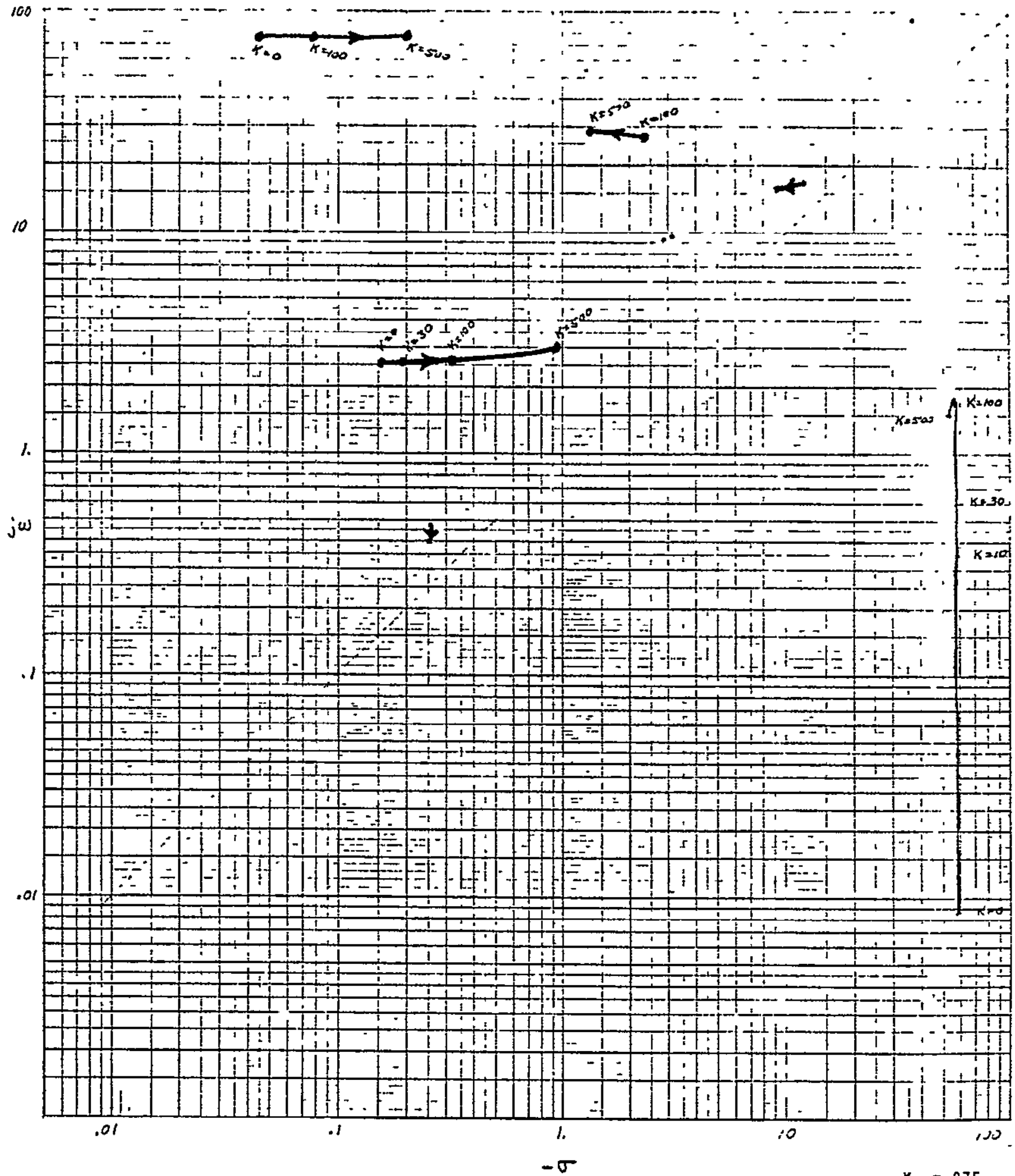


$K_A = 975$
$K_{STAB} = 0$
P = 1875KW

Root Locus Plot·
K_p varied from 0 to 75

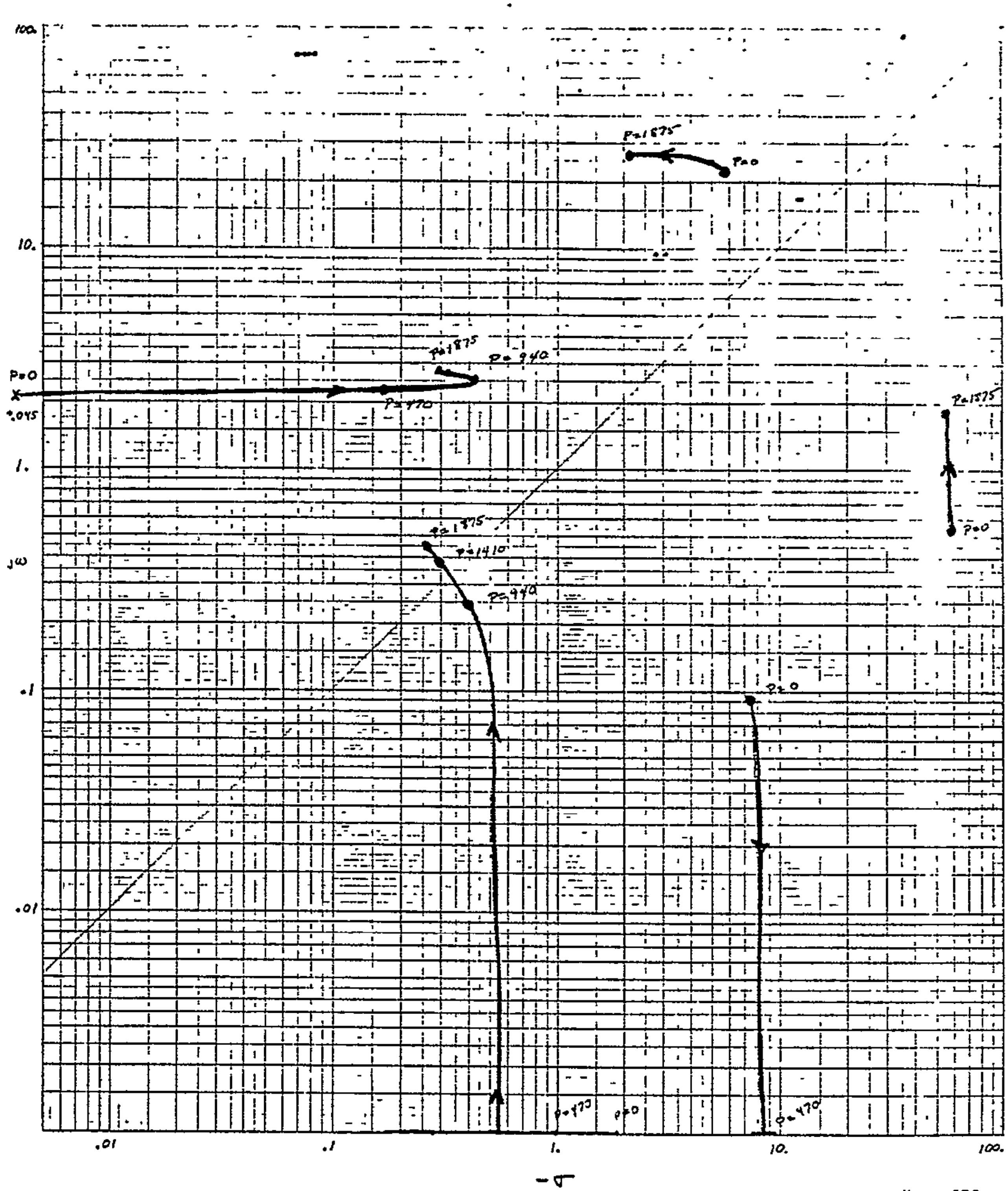
Figure 2B



K_A = 975
K_p = 75
P = 1875KW

Root Locus Plot:
K_{STAB} varied from 0 to 500

Figure 2C



K_A = 975
K_p = 75
K_{STAB} = 100

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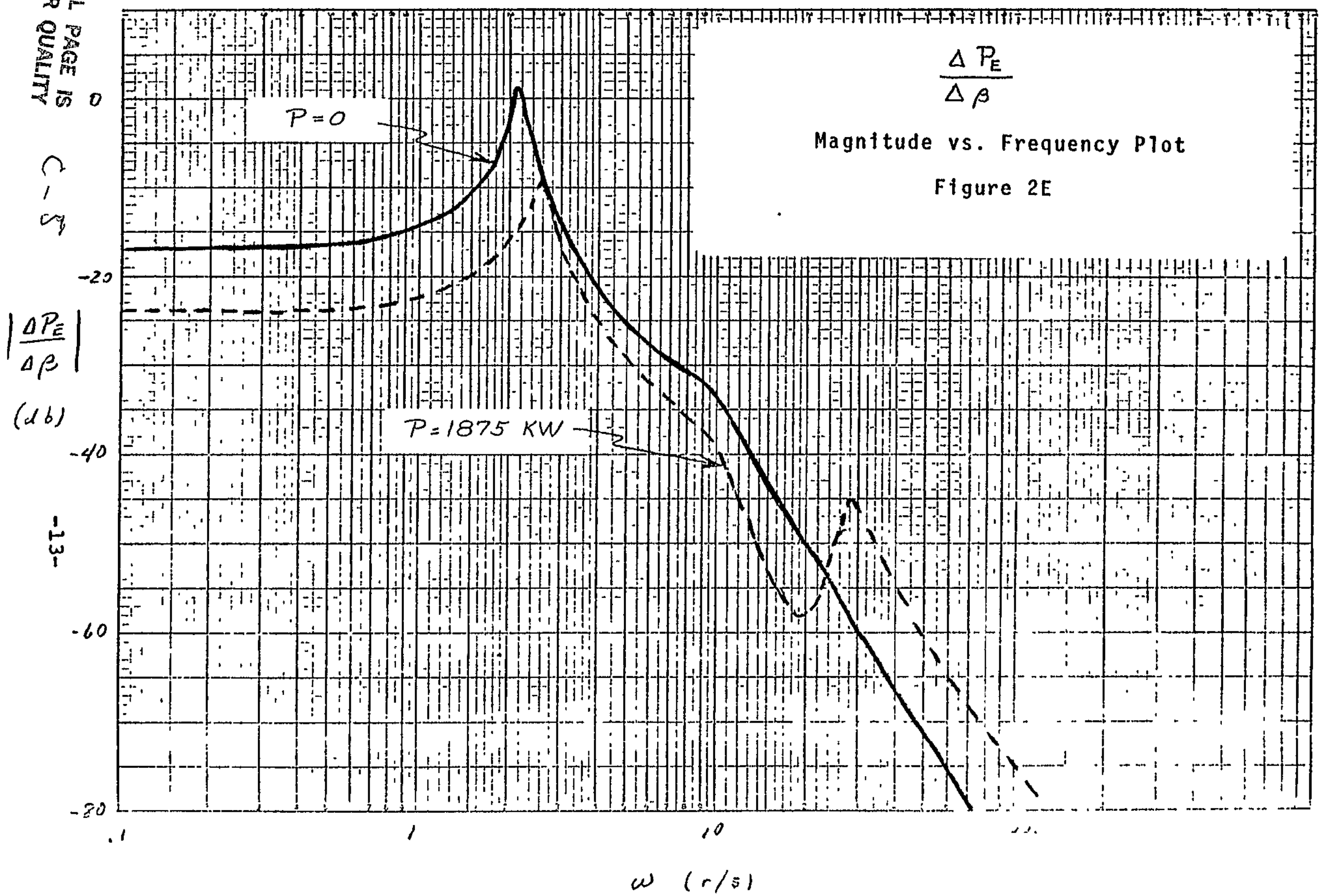
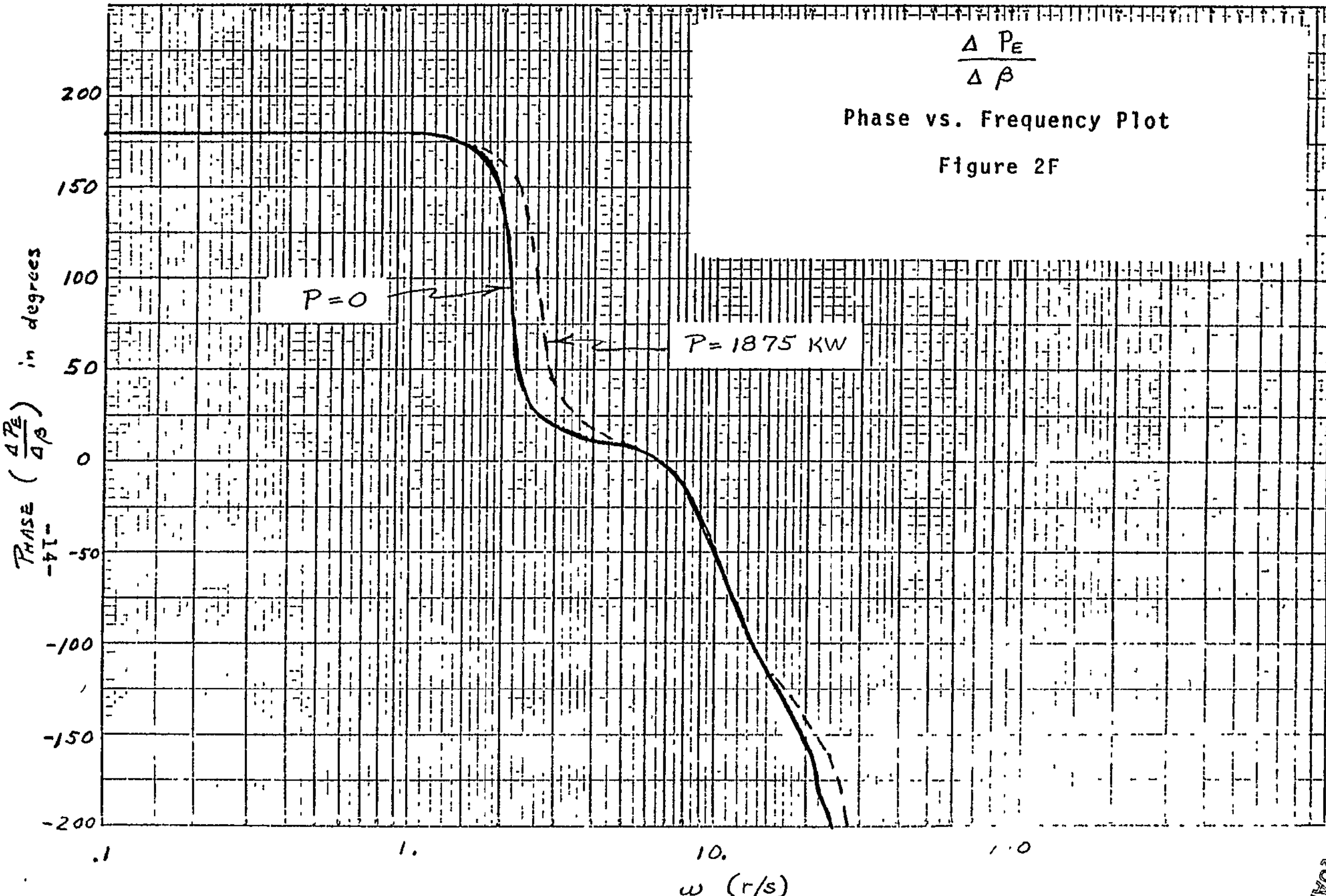
Root Locus Plot:
Power varied from 0 to 1875KW

