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STATIC AND DYNAMIC ANALYSIS OF A 1150 MW TURBINE-GENERATOR SYSTEM
Part I: Static Analysis

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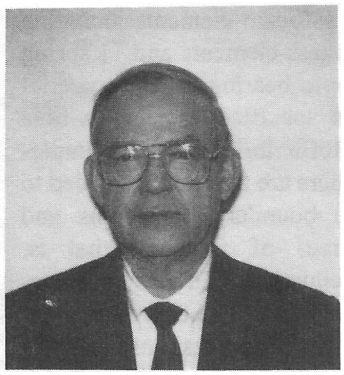

Edgar Gunter is a Professor of Mechanical and Aerospace Engineering at the University of Virginia. He received a Ph.D. degree in Engineering Mechanics in 1965 from the University of Pennsylvania. He has worked in the field of vibrations of rotating machinery and fluid bearings for over 30 years and has written over 150 technical papers and reports on various aspects of the dynamics of rotating machinery, fluid film bearings and balancing. For the past 6 years he has been working extensively with finite element analysis of rotating machinery on the microcomputer. He has had extensive experience with various types of vertical main recirculation pumps using both fixed and tilting pad bearings. Dr. Gunter has worked with utilities on various problems connected with high-pressure seal injection pumps, turbine-generators and fans.

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Jim Henderson is an Engineer for the Southern California Edison Co. at the San Onofre Nuclear Generating Station (SONGS) in San Clemente, California.

He has been dynamically monitoring the 1150 MW SONGS turbine-generators since 1983. He was involved with the initial field balancing operations following the Unit 3 Commercial Run, and has performed single and multi-plane trim balance shots.

Jim designed the customized shaft absolute vibration monitoring system using Bently dual probes and XY vibration monitors on the turbines, and also directed the SCE specification and procurement of the first hard bearing, transportable 250-ton-slow speed rotor balance machine sold by Schenck-Treble.

ABSTRACT

This paper deals with the determination of the static deflections, shaft stress distribution, and bearing reactions for a 1150 MW 11-bearing nuclear turbine-generator system. Computer models for this large and complicated system were developed using transfer matrix and finite element methods for the static and dynamic analysis. The effects of various vertical bearing alignments on the bearing reactions and shaft stress distribution were computed and analyzed by using finite element methods. The fluid film bearing stiffness and damping coefficients and bearing pressure profiles were computed for a range of bearing loads including foundation flexibility for a range of operating parameters. Bearing reaction loads are estimated based upon the observed hydrodynamically generated jacking oil hole pressures measured at the various bearings at running speed.

1 INTRODUCTION

This paper presents the static analysis for a large 1150 MW 11-bearing nuclear turbine-generator system, which weighs approximately 1.6 million lb (727,300 Kg) and is 2,427 in (61.65 M) long. It is composed of a high-pressure turbine, three low-pressure turbines, a generator and an exciter, which is one of the largest nuclear turbine-generator units in the world. The system has a normal operating speed of 1,800 RPM. Figure 1 shows the schematic diagram of the entire turbine-generator span.

In order to compute the critical speeds and unbalance response of a turbine-generator system, the bearing reaction loads must be known to compute the fluid film bearing characteristics. The bearing reactions are also highly dependent upon the initial vertical bearing alignment used. It is also necessary to evaluate the rotor static deflection due to the gravitational sag so that the shaft stress distribution may be determined in order to locate possible regions of high stress.

The problem of computing the rotor static deflection and bearing reactions represents a statically indeterminant system similar to a beam on an elastic foundation. A finite element model of the turbine-generator system was employed, and an iterative procedure was used to determine the bearing reactions and the shaft stress distributions for the various vertical bearing alignment cases.

A cubic spline curve fit for the 11 bearing deflections was found to significantly minimize shaft stresses and prevent excessive bearing loads. There are various cubic spline or catenary alignment curves that may be selected depending upon the specified initial offset of the outboard HP and the exciter bearings. The various bearing stiffness and damping coefficients were calculated for a range of loads and speeds. To accurately represent the bearing characteristics for the turbine-generator, foundation or support stiffness has to be included. The effective bearing damping may be reduced by 80 to 90% due to foundation flexibility effects. The bearing pressure as a function of loading was also computed and bearing reaction loads were estimated from the observed bearing pressures monitored at the individual bearings during operation.

2 EFFECT OF VERTICAL BEARING ALIGNMENT

The turbine-generator system was simplified to a reduced mass and beam-type model using the MacNeal-Schwendler Finite Element MSC/PAL2 program (1). This model is composed of 88 beam elements including shear deformation, 58 point mass elements and 11 spring elements, to simulate isotropic bearings. The material properties of the shaft was specified as steel with a Young's modulus of E = 30E6 lb/in^2. Before implementing the static analysis, there are two things needed to be done: establishing the boundary conditions and specifying the active degrees of freedom, that is, performing Guyan Matrix reduction. This is necessary only for the dynamic analysis phase. The overall number of active degrees of freedom for forced response analysis of the system is limited to 150 by the current PAL2 program. For static analysis only, PAL2 may handle 12,000 degrees of freedom.

The finite element model generated was used for the static deflection or gravitational sag analysis and also for dynamical analysis of the turbine-generator. The finite element method is superior to the transfer matrix method for the determination of the static deflection of the shaft. In the finite element procedure, initial vertical displacements of the bearing pedestals may be specified. In the original alignment of the 11-bearing turbine-generator, initial vertical offsets are specified for the various bearings.

The gravitational sag and bearing reactions of the shaft were computed for three cases with various initial vertical offsets. In the first case, no initial vertical offset of the bearing pedestals was specified. In the second case, the vertical alignment provided by the manufacturer was employed. In the third case, an optimized cubic spline curve fit was used to minimize the shaft stresses and bearing reactions.

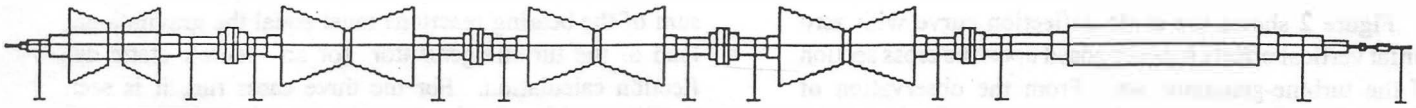

Figure 1 Schematic Outline of Turbine-Generator Systems.

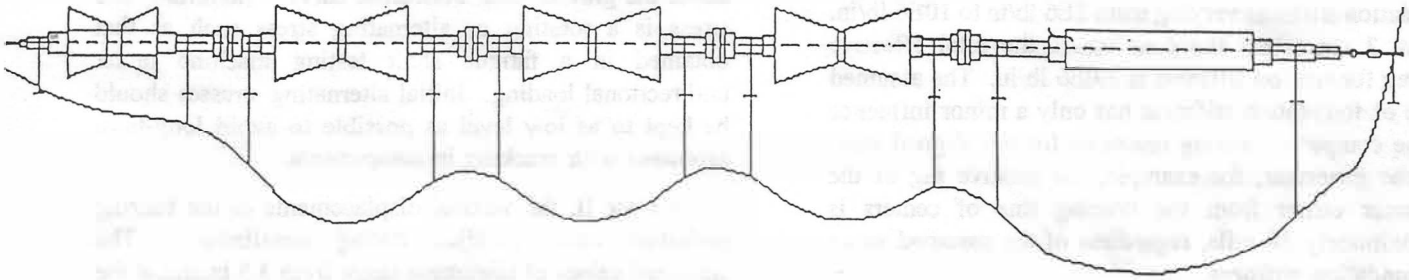

Figure 2 Static Deflection of Turbine-Generator With Zero Initial Vertical Offset.