Figure 2D

C-9

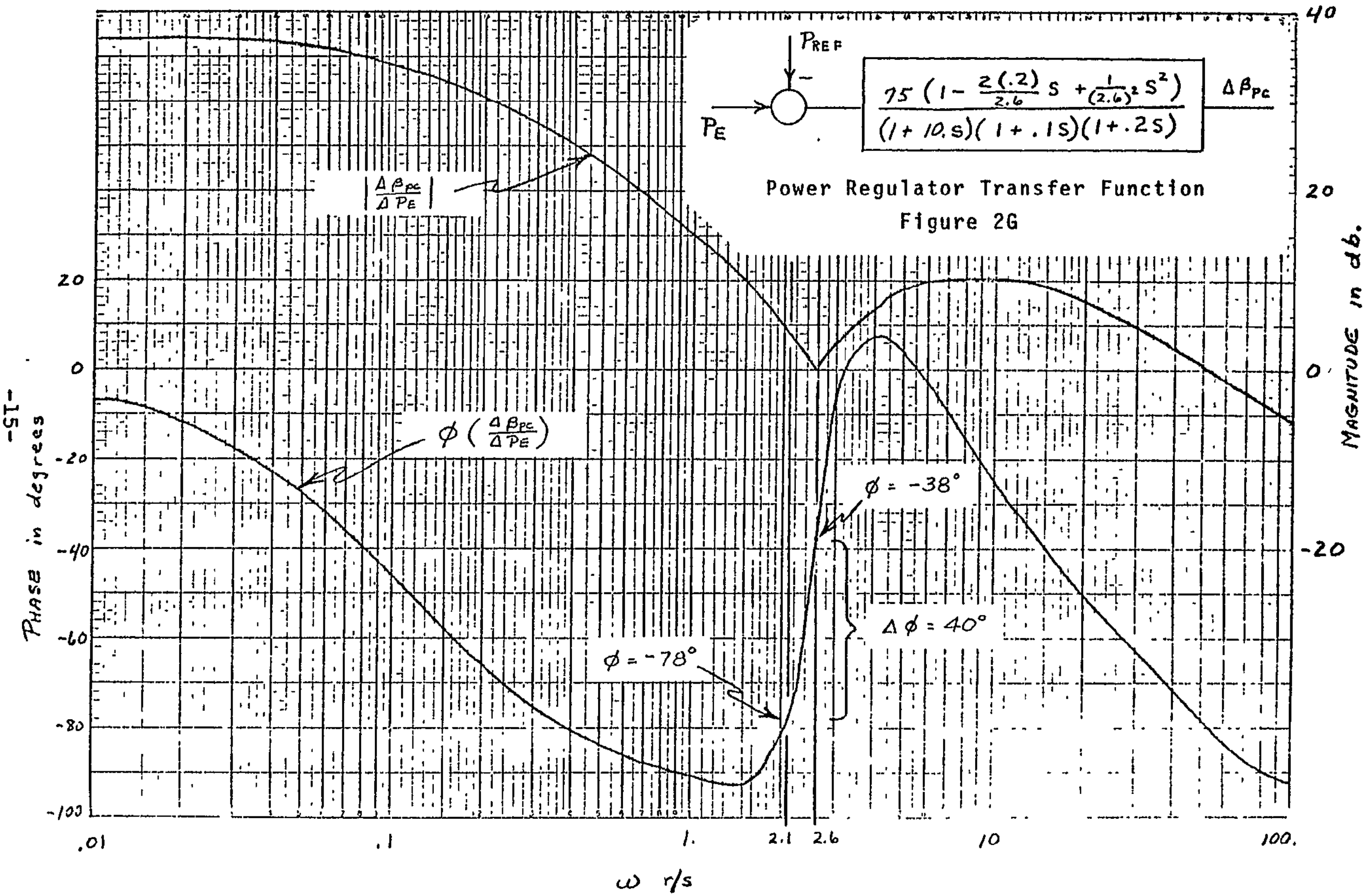


$\frac{\Delta P_E}{\Delta \beta}$
Magnitude vs. Frequency Plot
Figure 2E
P = 0
P = 1875 KW
0
-20
-40
-60
-80
$\left|\frac{\Delta P_E}{\Delta \beta}\right|$
(db)
.1
1
10
ω (r/s)



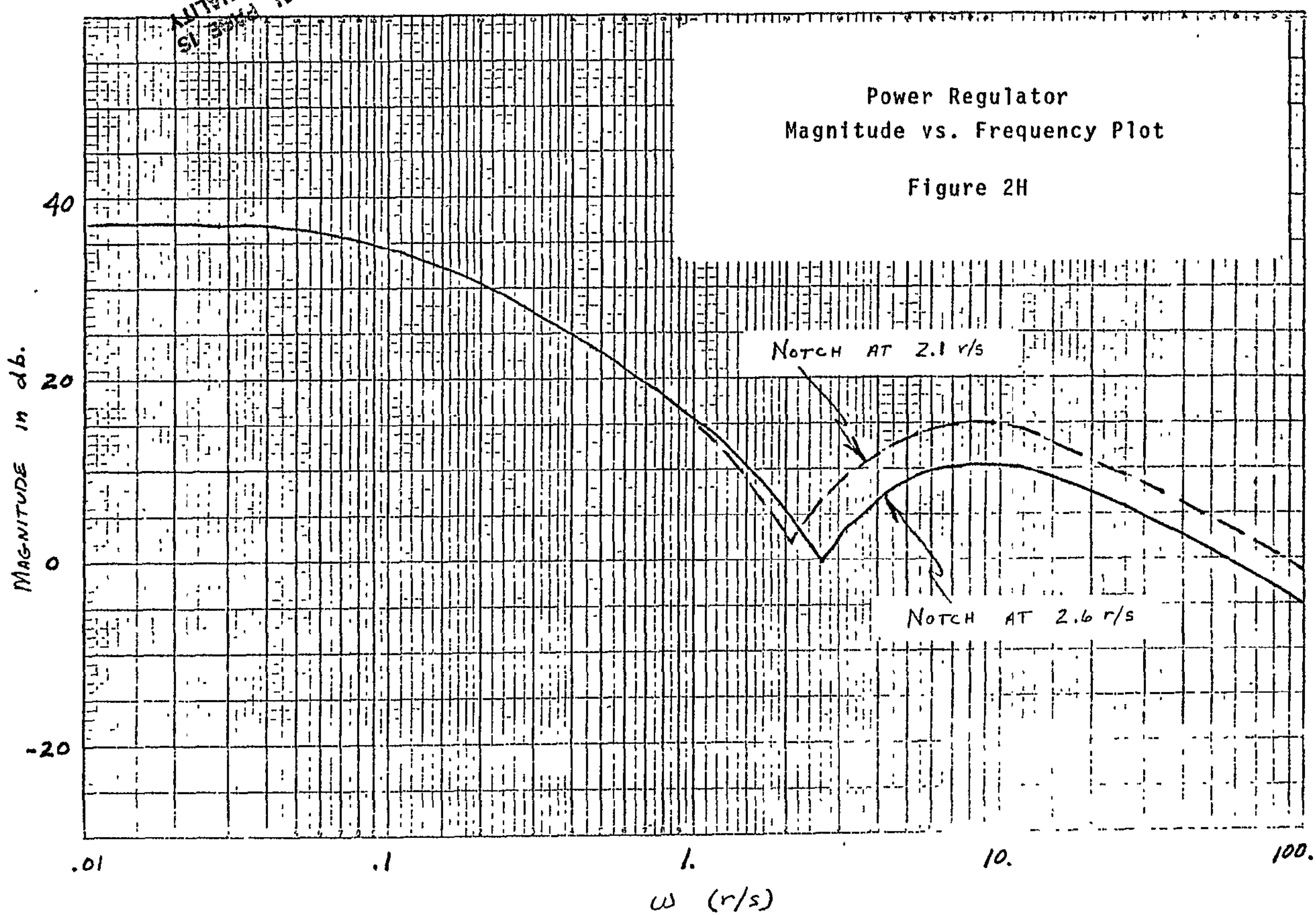
$\frac{\Delta P_E}{\Delta \beta}$
Phase vs. Frequency Plot
Figure 2F
P=0
P=1875 KW
200
150
100
50
0
-50
-100
-150
-200
PHASE $\left(\frac{\Delta P_E}{\Delta \beta}\right)$ in degrees
.1
1.
10.
1.0
ω (r/s)

-14-



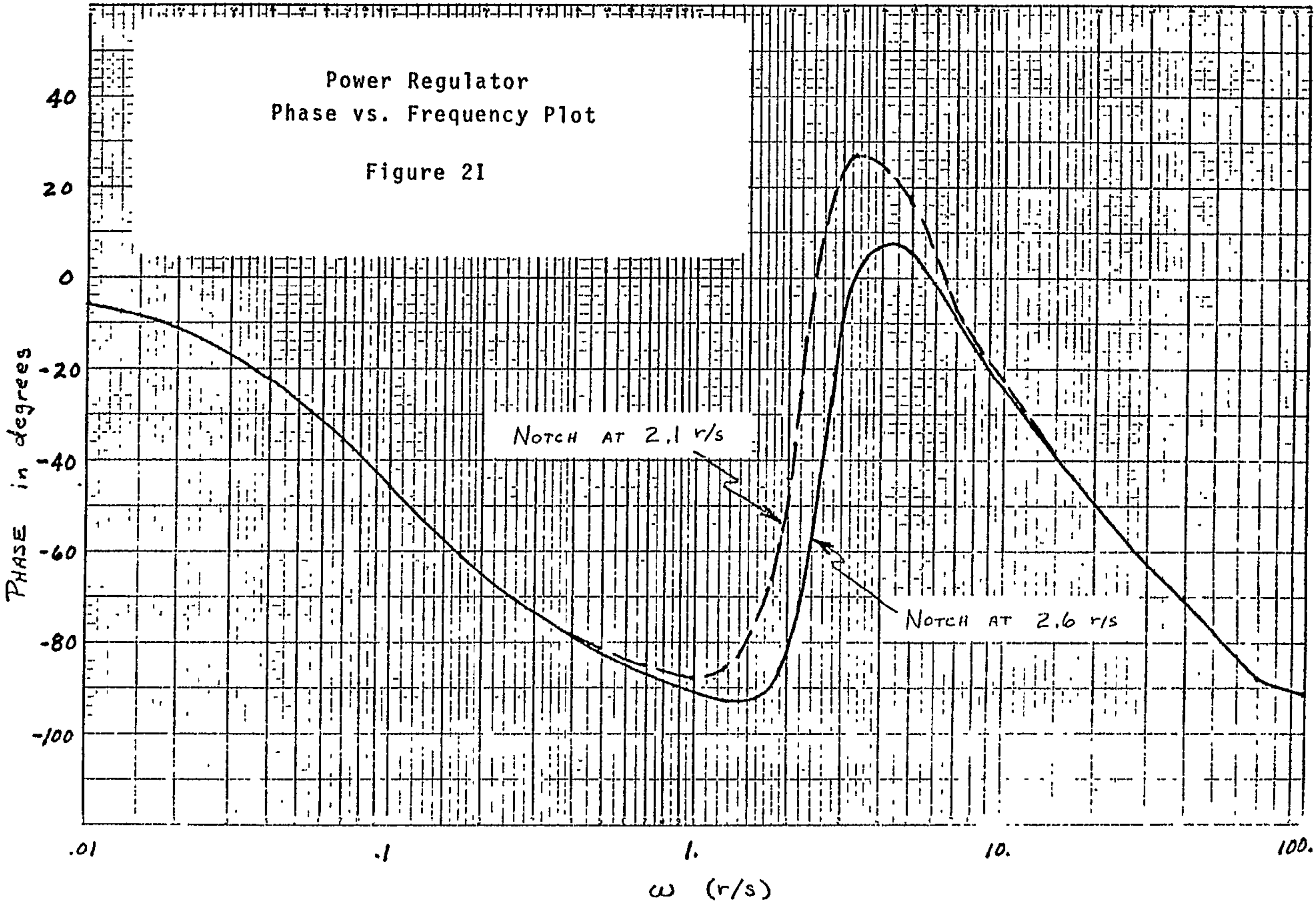
Power Regulator Transfer Function

Figure 2G



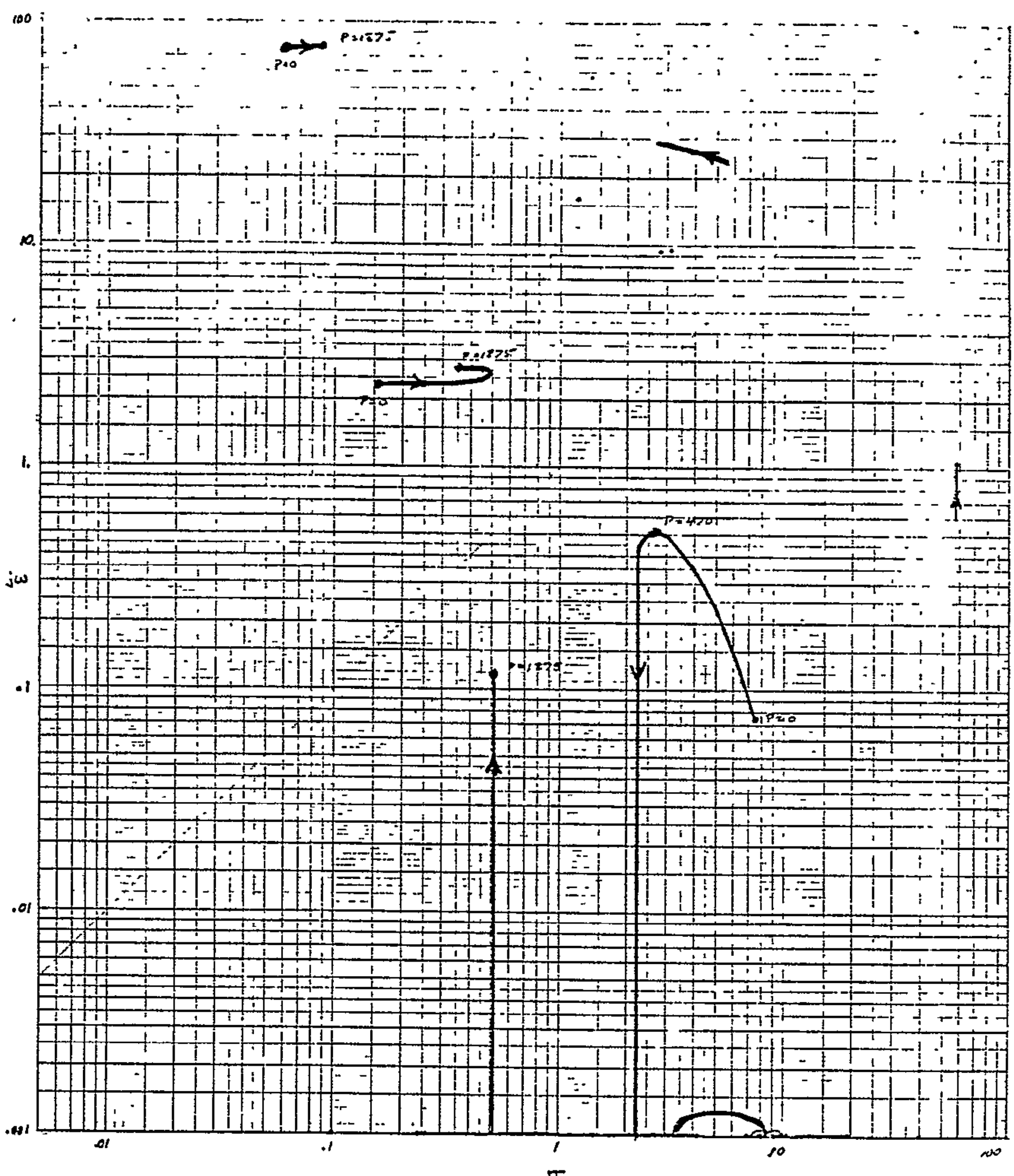
Power Regulator
Magnitude vs. Frequency Plot

Figure 2H



Power Regulator
Phase vs. Frequency Plot

Figure 21



Regulator notch at 2.1 r/s

$K_A = 975$
$K_p = 75$
$K_{STAB} = 100$

Root Locus Plot:
Power varied from 0 to 1875KW

Figure 2J

SYSTEM CONDITION		RIGID BODY MODE				12 HZ TORSIONAL MODE			
	Z EQUIV	P = 1875 KW		P = 0		P = 1875 KW		P = 0	
DESCRIPTION	(Z MVA BASE)	EIGENVALUE	ZETA	EIGENVALUE	ZETA	EIGENVALUE	ZETA	EIGENVALUE	ZETA
TIED TO BOONE 12 KV BUS	.142	-.365±j2.72	.1329	-.142±j2.24	.0634	-.0815±j76.8	.00106	-.0618±j76.6	.00081
TIED TO SHERWOOD 12 KV BUS	.172	-.373±j2.70	.1367	- 140±j2.23	.0625	-.0743±j76.8	.00097	-.0561±j76.6	.00073
TIED TO HOUND EARS 12 KV BUS	.172	-.374±j2.70	.1374	-.137±j2.22	.0617	-.0735±j76.8	.00096	-.0555±j76.6	.00072
TIED TO BOONE & SHERWOOD & HOUND EARS	.112	-.360±j2:75	.1299	-.169±j2.31	.0733	-.0907±j76.8	.00118	-.0746±j76.6	.00097
TIED TO BOONE W/ZSYS = .6 ON 10 MVA	.234	-.414±j2.67	.1531	-.185±j2.39	.0775	-.0696±j76.7	.00091	-.0701±j76.6	.00091
TIED TO BOONE W/ZSYS = 1.2 ON 10 MVA	.352	-.543±j2.55	.2086	--	--	-.0628±j76.6	.00082	--	--

EIGENVALUES FOR VARIOUS AC SYSTEM CONDITIONS

FIGURE 2K

3. Gust Response

Another aspect of assessment of the WTG performance at its installation site is its response to wind gusts. Time simulations were made to determine the response of the WTG to two wind gusts:

1. a two second duration one-minus-cosine gust, and

2. the .1% probability gust derived from site data.