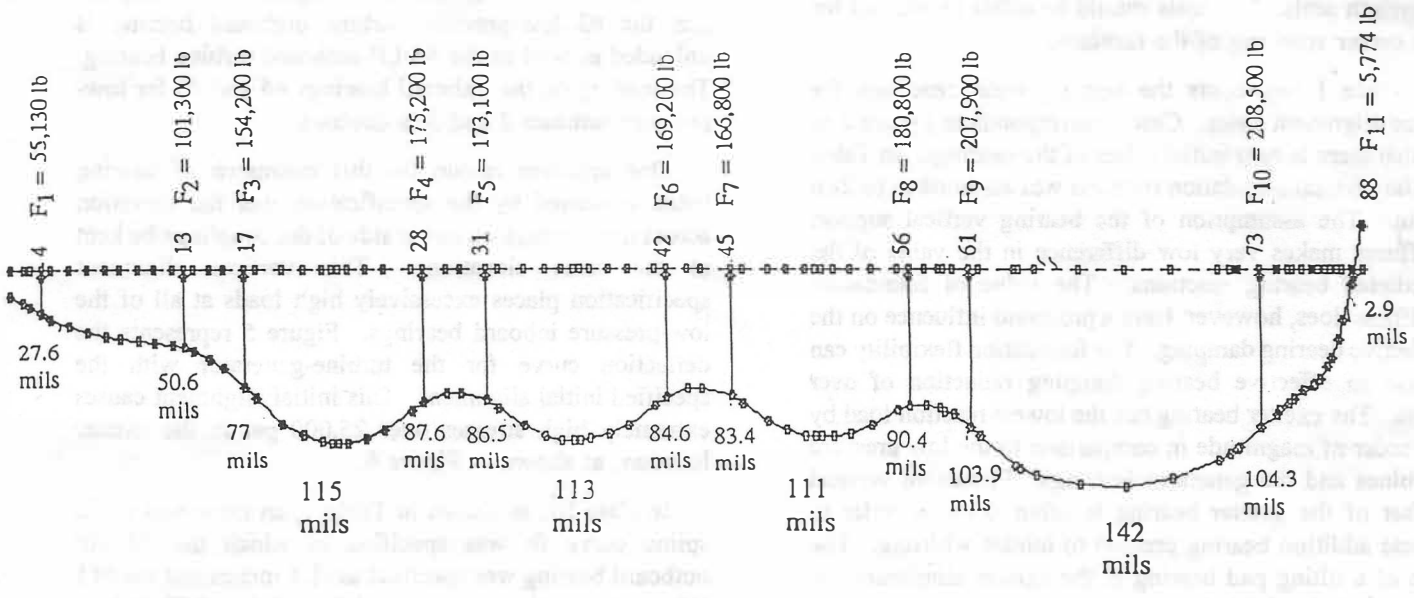

Figure 3 Static Deflection of the Turbine-Generator Showing Displacements and Reaction Forces.

Figure 2 shows the static deflection curve with zero initial vertical offsets superimposed upon the cross section of the turbine-generator set. From the observation of Figure 2, it is seen that the maximum displacements occur at the center span of the generator and at the centers of the low-pressure turbines. Figure 3 shows the initial and final static deflection curve of the nuclear turbine-generator for the case of zero initial vertical offset. The points along the curve represent the finite element nodes. Labeled on the figure are the bearing reactions and the bearing and center span displacements. The maximum static displacement is 142 mils and occurs at the center of the generator. The generator bearing displacements are approximately 104 mils. The shaft static deflections and bearing reactions were computed for foundation stiffness varying from 2E6 lb/in to 10E6 lb/in. Figure 3 represents the case where the total effective bearing foundation stiffness is 2.0E6 lb/in. The assumed value of foundation stiffness has only a minor influence on the computed bearing reactions for the aligned case. For the generator, for example, the relative sag of the generator center from the bearing line of centers is approximately 38 mils, regardless of the assumed value of foundation stiffness.

The average static deflection of the low-pressure turbines is 113 mils. The static sag of the low-pressure turbines is approximately 24 mils below the bearing centerline. This gravitational sag of 24 mils for the low-pressure turbine centers should be taken into consideration in the initial alignment of the turbines and the labyrinth seals. The seals should be offset to account for the center span sag of the turbines.

Table 1 represents the bearing static reactions for three alignment cases. Case I corresponds to Figure 2 in which there is zero initial offset of the bearings. In Table 1, the vertical foundation stiffness was assumed to be 2E6 lb/in. The assumption of the bearing vertical support stiffness makes very low difference in the value of the predicted bearing reactions. The value of foundation stiffness does, however, have a profound influence on the effective bearing damping. The foundation flexibility can cause an effective bearing damping reduction of over 80%. The exciter bearing has the lowest reaction load by an order of magnitude in comparison to the low pressure turbines and the generator bearings. Excessive vertical offset of the exciter bearing is often done in order to create addition bearing preload to inhibit whirling. The use of a tilting pad bearing at the exciter eliminates the need to preload this bearing.

In order to check the accuracy of the finite element static deflection, the bearing reactions are computed. The sum of the bearing reactions must equal the gravitational load of the turbine-generator for an accurate static deflection calculation. For the three cases run, it is seen that the total bearing loads are in very close agreement to each other.

Figure 4 represents the Von Mises shaft distribution for the case of zero initial bearing alignment. For the case of static shaft displacements, the Von Mises and major shaft stresses are identical. The highest stresses occur at nodes 83 and 62. The highest stress occurs at node 83 which is outboard of the exciter bearing. The stress at this node is 5,090 psi. Node 62 is located to the right of the outboard generator bearing. The major stress at this location is 2,870 psi. The rotating shaft will rotate about the gravitational deflection curve. Therefore, this stress is a rotating or alternating stress such as that obtained in a fatigue shaft testing machine under undirectional loading. Initial alternating stresses should be kept to as low level as possible to avoid long-term problems with cracking in components.

In Case II, the vertical displacements of the bearing pedestals were specified during installation. The specified values of alignment range from 3.5 inches at the exciter to -1.4 inches at the #1 HP outboard bearing. The bearing reactions and shaft stresses were computed with the specified installation alignment values. From Table 1 for Case II it can be seen that with this specified initial alignment, the bearing reaction at the #2 HP inboard bearing has been increased by over 50%, while the #3 LP1 outboard bearing reaction is negative. This implies that the #3 low-pressure turbine outboard bearing is unloaded as well as the #5 LP outboard turbine bearing. The loading on the inboard bearings #6 and #8 for low-pressure turbines 2 and 3 is doubled!

One apparent reason for this mismatch of bearing loads is caused by the specification that the elevation across the bearings on either side of the couplings be kept at the same elevation. This vertical alignment specification places excessively high loads at all of the low-pressure inboard bearings. Figure 5 represents the deflection curve for the turbine-generator with the specified initial alignment. This initial alignment causes extremely high stresses over 25,000 psi in the exciter location, as shown in Figure 6.