The two second gust consisted of an initial and final velocity of 37 ft/sec., with a peak velocity of 47.5 ft/sec. occurring one second after the start of the gust. This gust model was used extensively in prior phases of this study. The .1% probability gust was the result of a statistical reduction of wind data taken at the Boone site. Its initial velocity was 42 ft/sec., and it attained a maximum velocity of 74 ft/sec. Velocity versus time plots for both of these gusts are shown in subsequent figures. For all these simulations the distribution system was assumed to be in its normal condition; that is, with Howard's Knob Circuit No. 211 connected only to the Boone 12 kV bus.

The control characteristics of the power regulator have a very strong influence on WTG response to wind gusts. Of particular concern is the control response in the frequency range over 1 r/s. In order to quantify this sensitivity and to assist in tuning the power regulator for optimum transient response, three levels of transient gain were considered. Assuming that the rigid body mode notch were moved to 2.1 r/s in the present regulator design, the asymptotic magnitude-versus-frequency characteristic would be that of the lowest curve in Figure 3A. System response to a two second wind gust was simulated utilizing this regulator design and the results are plotted in Figure 3B. Plotted quantities for these and all subsequent gust response plots are defined as follows:

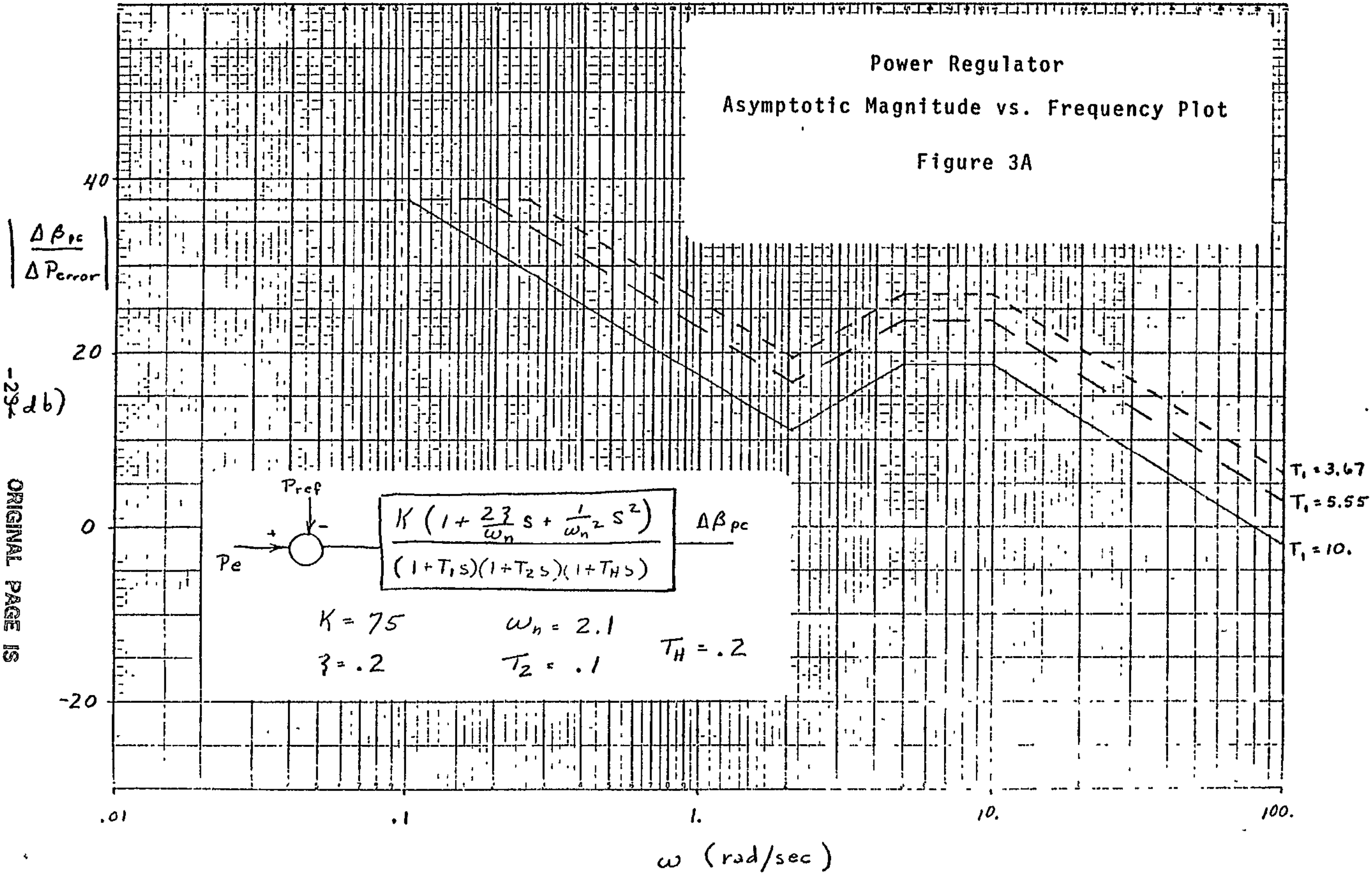
Variable	Description	Units
MCHSPD	Generator Rotor Speed	Hertz
PE	Generator Air Gap Power	Per Unit
PMECH	Windmill Rotor Power	Per Unit
BETA	Blade Pitch Angle	Degrees
WIND	Wind Velocity	Feet per Second
DELTPT	Deviation from Power Setpoint	Per Unit
VREG	Voltage Regulator Output	Per Unit
TERMINAL VOLTAGE	Generator Terminal Voltage	Per Unit
ANGLE	Angle of Generator "Q" Axis with respect to Infinite Bus	Degrees
VOLTAGE BUS 9		
VOLTAGE BUS 8		
VOLTAGE BUS 7	Bus Voltages as Identified in Figure 1	Per Unit
VOLTAGE BUS 6		
VOLTAGE BUS 5		

Response to the same gust was also simulated utilizing regulators with higher transient gains. A regulator with 5.2 db gain increase in the region above .2 r/s was produced by shifting the time constant T_1 from 10 to 5.5 seconds. Response to the 2 second gust using this regulator design is shown in Figure 3C. This design offers improved response over the lower gain regulator in that power-angle swings and voltage dips are substantially reduced. A third design with T_1 set at 3.67 seconds was also tested and the response is shown in Figure 3D. Further improvement in power swing and voltage dip response was achieved.

Significant data concerning maxima and minima from the 2 second gust response plots are summarized in tabular form in Figure 3E. For this forcing function, the highest gain regulator produces the best response.

A worst case, .1% probability, wind gust was also used as a forcing function in the assessment of WTG transient response. WTG response to the .1% probability gust was calculated with each of the three power regulator transfer functions tested with the 2 second gust. Transient response plots are shown in Figures 3F, 3G, and 3H. Voltage dip response is summarized in Figure 3I. Response to this gust has revealed characteristics not evident in the 2 second gust response. If plots of electrical and mechanical power response (PE and PMECH) for each regulator design are compared, it can be seen that the the highest gain regulator (with T_1 = 3.67) causes highly oscillatory behavior after the peak of the gust. Although the response is stable, this is an indication that the transient gain is too high.

Based on this gust response analysis it is evident that the power regulator design with T_1 = 5.5 seconds offers a necessary compromise in performance; it substantially improves voltage dip performance without risking post transient oscillations as with the higher gain regulator.