In Case III, as shown in Table 1, an improved cubic spline curve fit was specified in which the #1 HP outboard bearing was specified as -1.4 inches and the #11 HP exciter bearing was reduced from 3.5 to 2.77 inches. By specifying the cubic spline curve fit for the vertical bearing alignment as shown in Figure 7, the bearing reaction loads are improved and the stresses are reduced

Table 1 Bearing Static Loading Due to Distributed Weight and Bearing Support Displacements For Three Vertical Alignment Cases

Cases	I Without Inclination σmax=5,090 lb/in Zero Alignment		II With Pre-Inclination σmax=25,680 lb/in Specified Alignment		III With Cubic Spline Fit σmax=4,390 lb/in Optimal Alignment	
Bearing Location	Brg. Vertical Displ. (in)	Bearing Force (lb)	Brg Vertical Displ. (in)	Bearing Force (lb)	Brg Vertical Displ. (in)	Bearing Force (lb)
#1HP Outboard	0.0	55,130	-1.4	79,190	-1.400	62,600
#2 HP Inboard	0.0	101,300	-1.3	159,700	-1.084	94,090
#3 LP1 Outboard	0.0	154,200	-1.3	-8,254	-0.925	151,800
#4 LP1 Inboard	0.0	175,200	-0.4	394,000	-0.400	175,000
#5 LP2 Outboard	0.0	173,100	-0.4	-4,729	-0.191	173,800
#6 LP2 Inboard	0.0	169,200	0.4	347,400	0.424	168,500
#7 LP3 Outboard	0.0	166,800	0.4	-31,470	0.655	166,100
#8 LP3 Inboard	0.0	180,800	1.3	388,400	1.300	179,400
#9 Gen. Inboard	0.0	207,900	1.3	47,280	1.520	214,100
#10 Gen. Outboard	0.0	208,500	2.5	214,800	2.500	206,900
#11 Exciter	0.0	5,774	3.5	11,330	2.771	5,432
	Total Brg Load	1,597,904		1,597,647		1,597,722

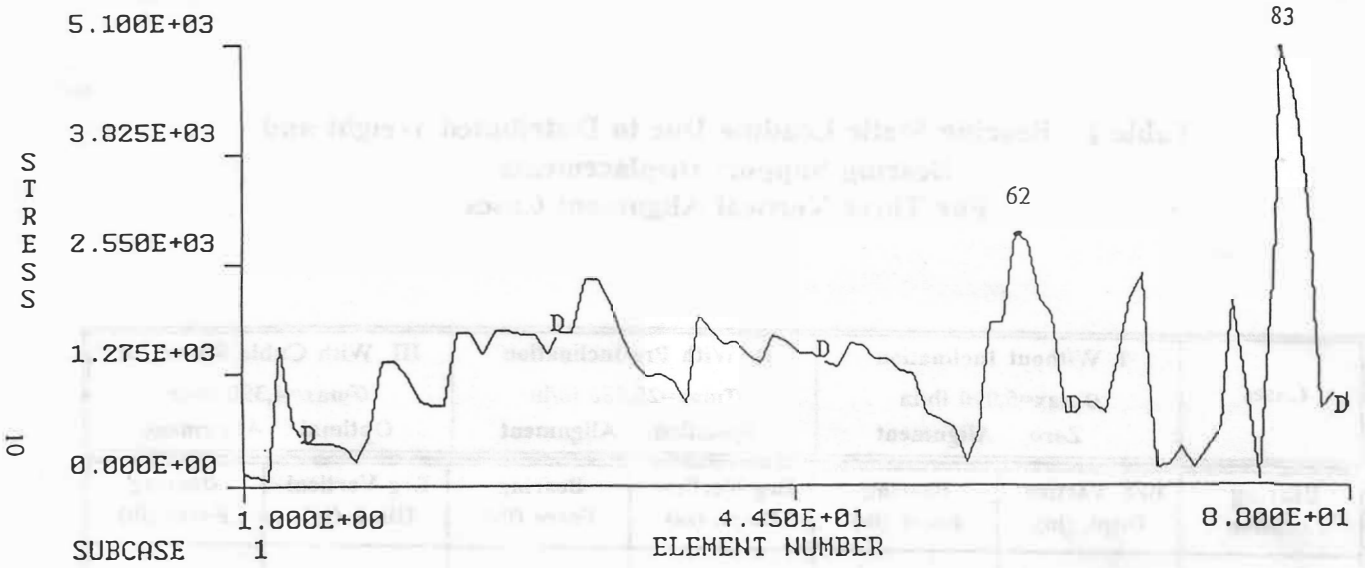

Figure 4 **Shaft Major or Von Mises Stress Distributions With Zero Initial Bearing Offset.**

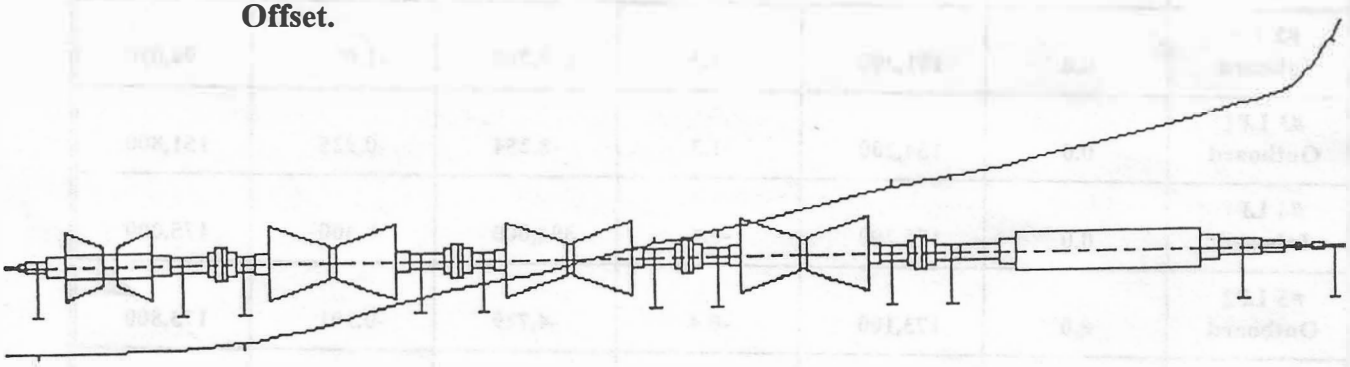

Figure 5 **Deflection Curve of Turbine-Generator With Specified Initial Alignment.**