Power Regulator

Asymptotic Magnitude vs. Frequency Plot

Figure 3A

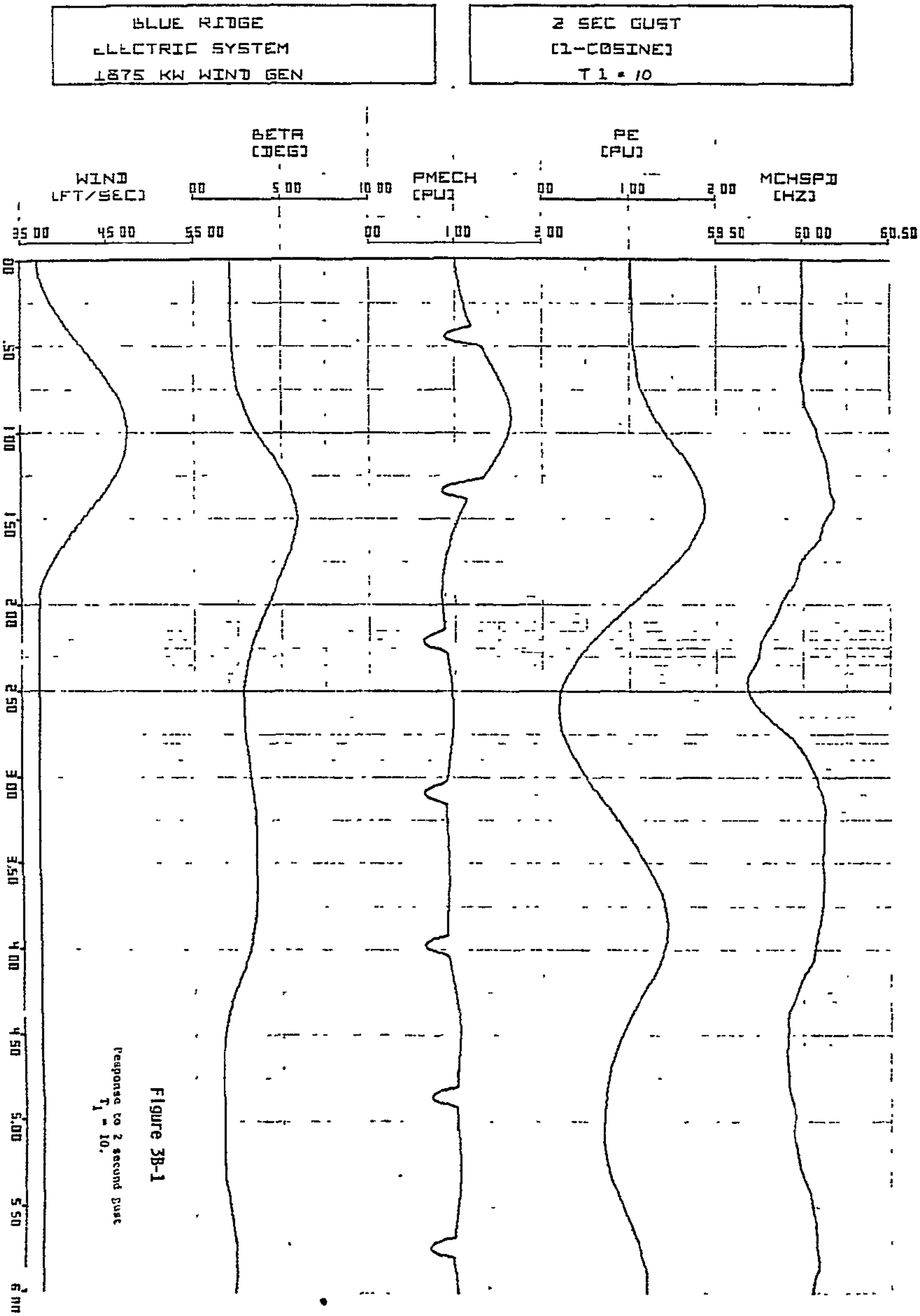


Figure 3B-1

Response to 2 second gust
$T_1 = 10.$

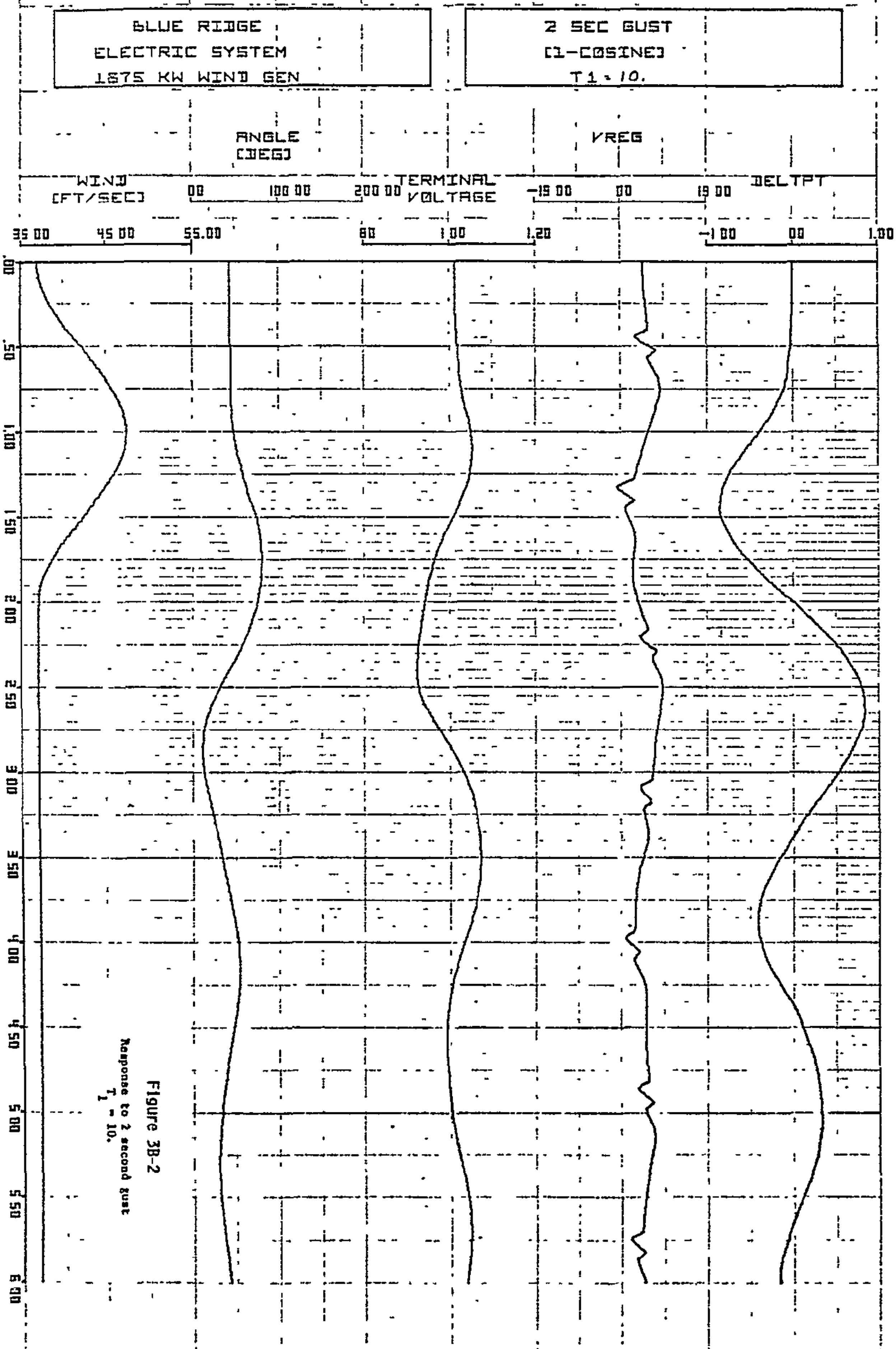


Figure 3B-2
Response to 2 second gust
$T_1 = 10.$

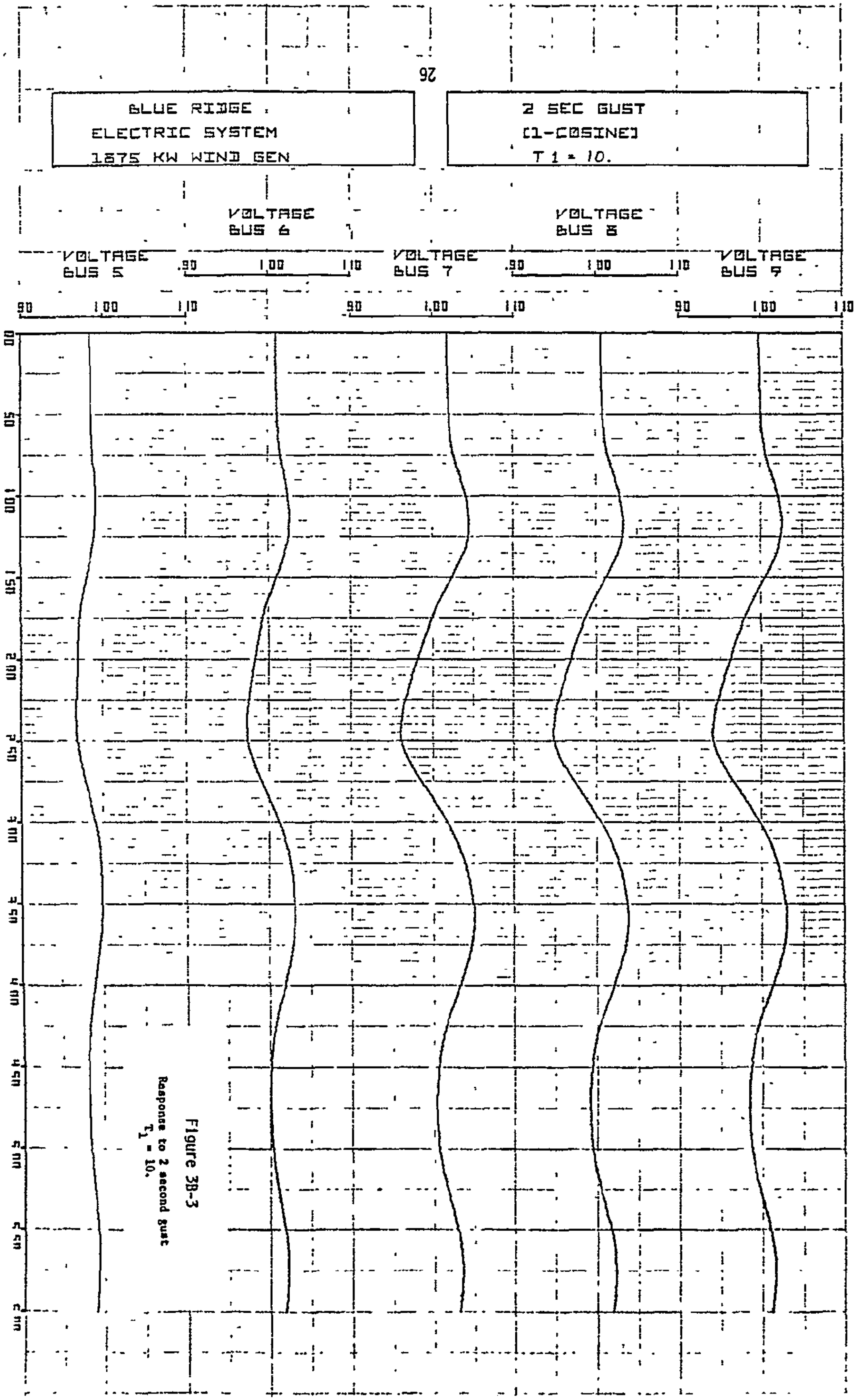


Figure 3B-3
Response to 2 second gust
T_1 = 10.