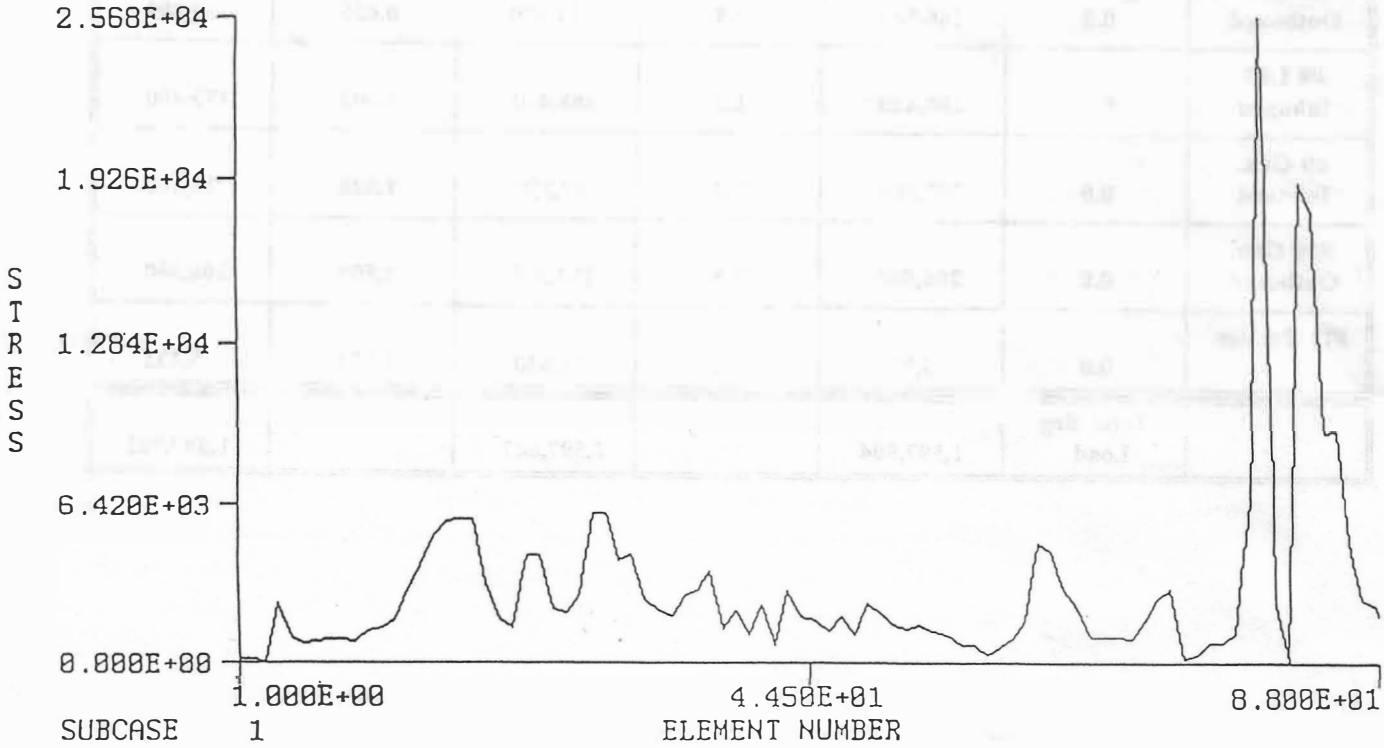

Figure 6 **Shaft Major Stresses With Specified Initial Alignment.**

below the case with 0 initial alignment. The maximum stress is reduced from 25,680 lb/in^2 to only 4,390 lb/in^2 as shown in Figure 8. It is also important to note that there is no reversal of curvature throughout the turbine-generator span. Figure 8 represents the major and minor shaft stress distribution for the Case III with the cubic spline displacement fit. Figure 9 represents a comparison of the deflection curve of the turbine-generator with the cubic spline curve fit as compared to the original alignment specifications.

It is clear from the above calculations that if the vertical bearing alignment curve is not accurately specified, then some bearings may be excessively loaded, such as bearings 4, 8, and 6 in Case II, while other bearings may be unloaded (a minus bearing reaction force means the bearing is unloaded), such as bearings 3, 5, and 7 in the same case, which could lead to shaft whirl. High stresses may result from improper vertical bearing alignment specifications which could eventually lead to shaft cracking. It is shown from Table 1 that the cubic spline curve fit is an improvement over the other two cases to reduce shaft stresses and prevent bearings from being over or underloaded. There are many spline or catenary deflection curves that may be chosen to minimize shaft stresses and bearing reactions depending upon assumed vertical offset end conditions.

3 HYDRODYNAMIC BEARING CHARACTERISTICS

The hydrodynamic bearing characteristics of the bearings for the turbine-generator and exciter were computed. The bearing stiffness and damping characteristics were computed over a range of speeds and loads. In addition to calculating the bearing stiffness and damping characteristics, the bearing pressure profile at various bearing loads was also computed by a two-dimensional finite element bearing program (3). Because of the high bearing loads, jacking oil is supplied at the bottom of each bearing. Upon reduction of speed, jacking oil under pressure is applied to each bearing to prevent bearing contact during shutdown. During operating conditions, the jacking oil supply hole located at the bottom of each bearing may be used to monitor the generated hydrodynamic bearing film pressure. An estimate of the reaction load at each bearing may be obtained from the hydrodynamic pressure measured at running speed based on the two-dimensional finite element solution of Reynolds equation.

Figure 10, for example, represents a typical low-pressure turbine with an arbitrary load of 200,000 lb. The bearing is operating at an eccentricity of .81 and an attitude angle of 33.7°. The maximum film pressure developed in the bearing is 1,074 psi. The interpolated pressure at the jacking oil hole is 817 psi.

Figure 11 shows the relationship between maximum film pressure and the bottom oil pressure at the operating speed of 1800 RPM assuming a viscosity of 1.5 microReyns. The pressure profiles were computed for the 11 bearings over a range of loads, and a plot was generated corresponding to the orifice oil pressure versus bearing load. Figure 12 represents the bearing load as a function of measured oil pressure for the various bearings. The generator bearings are the largest bearings and hence can carry the highest loads. The exciter bearing is much smaller and is much more lightly loaded.

Table 2 represents the measured orifice pressure on units 2 and 3 operating at 1800 RPM. For unit #2, the highest pressure is at the LP1 outboard bearing with 700 psi pressure. A cubic spline curve using MATLAB (4) was used to generate an analytic function for the prediction of bearing load as a function of orifice pressure. The predicted load corresponding to 700 psi for the LP1 outboard bearing is 174,700 lb. The lowest loads are predicted to be on the LP3 inboard and outboard bearings. These bearings only have a predicted load of 58,000 and 34,000 lb approximately. It would therefore appear that the LP3 bearings were not carrying sufficient loading, whereas the LP1 outboard bearing is carrying excessive loading for unit #2. In the case of unit #3, it appears that the LP3 outboard bearing is only carrying 47,920 lb. The vertical alignment of the bearing pedestals may be changed in order to increase or reduce the bearing loading.

From the observation of the measured orifice bearing pressures versus estimated loading for unit #2, it is apparent that there are some adjustments in the vertical alignment of the turbine-generator that would more uniformly distribute the bearing loadings. The LP1 outboard bearing is somewhat high and the loading could be reduced by dropping the bearing pedestal. Both the LP3 inboard and outboard bearing pedestals are not carrying their required loads. The LP3 bearing loads could be increased by raising the pedestals at these two bearing locations.

Experience has indicated that there is little change in the dynamic behavior of the turbine-generator with changes in the vertical alignment of the bearings. Changes in the vertical alignment change the bearing eccentricity ratio and hence the bearing stiffness and damping characteristics. Calculations of the operating eccentricity of these bearings indicate that these bearings are operating with eccentricities over 70%. For turbine-generators of this size, foundation flexibility must be

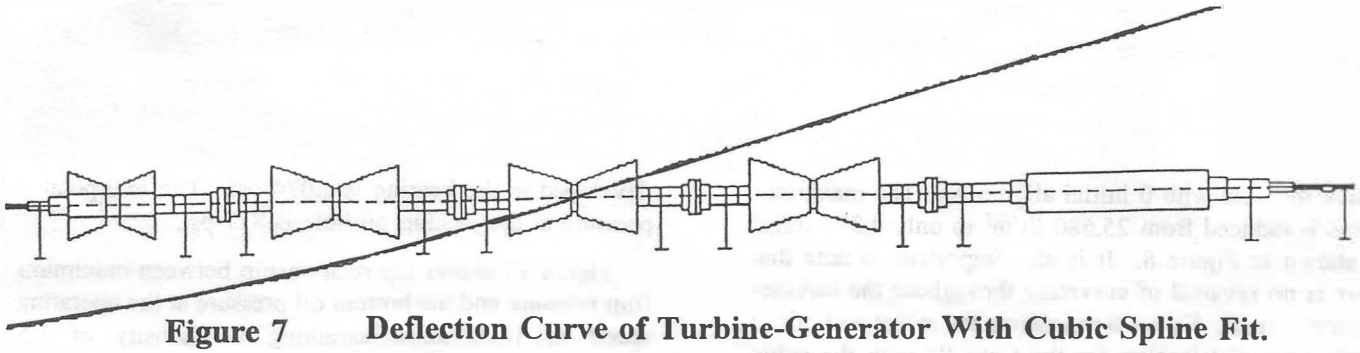

Figure 7 **Deflection Curve of Turbine-Generator With Cubic Spline Fit.**

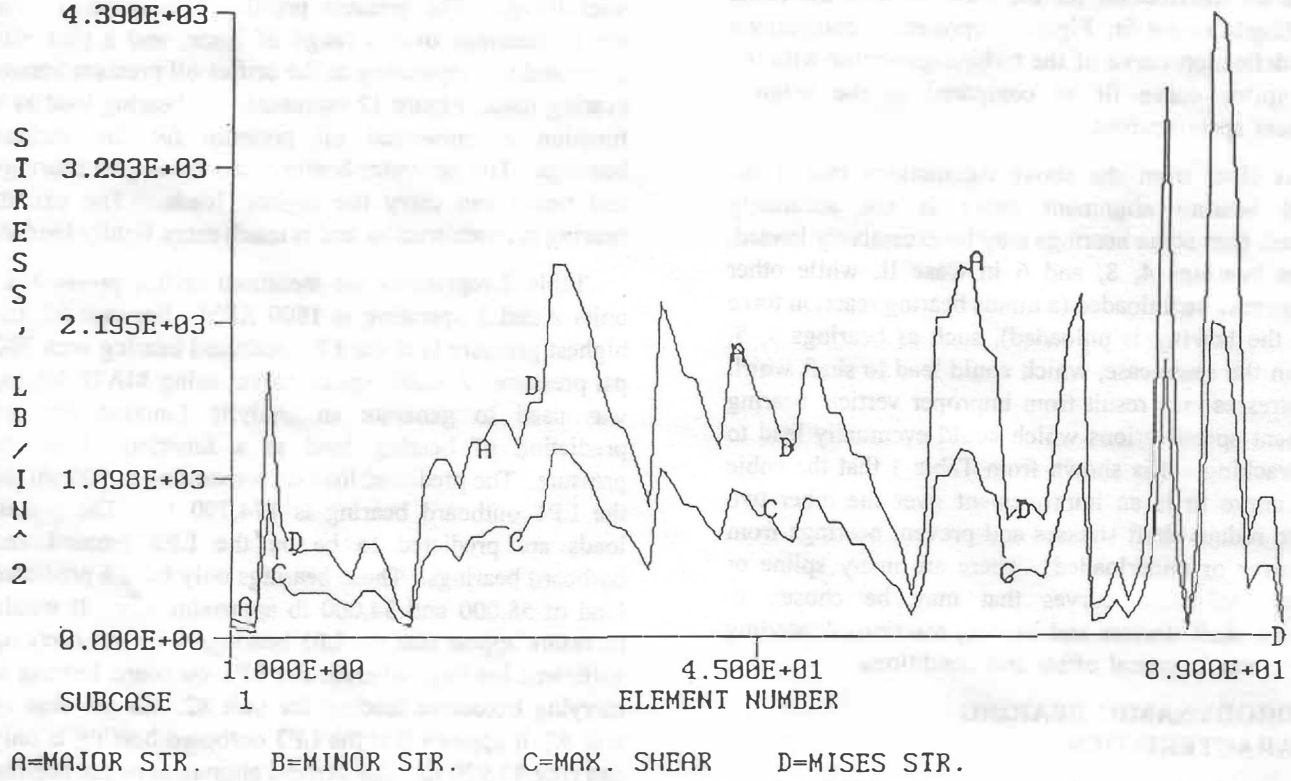

Figure 8 **Shaft Major and Minor Stresses With Cubic Spline Curve Fit.**

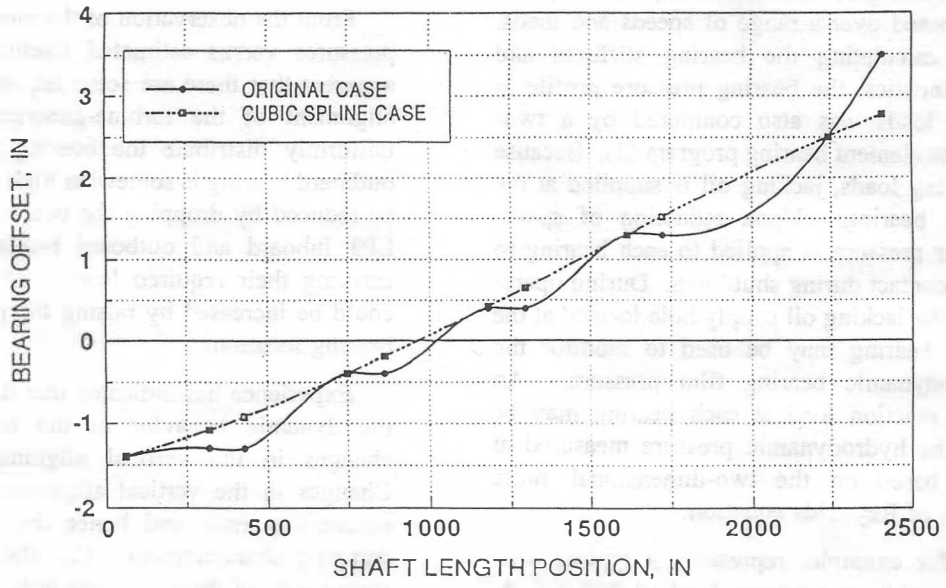

Figure 9 **Comparison of Initial Deflection Curves Between Original and Cubic Spline Curve Fit.**