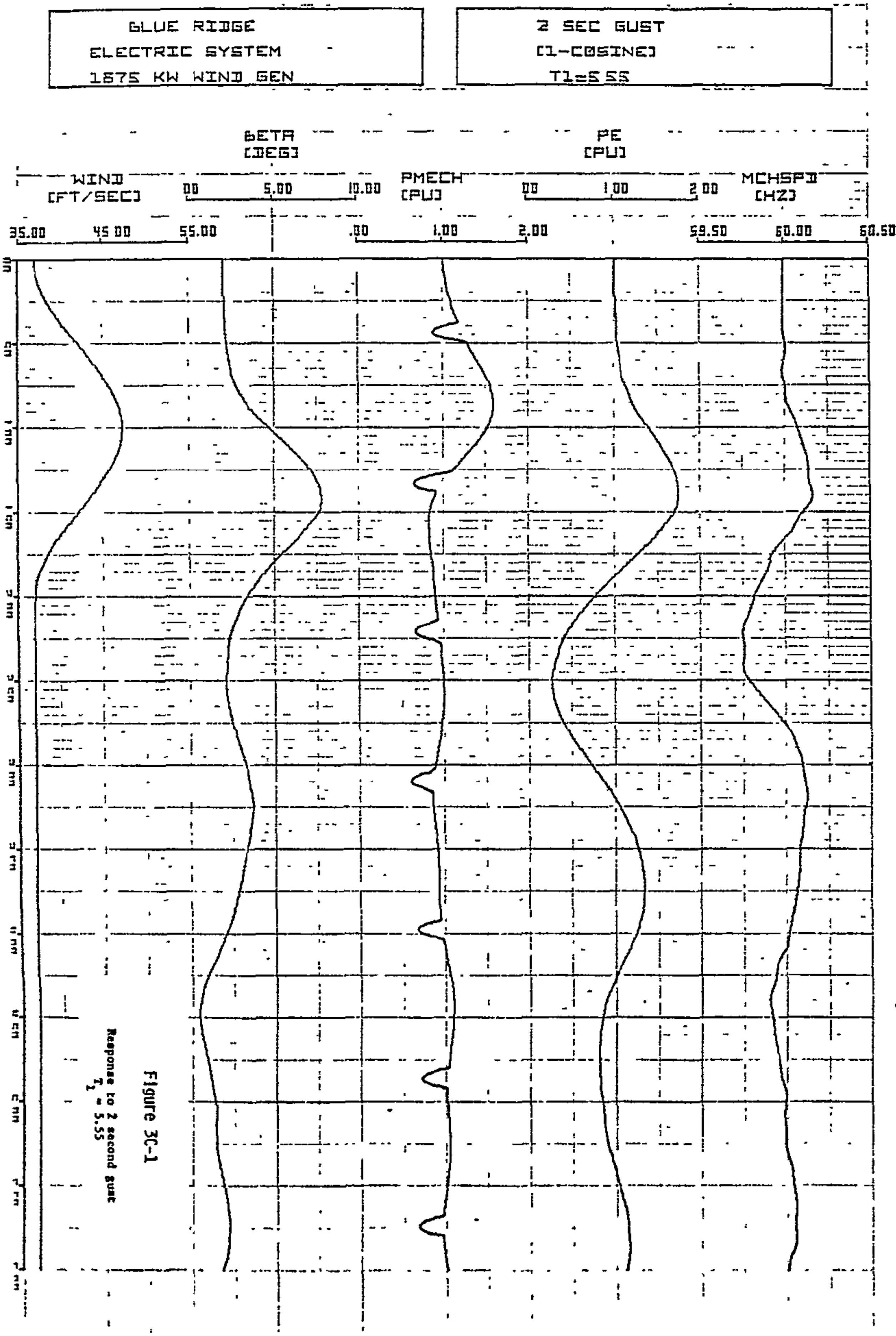


Figure 3C-1

Response to 2 second gust $T_1 = 5.55$

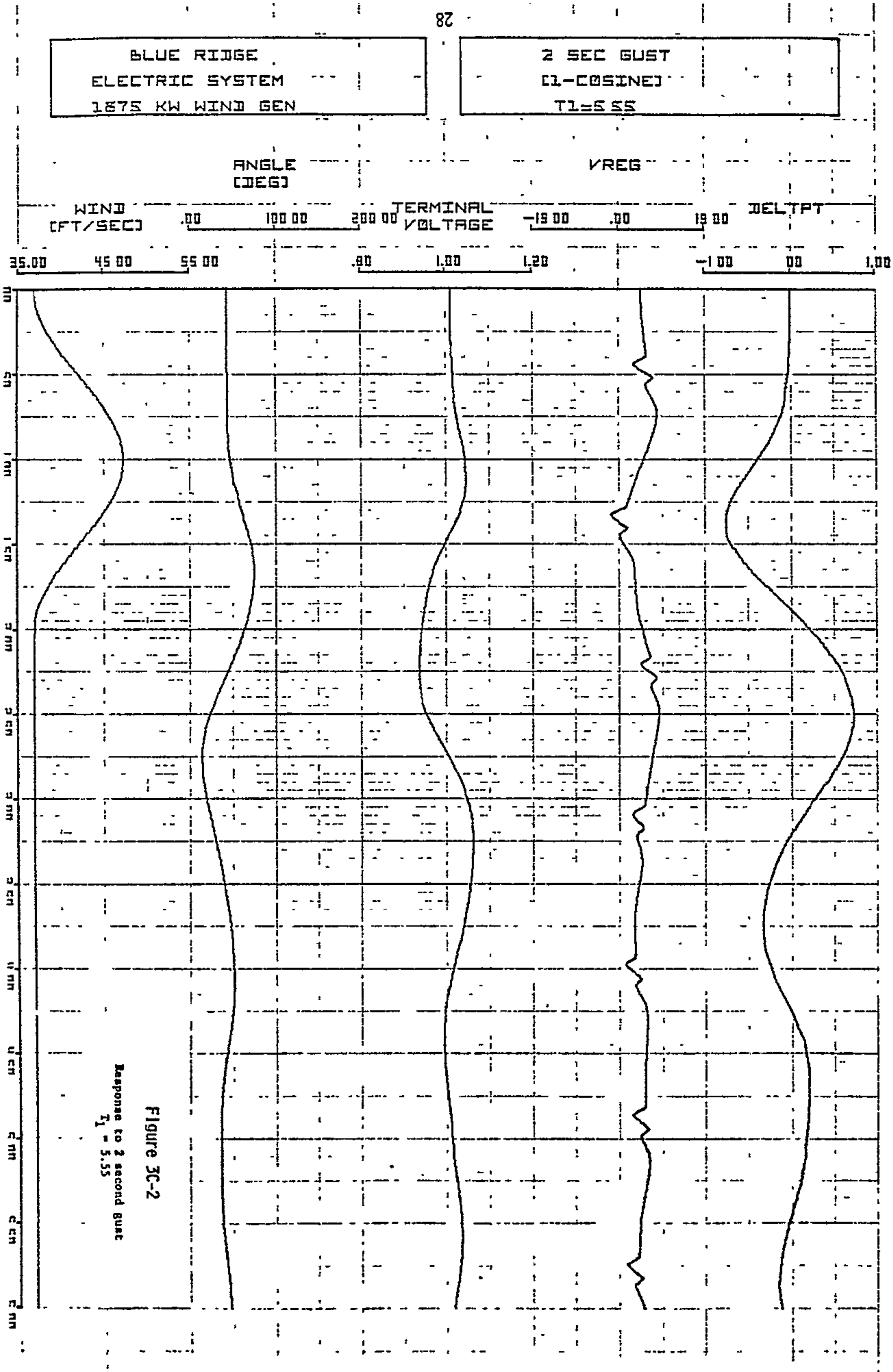


Figure 3C-2
Response to 2 second gust
$T_1 = 5.55$

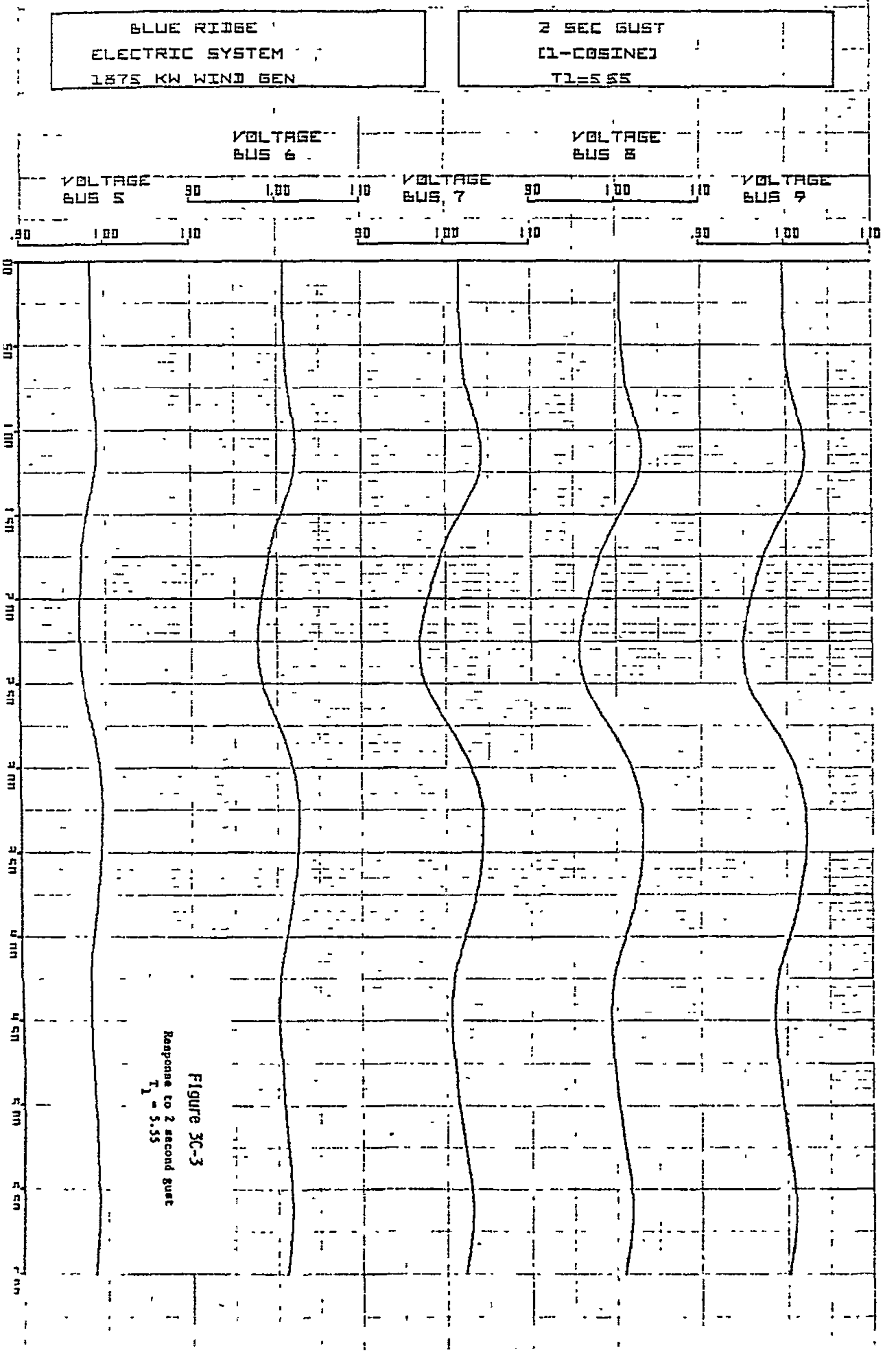


Figure 3C-3
Response to 2 second gust
$T_1 = 5.55$

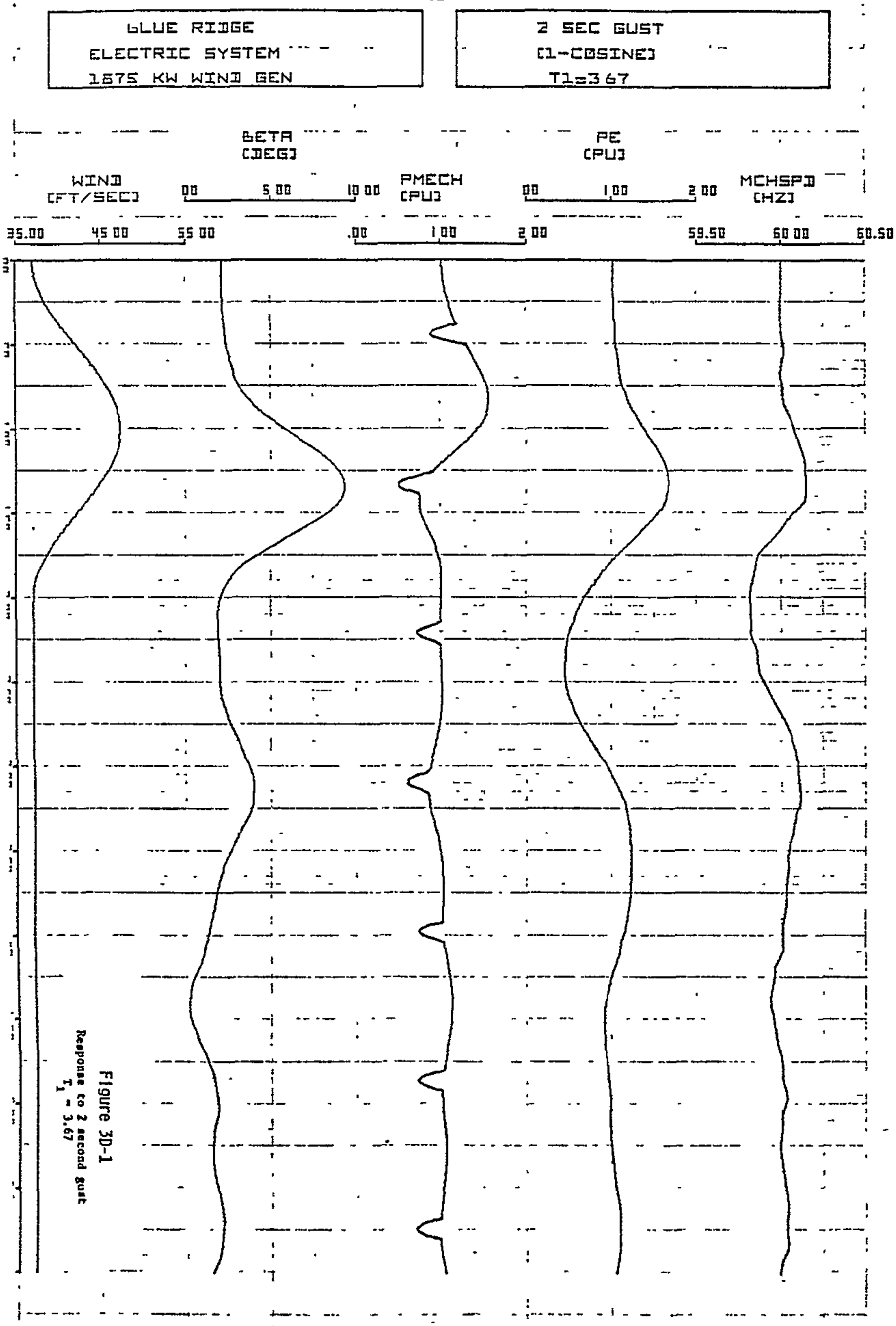


Figure 3D-1
Response to 2 second gust
$T_1 = 3.67$

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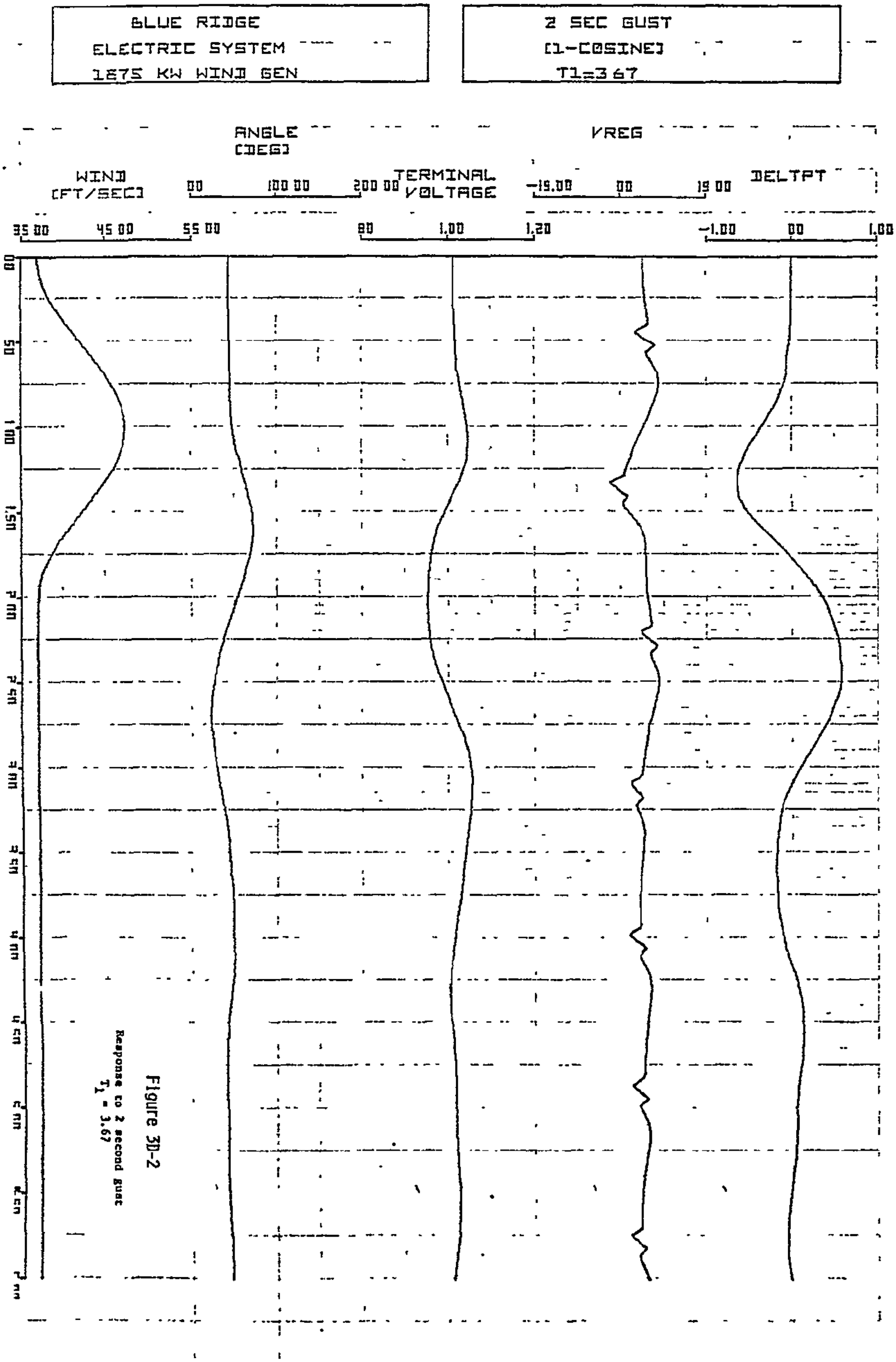


Figure 3D-2
Response to 2 second gust
T_1 = 3.67

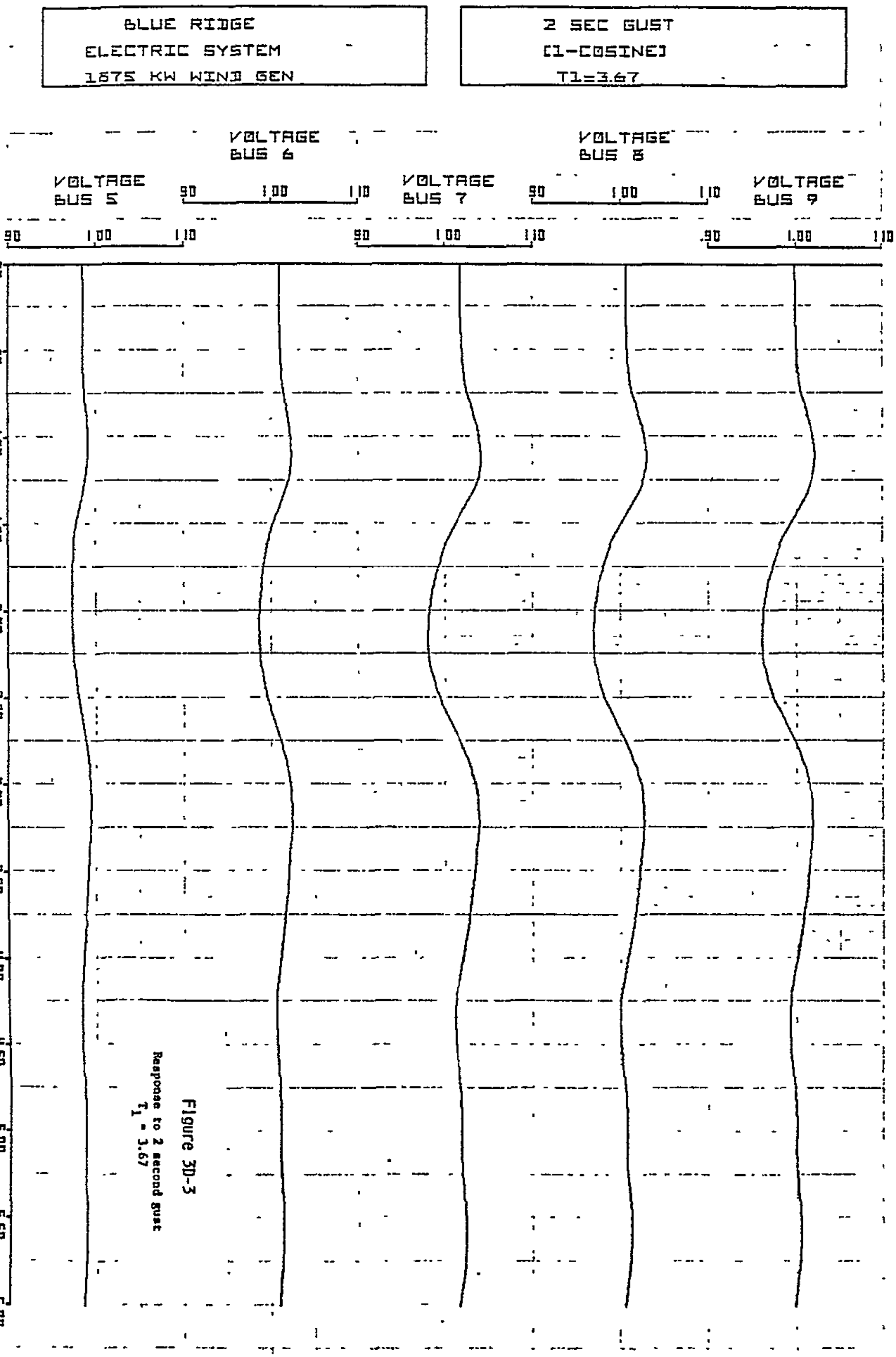


Figure 3D-3
Response to 2 second gust
T_1 = 3.67

Figure 3E: Time Response Summary

2 sec 1- Cosine Gust

		T1 = 10	T1 = 5.55	T1 = 3.67
PE (pu)	initial	1.02	1.02	1.02
	maximum	1.88	1.75	1.67
	minimum	.19	.26	.44
ANGLE (deg)	initial	44.3	44.3	44.3
	maximum	80.2	76.1	72.0
	minimum	9.9	13.9	21.0
TERMINAL VOLTAGE (pu)	initial	1.014	1.014	1.014
	maximum	1.072	1.064	1.056
	minimum	.924 (8.9%)	.937 (7.6%)	.953 (6.0%)
BETA (deg)	initial	2.1	2.1	2.1
	maximum	6.0	7.8	9.4
	minimum	1.6	0.5	.1
VOLTAGE BUS 5 (pu)	initial	.985	.985	.985
	maximum	.999	.998	.995
	minimum	.968 (1.7%)	.972 (1.3%)	.973 (1.2%)
VOLTAGE BUS 7 (pu)	initial	1.019	1.019	1.019
	maximum	1.050	1.044	1.039
	minimum	.960 (5.8%)	.970 (4.8%)	.980 (3.8%)
VOLTAGE BUS 9 (pu)	initial	.998	.998	.998
	maximum	1.030	1.024	1.019
	minimum	.941 (5.7%)	.951 (4.7%)	.961 (3.7%)