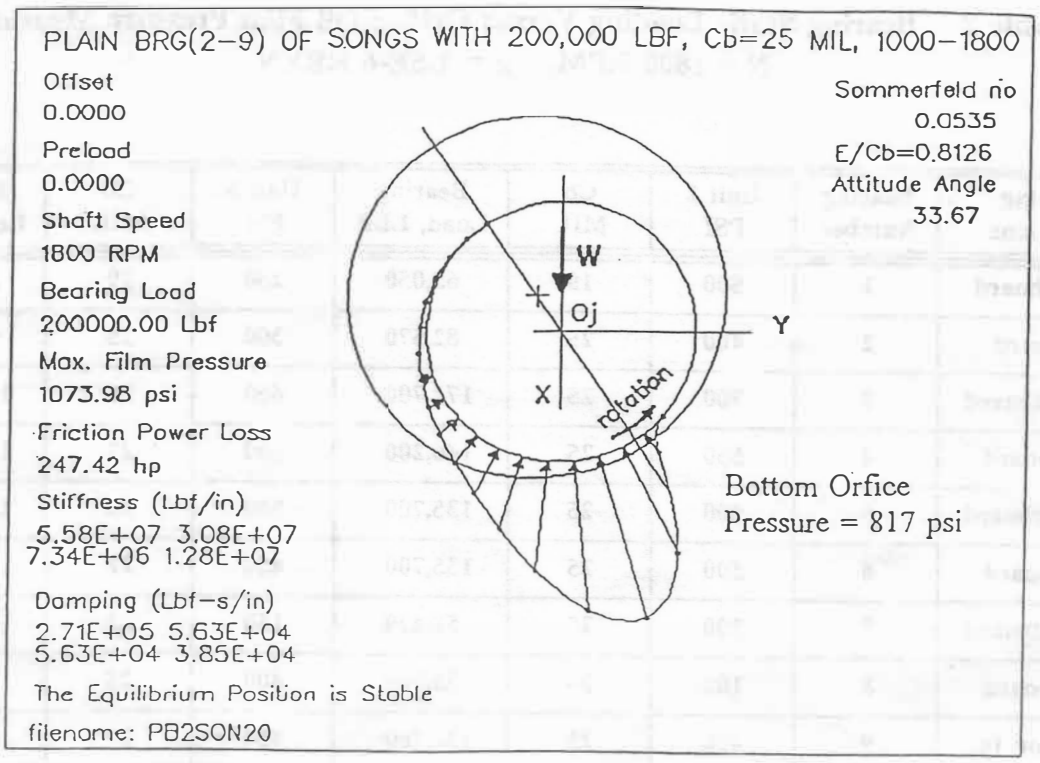

Figure 10 **Turbine Bearing Steady State Characteristics and Pressure Profile With 200,000 lb Load at 1,800 RPM.**

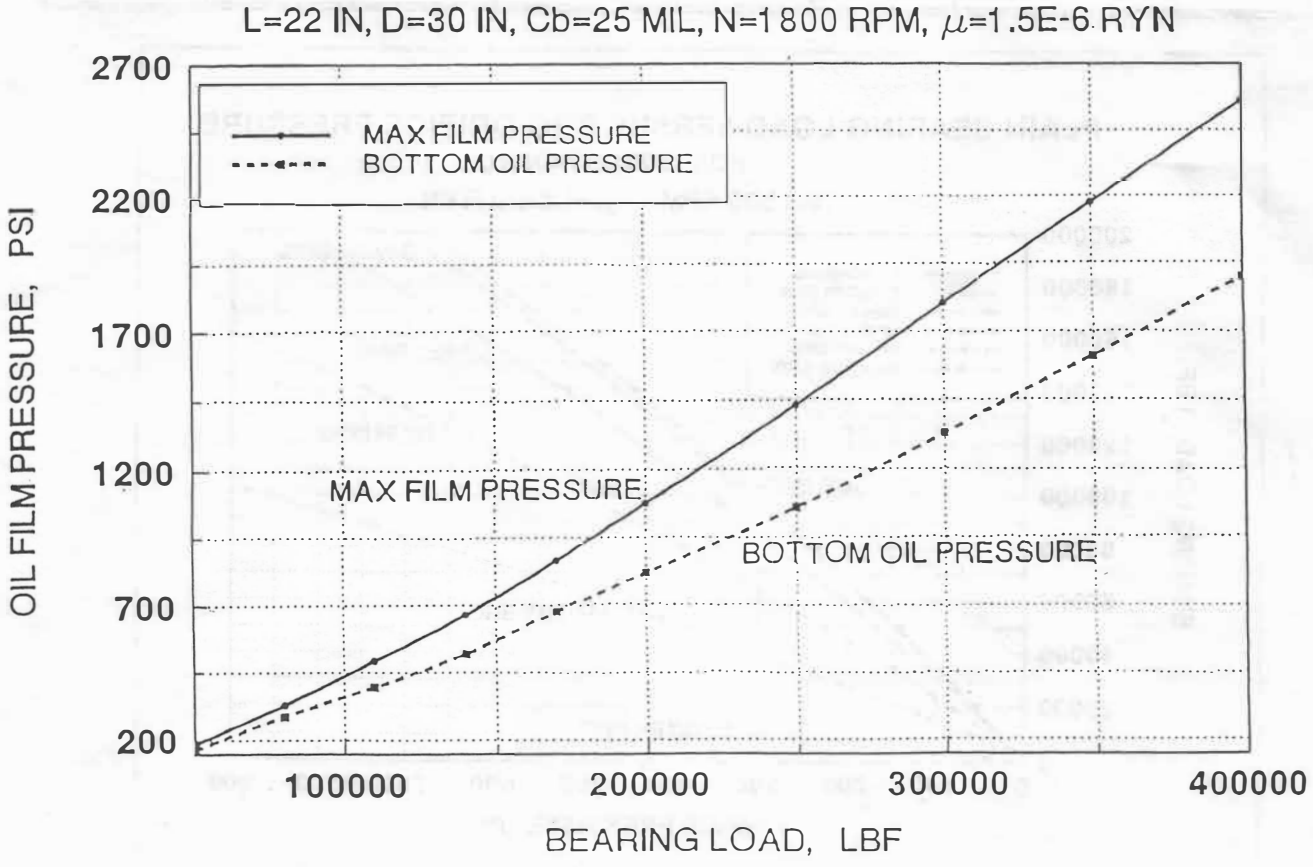

Figure 11 **Turbine Bearing Maximum Film Pressure and Bottom Oil Hole Pressure for Various Loads at 1,800 RPM.**

Table 2 Bearing Static Loading Versus Orifice Oil Film Pressure Measured
N = 1800 RPM, μ = 1.5E-6 REYN

Bearing Locations	Bearing Number	Unit 2 PSI	Cb MIL	Bearing Load, LBF	Unit 3 PSI	Cb MIL	Bearing Load, LBF
HP Outboard	1	500	19	63,050	250	19	34,200
HP Inboard	2	400	25	82,570	300	25	64,220
LP1 Outboard	3	700	25	174,700	650	25	165,300
LP1 Inboard	4	550	25	146,200	500	22	136,700
LP2 Outboard	5	500	25	135,700	500	22	136,700
LP2 Inboard	6	500	25	135,700	450	22	124,800
LP3 Outboard	7	200	25	58,430	150	22	47,920
LP3 Inboard	8	100	20	33,700	400	25	111,300
Generator In.	9	400	25	122,700	450	20	142,800
Generator Out	10	450	20	142,800	200	20	69,070
Exciter	11	200	6.5	6,780	250	6.5	8,130
			Total	1,102,330		Total	1,041,140