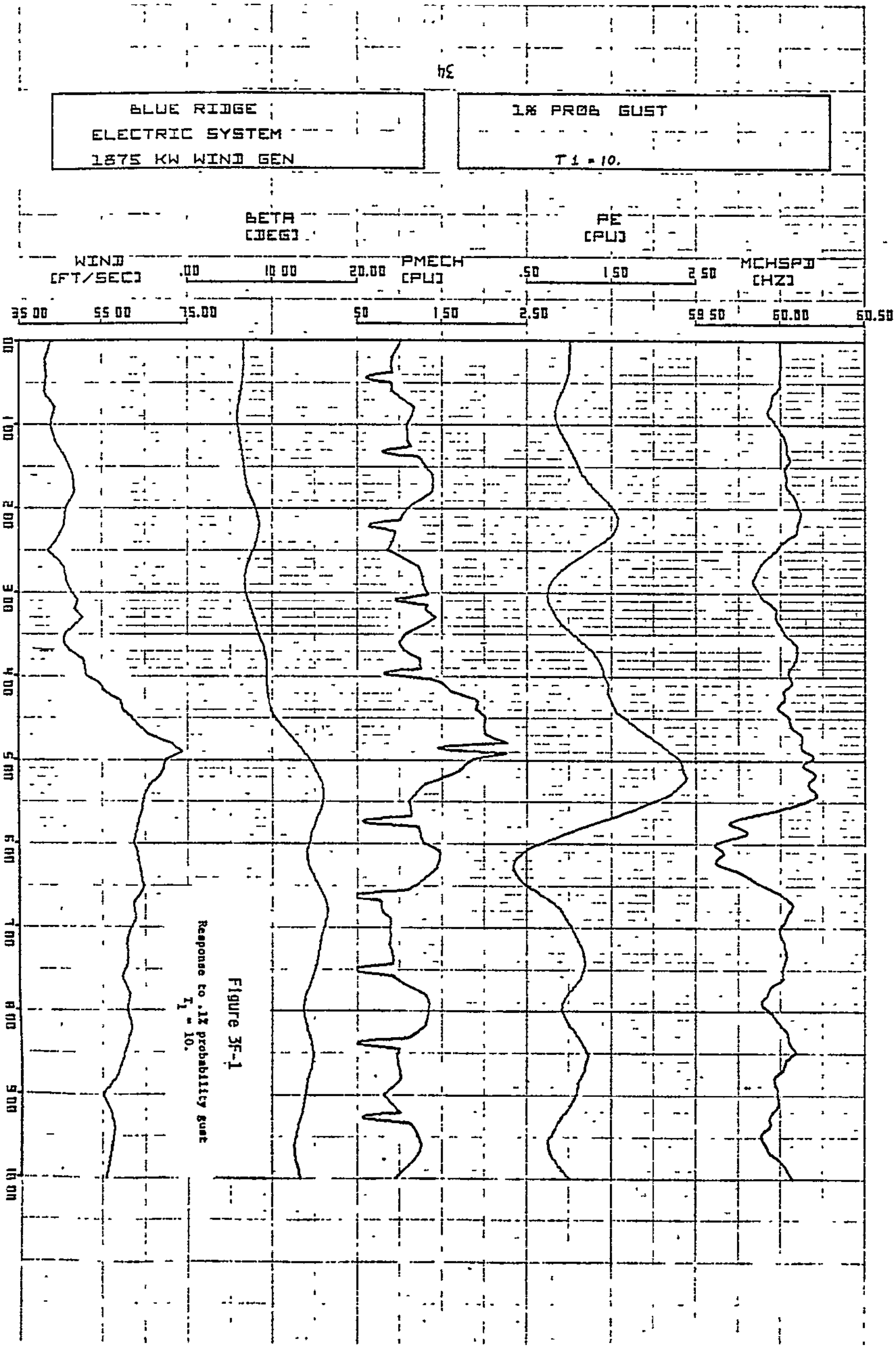


Figure 3F-1
Response to .1% probability gust
$T_1 = 10.$

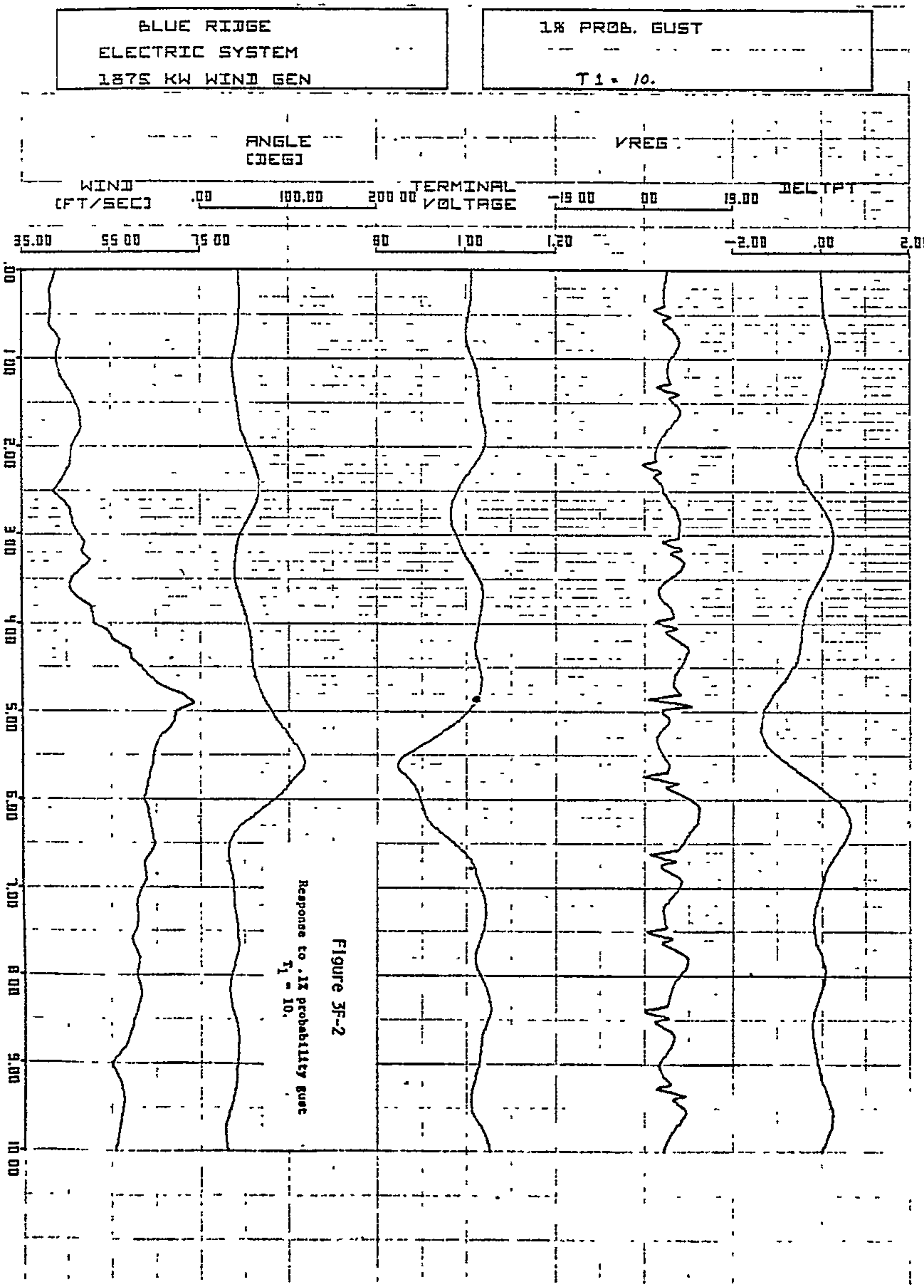


Figure 3F-2

Response to .1% probability gust
T_1 = 10.

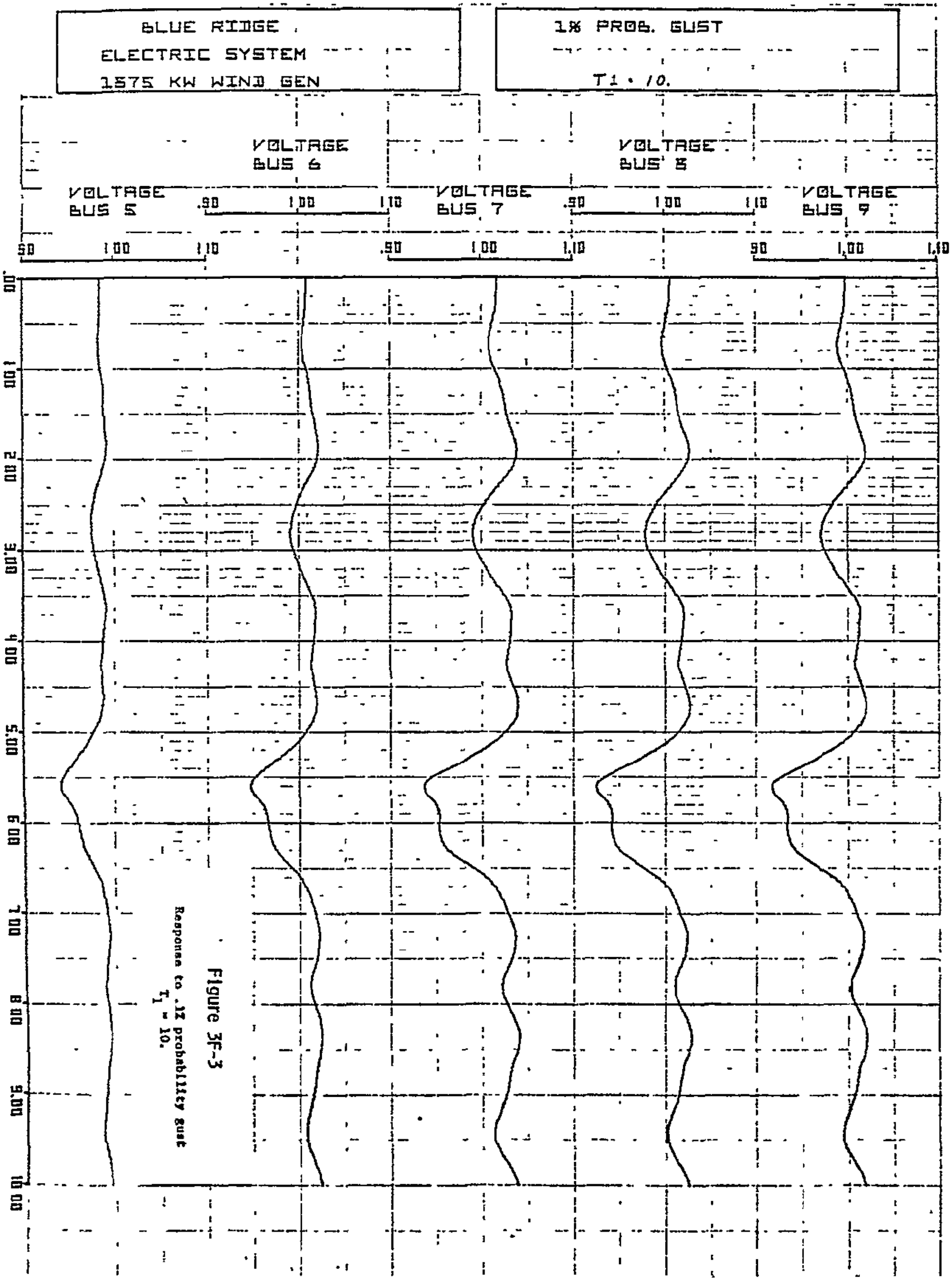


Figure 3F-3

Response to .1% probability gust
$T_1 = 10.$

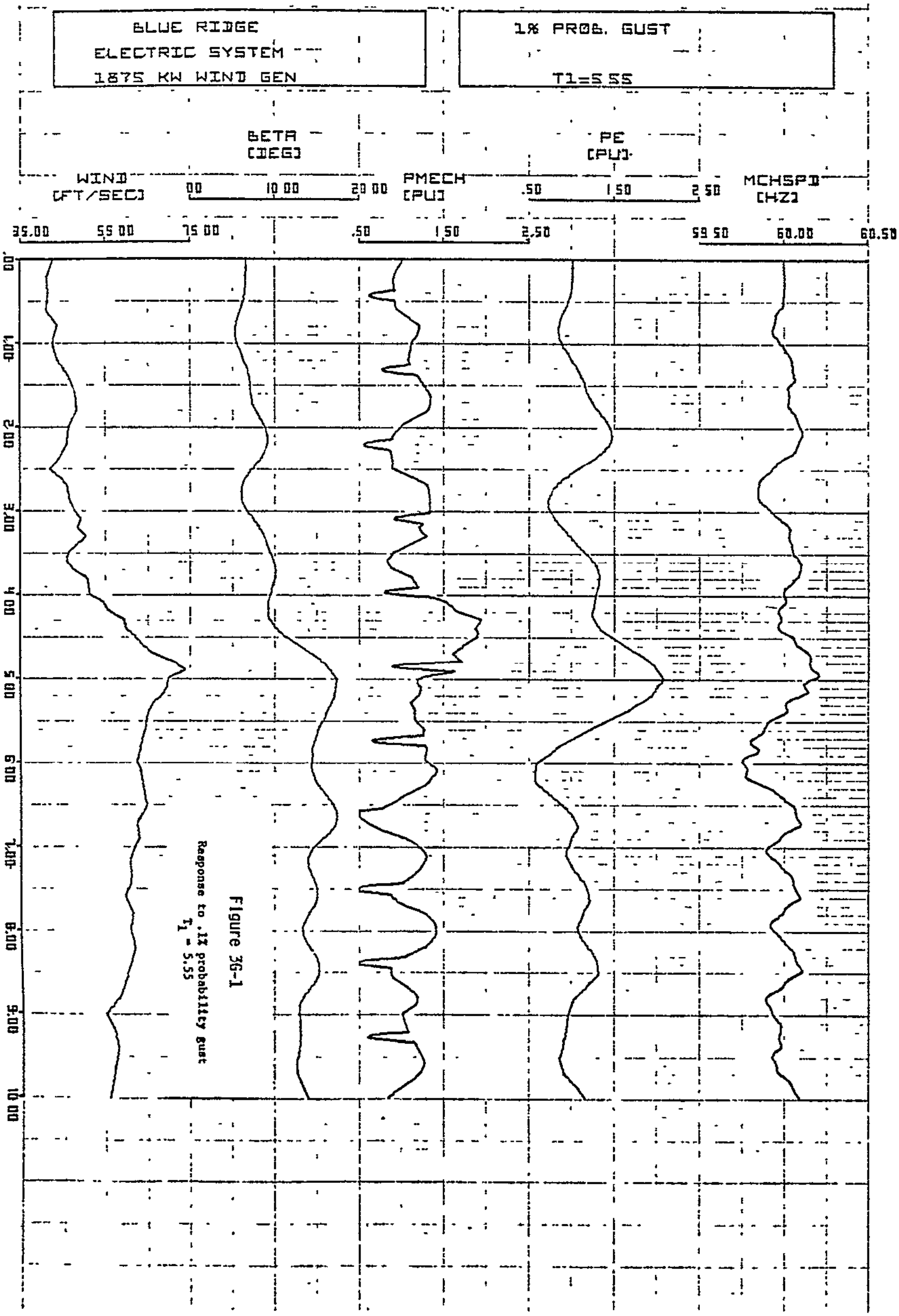


Figure 3G-1
Response to .1% probability gust
$T_1 = 5.55$

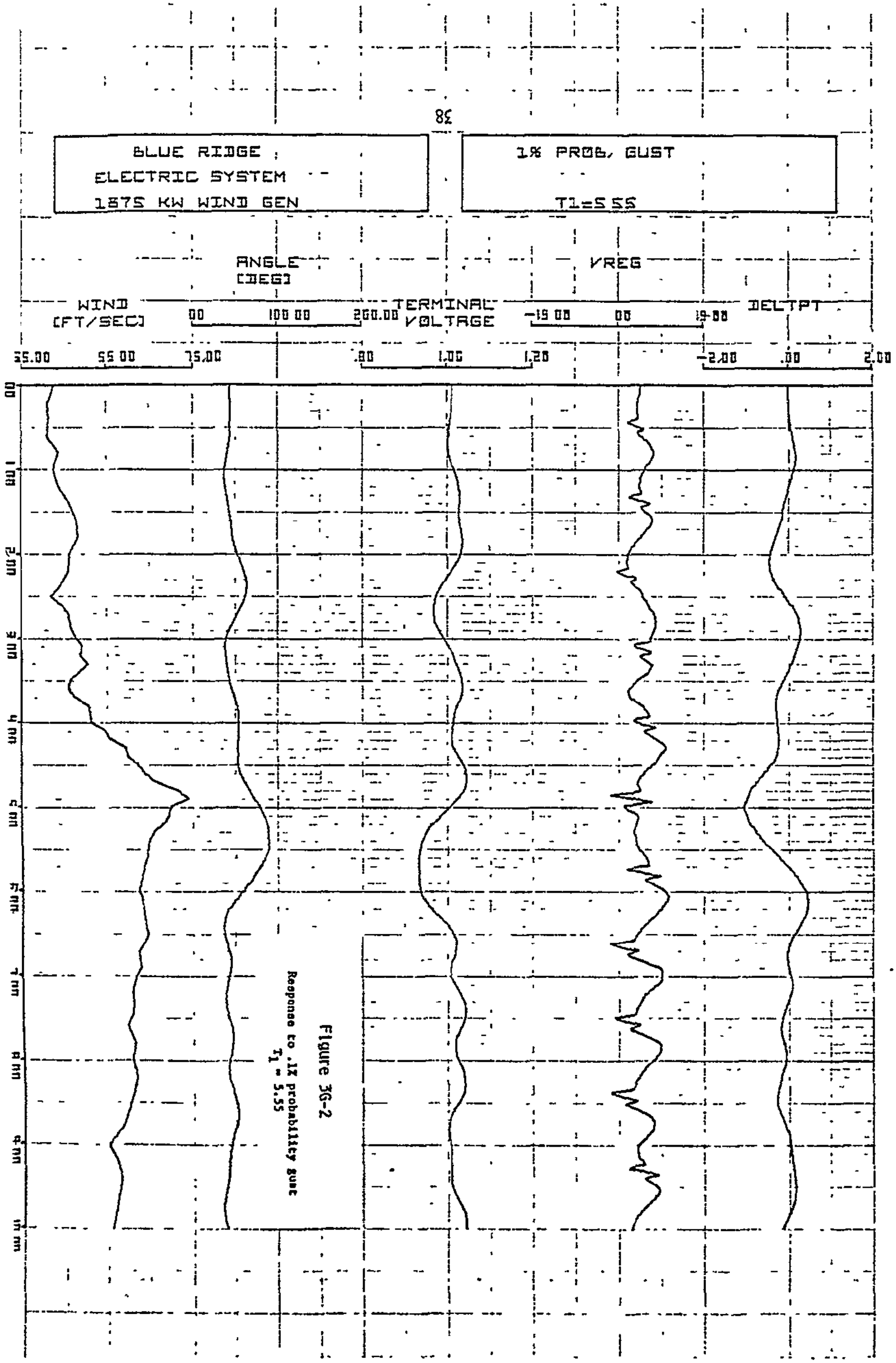


Figure 3G-2
Response to .1% probability gust
T_1 = 5.55

39

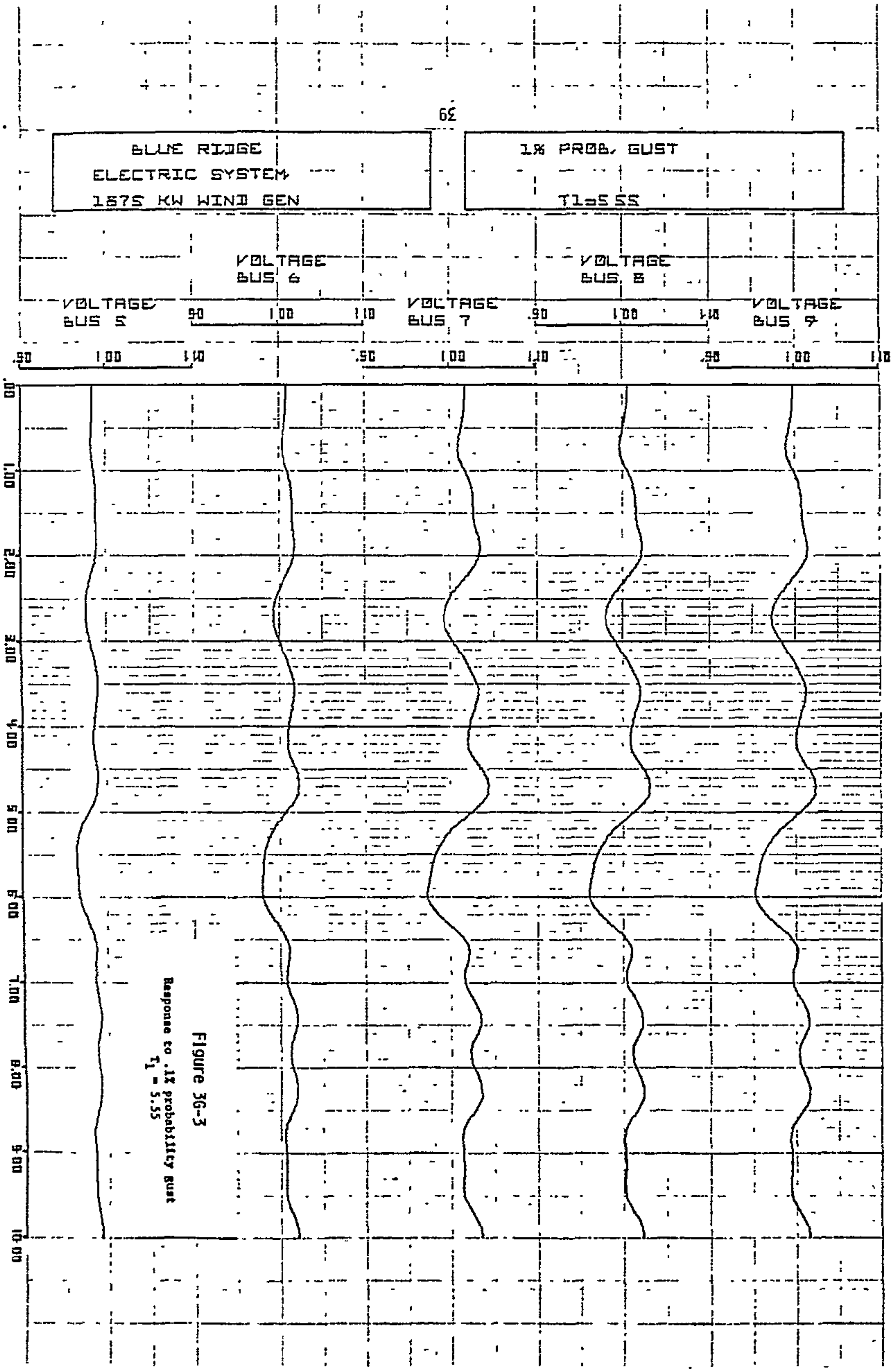


Figure 3G-3
Response to .1% probability gust
$T_1 = 5.55$

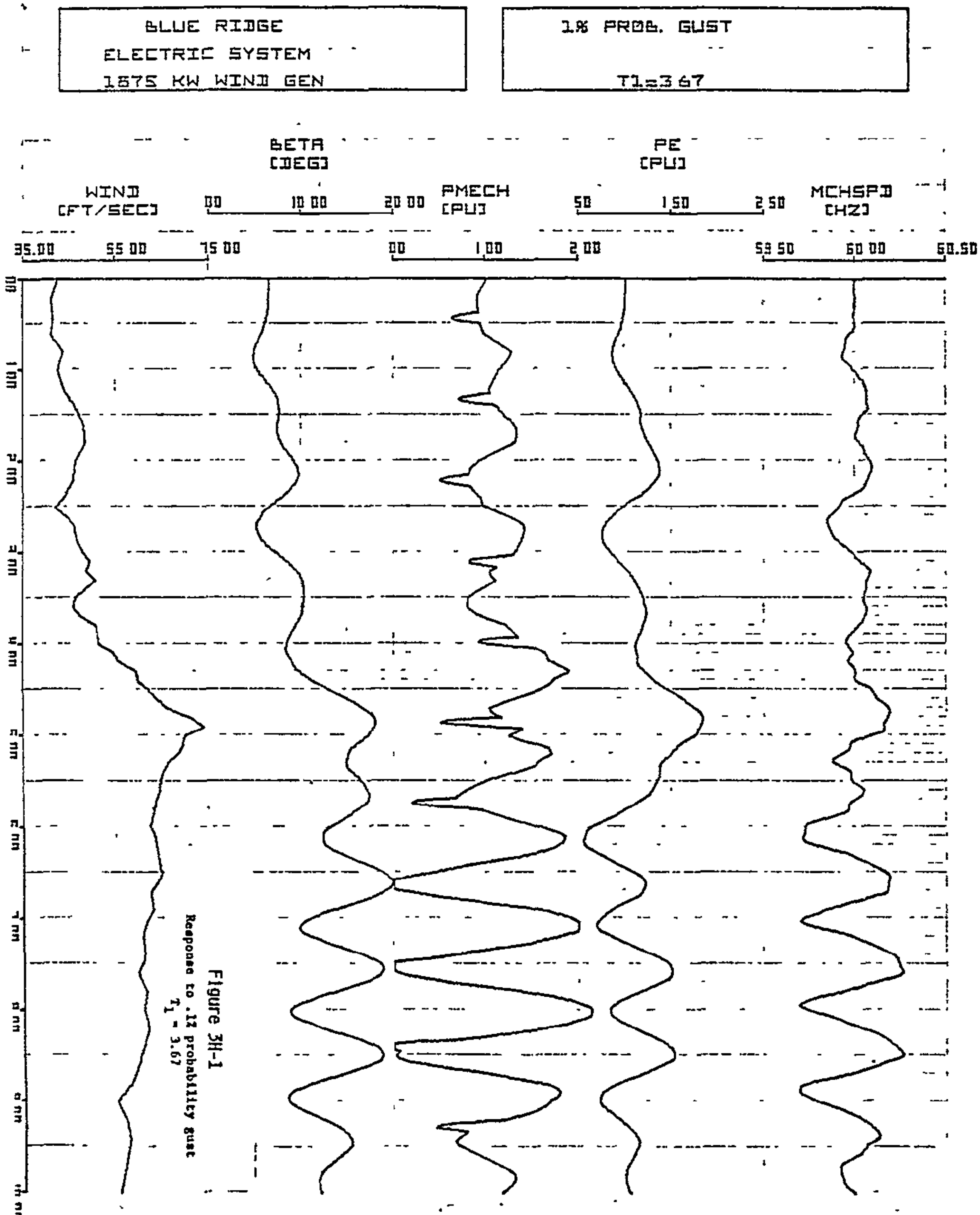


Figure 3H-1
Response to .1% probability gust
T_1 = 3.67

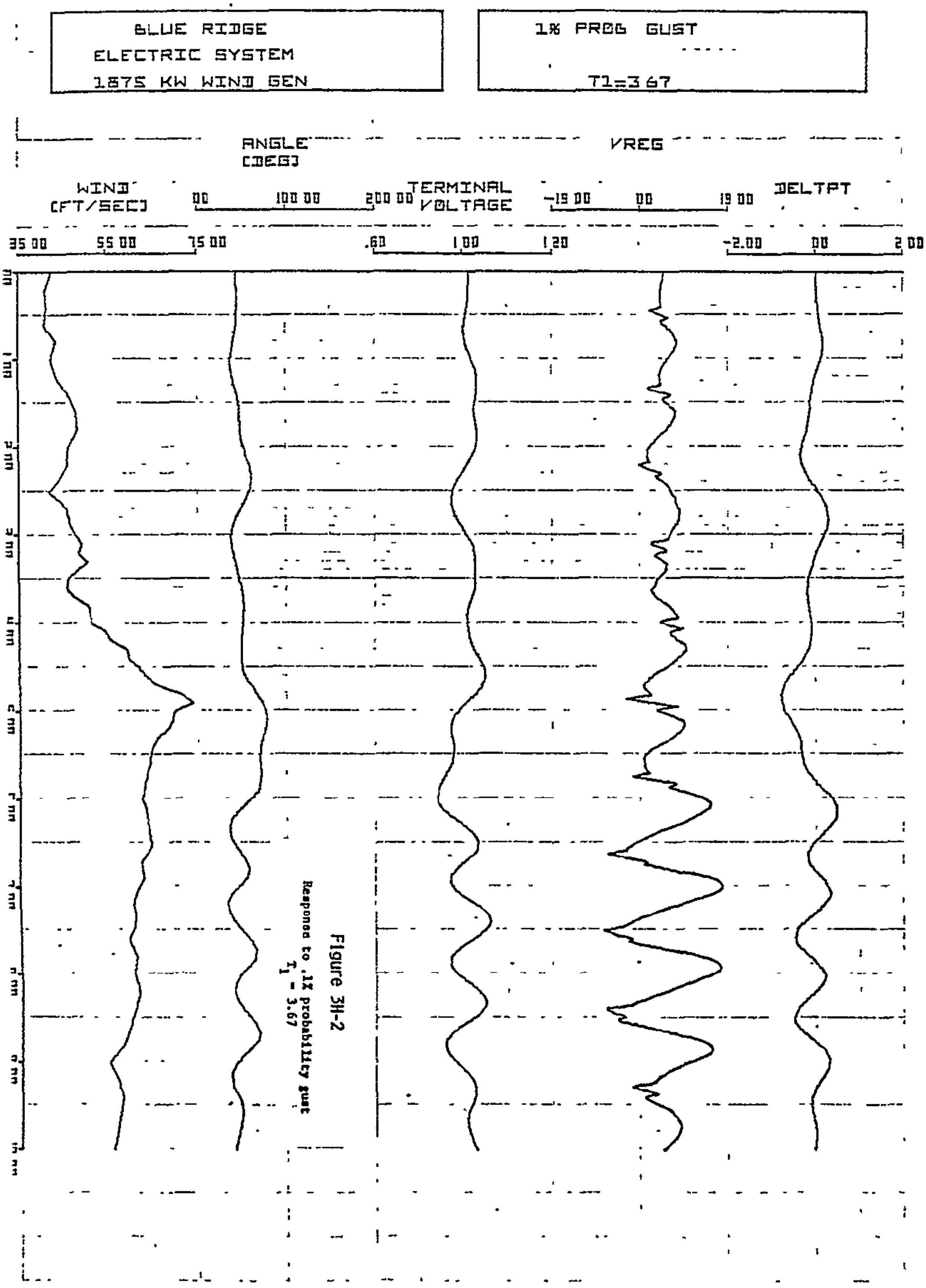


Figure 3H-2
Response to .1% probability gust
$T_1 = 3.67$

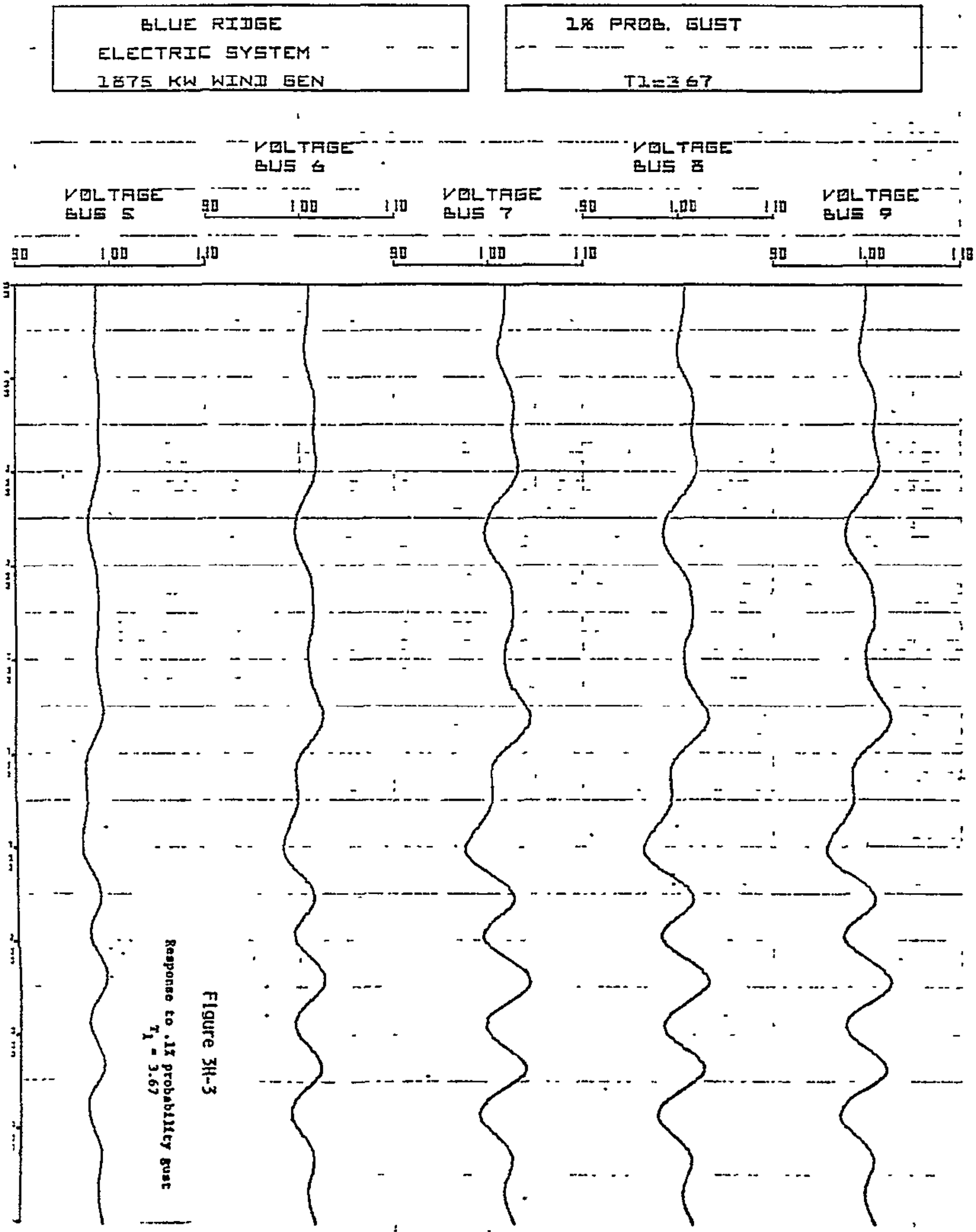


Figure 3H-3
Response to .1% probability gust
T_1 = 3.67

Figure 3I: Voltage Dip Response Summary

.1% Probability Gust

		Tl = 10	Tl = 5.55	Tl = 3.67
TERMINAL VOLTAGE (pu)	initial	1.014	1.014	1.014
	maximum	1.056	1.044	1.058
	minimum	.848 (16.4%)	.932 (8.1%)	.940 (7.3%)
VOLTAGE BUS 5 (pu)	initial	.985	.985	.985
	maximum	.994	.992	.991
	minimum	.942 (4.4%)	.965 (2.0%)	.970 (1.5%)
VOLTAGE BUS 7 (pu)	initial	1.019	1.019	1.019
	maximum	1.041	1.043	1.044
	minimum	.937 (8.0%)	.972 (4.6%)	.975 (4.3%)
VOLTAGE BUS 9 (pu)	initial	.998	.998	.998
	maximum	1.020	1.022	1.023
	minimum	.918 (8.0%)	.953 (4.5%)	.956 (4.2%)

4. Conclusions and Recommendations

The following conclusions can be drawn from the results of this study:

1. Overall performance of the MOD-1 Wind turbine Generator is strongly influenced by the characteristics of both the voltage regulator and power controller. Interactions between these two control devices can produce dangerously high torques in the shaft at torsional natural frequencies. It is important, therefore, to implement a coordinated control policy for these devices.

2. Dynamic instability at low power output was indicated when the wind turbine-generator was operated with its original design power regulator. Adjustment of the power regulator notch filter from 2.6 r/s to 2.1 r/s eliminated this instability. (The original design data is included in the appendix.)

3. The transient stability margin and voltage dip response of the wind turbine-generator to wind gusts were substantially improved by increasing the transient gain of the power regulator. This gain modification was implemented by adjusting the time constant T_1 from 10. to 5.5 seconds.

4. With the control parameter adjustments mentioned above, the wind turbine-generator responded well to the .1% probability (worst case) wind gust. Angular excursions were very small and the maximum voltage dip on the distribution feeder was 4.6%. No stability problems were indicated.