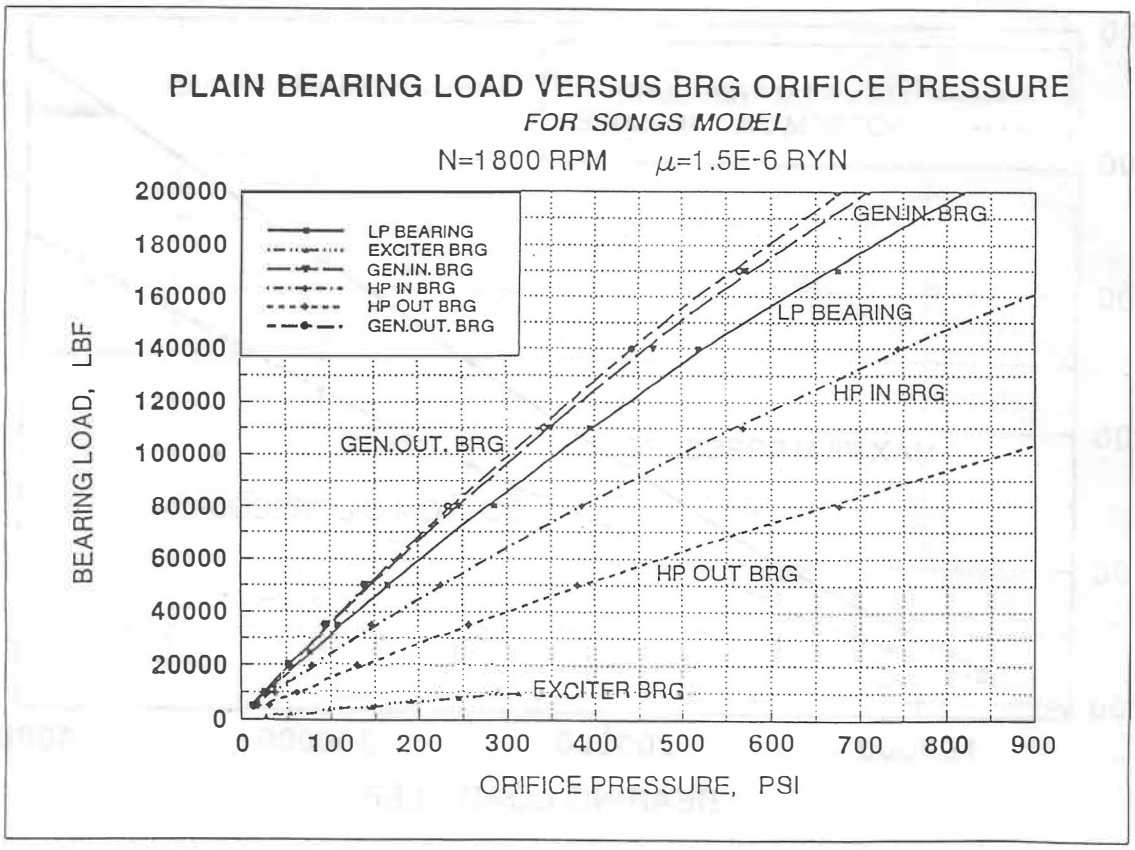

Figure 12 Bearing Vertical Static Loading vs Orifice Pressure for the T-G Bearings

considered when computing the effective bearing stiffness and damping characteristics or in performing a rotor dynamics unbalance response.

Table 3 represents the bearing stiffness and damping characteristics of a typical low-pressure turbine bearing with a 30-in. diameter and length of 18 in. The bearing has an assumed 100,000 lb vertical loading and is operating at an eccentricity ratio of .8. In this case, it is of interest to note that the stiffness in the vertical direction is approaching 37E6 lb/in. The damping in the vertical direction is approaching 100,000 lb-s/in. With these large values of stiffness and damping, the foundation flexibility cannot be ignored. The foundation flexibility governs the effective stiffness and damping obtained in the bearing.

In the second row, the bearing synchronous stiffness and damping coefficients are computed assuming circular orbiting (see Appendix A). The total dynamic stiffness in the vertical direction Z_{yys} including damping is 39E6 lb/in and Z_{xxs} is 5.6E lb/in for the horizontal direction. The total effective bearing stiffness of the combined bearing and foundation cannot exceed the stiffness of the foundation. When a vertical foundation stiffness of 4E6 lb/in assumed for the vertical direction and 2.0E6 lb/in for the horizontal direction, the effective vertical synchronous stiffness K_{yysf} is approximately 3.5E6 lb/in and the horizontal synchronous K_{xxsf} is 1.7E6 lb/in. Of greatest interest is the influence of the foundation flexibility on the effective damping. Notice that the synchronous vertical damping C_{yys} has been reduced from 73,000 lb-sec/in to C_{yysf} equals 578 lb-sec/in when foundation effects are included. The horizontal damping has been reduced from C_{xxs}= 29,000 lb-s/in to 3,000 lb-s/in. It is of interest to note that the effective horizontal damping is actually higher than the vertical damping. The equivalent bearing stiffness and damping characteristics, including foundation effects, are shown in Appendix A.

Table 4 represents a 30-in diameter bearing with a length of 22 inches and an external loading of 200,000 lb. In this case, the bearing stiffness and damping characteristics are higher than the case of the 18-in bearing with the 100,000 lb load as shown in Table 3. However, when the foundation flexibility is taken into consideration, the overall effective damping for this bearing is actually lower than the first case. The effective synchronous damping C_{yysf} is only 438 lb-sec/in for the vertical direction and C_{xxsf} is only 1,102 lb-sec/in in the horizontal direction.

4 CONCLUSIONS

This paper examines the influence of various types of vertical alignment procedures on the bearing reactions and stresses generated in a multi-bearing turbine generator. The study shows that a theoretical uniform or optical bearing alignment of a large turbine-generator is not optimum due to the gravitational sag between the spans. If all bearings are at the same elevation initially, then alternating shaft stresses will be developed between the bearing spans and couplings during operation.

To minimize the alternating shaft stresses, the bearing pedestals may be arranged along a catenary curve. This will result in the more uniform distribution of shaft stresses and bearing reaction forces. The study shows that if improper vertical alignment is made, then the bearings may be either excessively loaded, or could become completely unloaded, which could lead to self-excited whirl instability.

One of the objectives of the static analysis was to determine the bearing reaction forces. It is necessary to estimate the bearing reaction forces in order to calculate the hydrodynamic bearing characteristics which are required for dynamic analysis of the turbine-generator set. It was found that there is a good correlation between the measured bearing film pressure and the load on the bearing. It is therefore suggested that it may be possible to change alignment conditions while operating based on the observed bearing static pressure measurements.

It was also determined that foundation effects must be included in the computation of bearing characteristics for large turbine-generator sets. Foundation flexibility can cause over 90% reduction in the bearing effective damping. Therefore, dynamical analysis of large turbine-generator systems without considering foundation effects would be inaccurate and misleading.

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Table 3 Bearing Synchronous Stiffness and Damping Coefficients Bearing Parameters: Rigid Foundation

N=1,800 RPM Whirl Ratio=1.00
L=18.000 in. D=30.000 in. L/D=.600
Cbrg=25.00 mils Cradial=25.00 Cb/R=1.67 mils/in
Kxf=2,000,000 lb/in Kyf=4,000,000 lb/in
Cxf=0 lb-sec/in Cyf=0 lb-sec/in

Bearing Pedestal Weight=2,000 lb
Load Number = S*W = 8748 MU = 1.5 MicroREYNS

Sb	W (lb)	Eb	Kxx	Kxy	Kyx	Kyy (lb/in)* 10-6	Cxx	Cxy	Cyy (lb-s/in)* 10-3
.087	100,000.0	.80	6.32	- 1.47 -	16.83	29.75	21.85	- 37.58	99.00

Zxxs	Zyys Total Dynamic Stiffness	Kxxs	Kyys Synchronous Stiffness	Cxxs	Cyys Synchronous Damping
5,638,010	39,324,676	- 763,575	36,831,958	29,637	73,101

Total Dynamic Stiffness & Damping with Foundation Effects
Kxf = 2,000,000 lb/in Kyf = 4,000,000 lb/in

Zxxsf	Zyysf	Kxxsf	Kyysf	Cxxsf	Cyysf
1,801,135	3,496,314	1,708,531	3,494,618	3,025	578

Table 4 Bearing Synchronous Stiffness & Damping Coefficients With Rigid and Flexible Foundation