Appendices

A.1 System Reactance Diagram

A.2 System Load Diagram

A.3 Voltage Sensitive Load Characteristics

A.4 Generator Electrical Data

A.5 Excitation System Data

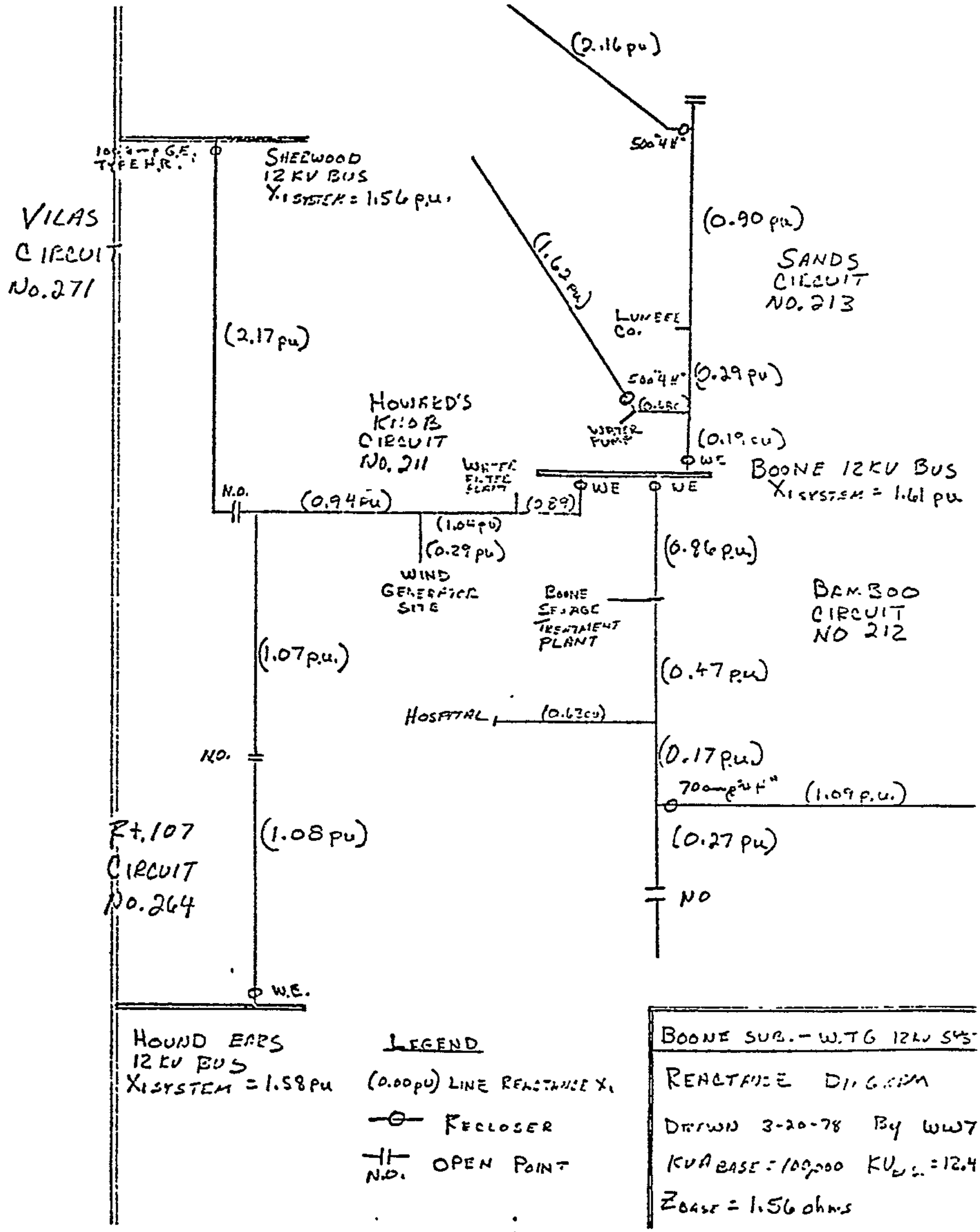
A.6 Power System Stabilizer Data

A.7 Power Regulator Data

A.8 Torsional System Data

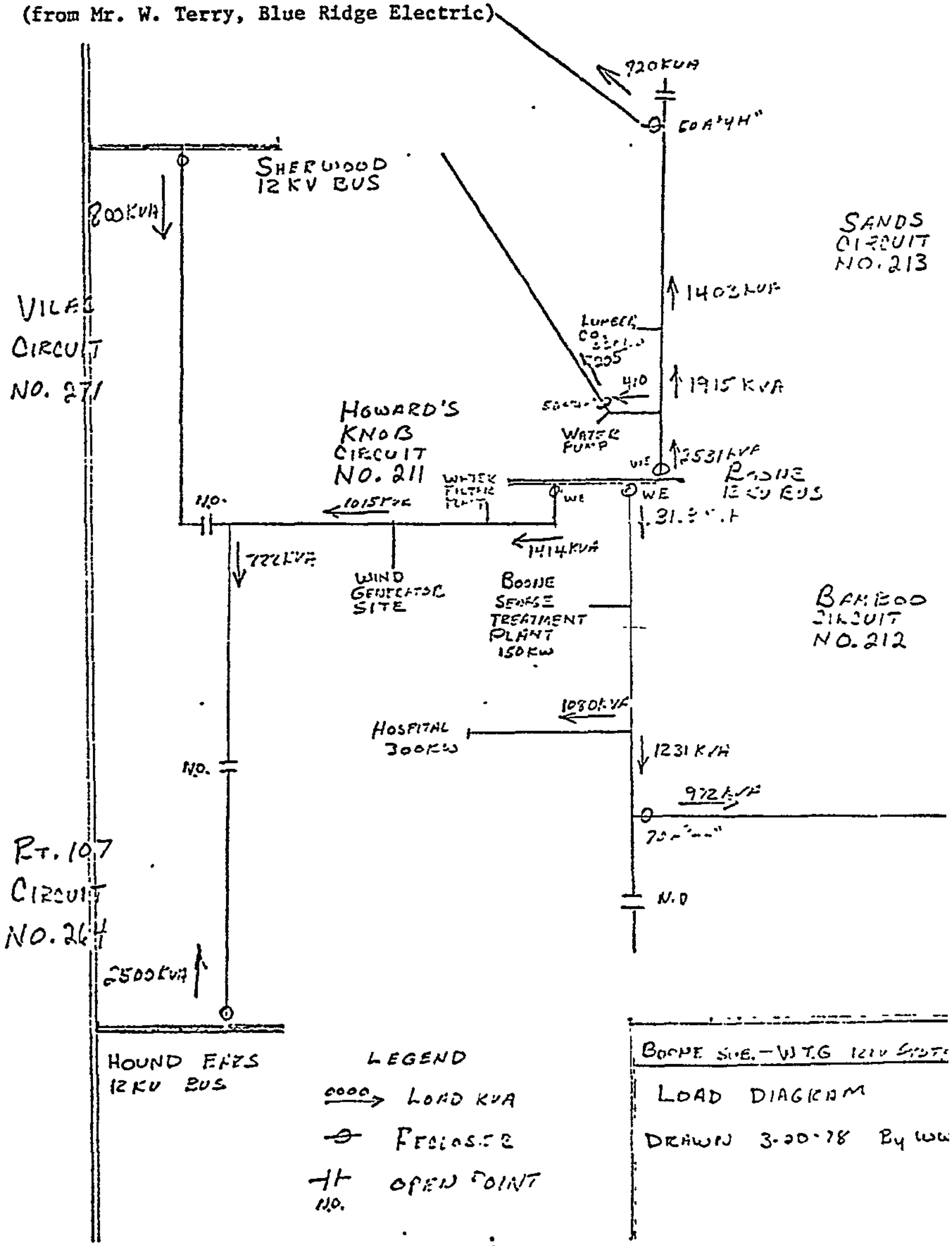
A.1 System Reactance Diagram

(from Mr. W. Terry, Blue Ridge Electric)



A.2 System Load Diagram

(from Mr. W. Terry, Blue Ridge Electric)



A.3 <u>Voltage Sensitive Load Characteristics</u>
(From Mr. W. Terry, Blue Ridge Electric)

1. <u>Water Filter Plant</u>: The plant consists of motors as follows:

    3 - 60 HP
    2 - 75 HP
    1 - 10 HP
    1 - 3 HP
    5 - 1 HP

   The plant's function is to filter and treat water for use on the campus of Appalachian State University. Undervoltage tripping is used on the main breaker to protect the plant from undervoltages. The main breaker trips instantaneously when the voltage dips below 80 volts. Operation is restored manually after an undervoltage trip.

2. <u>Water Pump</u>: This load consists of the following motors:

    1 - 200 HP
    1 - 1 1/2 HP

   These pumps are used to provide supplementary water to the reservoir for the water filter plant in Item I. The motors are put into and removed from service manually and used only occasionally. This plant also has undervoltage protection that operates instantaneously when the voltage drops below 80 volts.

3. <u>Lumber Company</u>: The largest motor at this location is 100 HP. It is one of many that contribute to a demand of 430 KVA. The motors are used to power saws and other woodworking, gluing and drying equipment. The dry kilns operate twenty-four hours a day and have fans with magnetic controllers that must be manually reset after an outage or severe voltage dip.

4. <u>Boone Sewage Treatment Plant</u>: The following motors are used at this location.

    4 - 20 HP
    1 - 15 HP
    1 - 5 HP

At any given time, the maximum load could consist of 2 - 20 HP, 1 - 15 HP, and the 5 HP motors. This plant has no undervoltage tripping, but it is subject to controller dropout during extreme voltage dips. This plant is manned twelve hours Monday through Saturday and eight hours on Sunday.

5. <u>Hospital</u>: The peak load of 300 KW for this load occurs during the air conditioning season. However, the total load is made up of many small loads as lighting, cooking, and medical instruments such as X-Ray. The hospital is presently undergoing an expansion program which will increase its maximum demand to an estimated 1200 KW. This new load will consist of the following:

    Two central water chillers at 419 KVA each.
    Food preparation equipment.
    X-Ray equipment.

Large motors consist of the following:

    1 - 125 HP
    1 - 100 HP
    4 - 30 HP
    2 - 25 HP
    2 - 20 HP
    4 - 15 HP

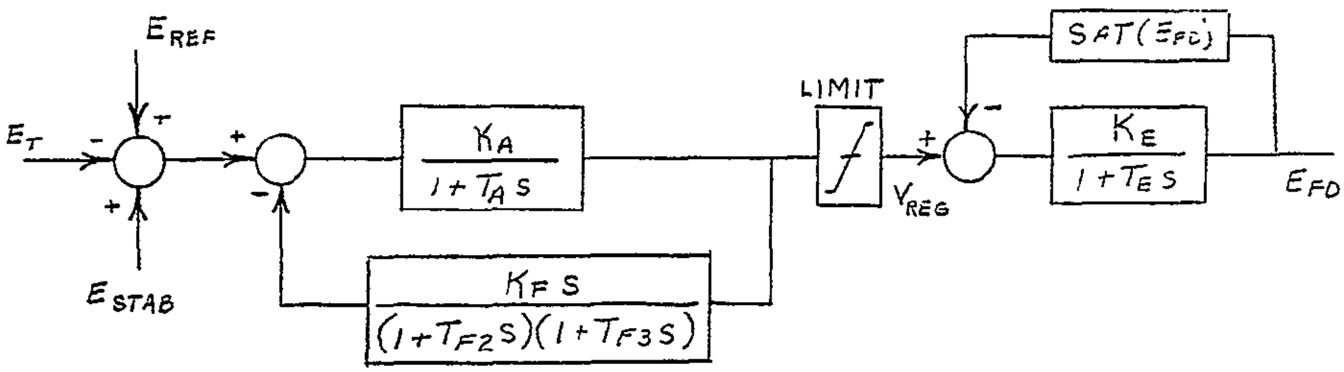
Extreme voltage dips or momentary outages can cause the emergency generator to start, boiler and pump motors to drop out, and terminate telephone conversations. The effect of voltage dips on medical instruments is uncertain; however, some lab instruments that are computer controlled would be sensitive to voltage dips.

A.4 Generator Electrical Data:

KVA Rating	1875.
X_{ℓ}	.11795
Ra	.0227
X_d	2.54
X_d'	.304
X_d''	.22
T_{do}'	3.34
T_{do}''	.0318
X_q	1.57
X_q'	.341
X_q''	.22
T_{qo}'	.1649
T_{qo}''	.003

All impedances in per unit on 1875 KVA base.

A.5 Excitation System Data



K_A = 975 pu/pu T_A = .26 sec.

K_E = .86 T_E = .26 sec.

K_F = .08 T_{F2} = 1.5 sec.

LIMIT = $\pm$ 19. pu T_{F3} = .26 sec.

A.6 Power System Stabilizer Data

$$\underset{\text{HUB SPEED}}{\omega_R} \longrightarrow \boxed{\frac{K_{STAB}\, s}{1 + T_W s}} \longrightarrow \boxed{\frac{(1 + T_1 s)(1 + T_3 s)}{(1 + T_2 s)(1 + T_4 s)}} \longrightarrow \boxed{\frac{1 + \frac{2\zeta_1}{\omega_1} s + \frac{1}{(\omega_1)^2} s^2}{1 + \frac{2\zeta_2}{\omega_2} s + \frac{1}{(\omega_2)^2} s^2}} \longrightarrow E_{STAB}$$

K_{STAB} = 100 pu/pu　　T_W = 10. sec.

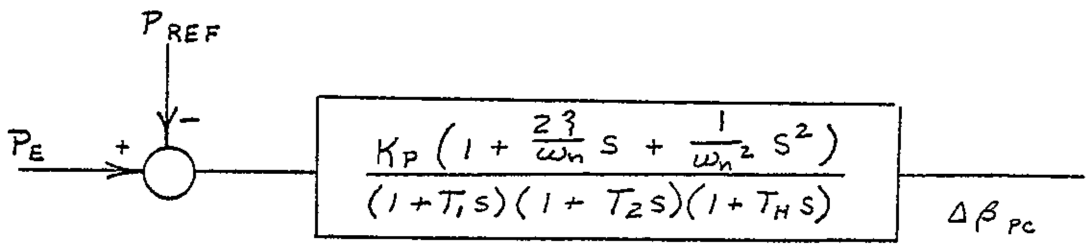
T_1 = .17 sec.　　T_2 = .017 sec.

T_3 = .17 sec.　　T_4 = .017 sec.

ω_1 = 21. rad/sec.　　ζ_1 = .1

ω_2 = 21. rad/sec.　　ζ_2 = .6

A.7 Power Regulator Data



Parameters for original system:

K_P = 75 deg/pu $\qquad$ T_H = .2 sec.

T_1 = 10 sec. $\qquad$ T_2 = .1 sec.

ζ = .2 $\qquad$ ω_n = 2.6 rad/sec.

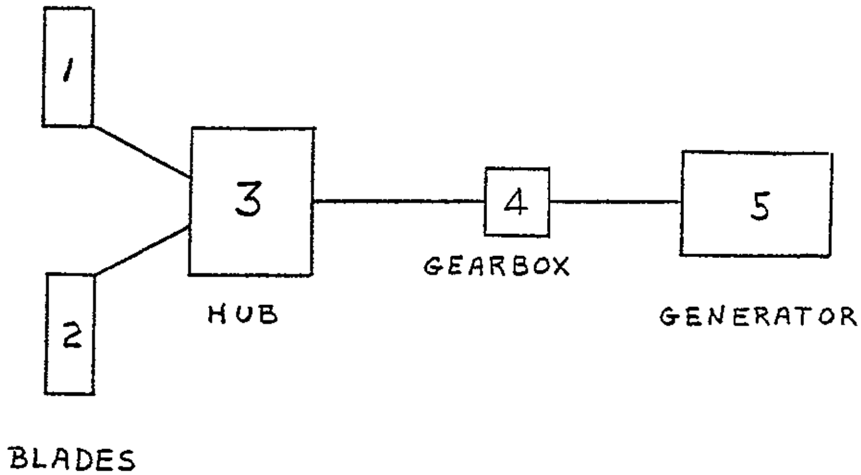
Recommended parameters for Boone site:

K_P = 75 deg/pu $\qquad$ T_H = .2 sec.

T_1 = 5.55 sec. $\qquad$ T_2 = .1 sec.

ζ = .2 $\qquad$ ω_n = 2.1 rad/sec.

A.8 Torsional System Data



I_1 = 28.96 lb ft sec^2 = .744 pu

I_2 = 29.70 lb ft sec^2 = .763 pu

I_3 = 779.6 lb ft sec^2 = 20.04 pu

I_4 = 5.829 lb ft sec^2 = .144 pu

I_5 = 50.5 lb ft sec^2 = 1.298 pu

K_{13} = 2859. lb ft/rad = .390 pu

K_{23} = 2859. lb ft/rad = .390 pu

K_{34} = 10284. lb ft/rad = 1.402 pu

K_{45} = 22000. lb ft/rad = 2.999 pu

C_{13} = 167.1 lb ft sec/rad = 4.296 pu

C_{23} = 167.1 lb ft sec/rad = 4.296 pu

All values referenced to 1800 rpm system.

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16. Abstract

The MOD-1 program is being conducted in six phases: Analysis and Preliminary Design (culminating in a Preliminary Design Review), Detail Design (ending with a Final Design Review), Fabrication and Assembly, System Testing, Site Preparation, and Installation and Checkout.

This report is the Appendix (Vol. II) of the Final Report (Vol. I) which describes the results of the first two phases; that is, activities leading to the completion of Detail Design. The Appendix contains supporting analyses and back-up material too voluminous for the Final Report.

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