N=1,800 RPM Whirl Ratio=1.00
L=22.000 in. D=30.000 in. L/D=.733
Cbrg=25.00 mils Cradial=25.00 Cb/R=1.67 mils/in
Kxf=2,000,000 lb/in Kyf=4,000,000 lb/in
Cxf=0 lb-sec/in Cyf=0 lb-sec/in

Bearing Pedestal Weight=2,000 lb
Load Number = S*W = 10692 MU = 1.5 MicroREYNS

Rigid Foundations

Sb	W (lb)	Eb	Kxx	Kxy	Kyx	Kyy (lb/in)* 10-6	Cxx	Cxy	Cyy (lb-s/in)* 10-3
.053	200,000.0	.79	11.57	- 7.07 -	28.94	47.49	46.62	- 74.19	99.00

Zxxs	Zyys Total Dynamic Stiffness	Kxxs	Kyys Synchronous Stiffness	Cxxs	Cyys Synchronous Damping
16,036,828	67,914,021	- 2,414,488	61,473,741	84,114	153,155

Total Dynamic Stiffness & Damping with Foundation Effects
Kxf = 2,000,000 lb/in Kyf = 4,000,000 lb/in

Zxxsf	Zyysf	Kxxsf	Kyysf	Cxxsf	Cyysf
1,835,562	3,630,310	1,823,773	3,629,373	1,102	438

APPENDIX A

Bearing Synchronous Stiffness and Damping Coefficients Including Foundation Effects

The fluid film bearing characteristics are given by 8 bearing stiffness and damping coefficients. However, in most finite element programs such as MSC/PAL2, only 2 stiffness and damping coefficients per plane may be used. The eight bearing stiffness and damping coefficients may be reduced to four synchronous stiffness and damping coefficients for the horizontal and vertical directions by the assumption of a circular synchronous orbit. Reasonable results have been obtained by this approximation in comparison to the use of the full eight bearing coefficients as long as the rotor orbits are not highly elliptical.

The synchronous stiffness and damping coefficients are given by

$$\left.\begin{aligned} K_{xxs} &= K_{xx} + \omega C_{xy} \\ K_{yys} &= K_{yy} - \omega C_{yx} \end{aligned}\right\} \text{ Synchronous Stiffness} \qquad \text{(A1)}$$

$$\left.\begin{aligned} C_{xxs} &= C_{xx} - \frac{K_{xy}}{\omega} \\ C_{yys} &= C_{yy} + \frac{K_{yx}}{\omega} \end{aligned}\right\} \text{ Synchronous Damping} \qquad \text{(A2)}$$

Where ω = shaft speed, in rad/sec.

The synchronous stiffness and damping coefficients can be combined into a dynamical stiffness or impedance which is useful in critical speed calculations.

$$\left.\begin{aligned} Z_{xs} &= \sqrt{K_{xxs}^2 + (\omega C_{xxs})^2} \\ Z_{ys} &= \sqrt{K_{yys}^2 + (\omega C_{yys})^2} \end{aligned}\right\} \qquad \text{(A3)}$$

The values of Z_{xs} and Z_{ys} are the effective bearing dynamical stiffness coefficients for the X and Y directions respectively, that should be used for the computation of the rotor undamped critical speeds. The use of only K_{xx} and K_{yy} for critical speed calculation is in considerable error.

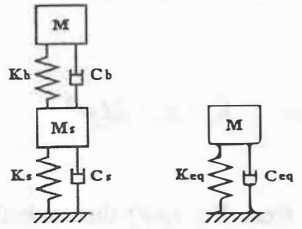

Figure A1. Two-mass system and single-mass equivalent.

The bearing stiffness and damping may be combined to the foundation to compute an equivalent bearing stiffness and damping including foundation effects. In a general finite element rotor dynamic code with foundation effects, this approximation of effective stiffness and damping is not necessary. Figure A1, for example, represents a simple bearing on a flexible support. The system is then reduced to an effective speed dependent stiffness and damping coefficient by assuming synchronous motion. In the following derivation, the bearing characteristics are assumed to be the previously determined synchronous coefficients.

$$M\ddot{x} + C_b(\dot{x} - \dot{x}_s) + K_b(x - x_s) = f(t) \qquad \text{(A4)}$$

$$M_s\ddot{x} + C_s\dot{x}_s + K_s x_s + C_b(\dot{x}_s - \dot{x}) + K_b(x_s - x) = 0 \qquad \text{(A5)}$$

Assuming synchronous forced response at a frequency of ω.

$$x = Xe^{j\omega t} \quad x_s = X_s e^{j\omega t} \quad f = Fe^{j\omega t} \qquad \text{(A6)}$$

Equation (A4) becomes

$$\begin{aligned} &(K_b - M\omega^2 + j\omega C_b)X - (K_b + j\omega C_b)X_s \\ F = &K_b X - K_b X_s + j\omega(C_b X - C_b X_s) - M\omega^2 X \end{aligned} \qquad \text{(A7)}$$

Solving for X_s from Eq. (A8) then substituting into Eq. (A7) yields the complex relationship between total bearing displacement X and the displacement X_s of the support as follows

$$(\hat{K}_s + K_b + j\omega C_s) X_s = (K_b + j\omega C_b) X$$

$$\text{where} \quad \hat{K}_s = K_s - M_s\omega^2 \qquad \text{(A8)}$$

Solving for X_s from Eq. (A8) then substituting into Eq. (A7) yields

$$F = \left(K_b - \frac{K_b^2 + j\omega C_b K_b}{\hat{K}_s + K_b + j\omega(C_b + C_s)} \right) X$$

$$+ j\omega \left(C_b - \frac{K_b C_b + j\omega C_b^2}{\hat{K}_s + K_b + j\omega(C_b + C_s)} \right) X - M\omega^2 X \qquad \text{(A9)}$$

The equivalent system is expressed as

$$F = K_{eq} X + j\omega C_{eq} X - M\omega^2 X \qquad \text{(A10)}$$

Therefore

$$K_{eq} = \frac{K_b(\hat{K}_s + j\omega C_s)}{\hat{K}_s + K_b + j\omega(C_s + C_b)} \qquad \text{(A11)}$$

$$C_{eq} = \frac{C_b(\hat{K}_s + j\omega C_s)}{(\hat{K}_s + K_b) j\omega(C_s + C_b)} \qquad \text{(A12)}$$

$$K_{eq} + j\omega C_{eq} = \frac{(K_b + j\omega C_b)(\hat{K}_s + j\omega C_s)}{(\hat{K}_z + j\omega C_s) + (K_b + j\omega C_b)} \qquad \text{(A13)}$$

Rationalizing and simplifying Eq. (A13) yields the total or effective bearing support stiffness K_{eq} as follows

$$K_{eq} = \frac{\hat{K}_s K_b(\hat{K}_s + K_b) + \omega^2(K_b C_s^2 + \hat{K}_s C_b^2)}{(\hat{K}_s + K_b)^2 + \omega^2(C_s + C_b)^2} \qquad \text{(A14)}$$

The effective bearing support damping is given by

$$C_{eq} = \frac{K_b^2 C_s + \hat{K}_s^2 C_b + \omega^2 C_s C_b (C_s + C_b)}{(\hat{K}_s + K_b)^2 + \omega^2(C_s + C_b)^2} \qquad \text{(A15)}$$

For the case where the foundation damping C_s is negligible, the equivalent damping C_{eq} reduces to

$$C_{eq} = \frac{\hat{K}_s^2 C_b}{(\hat{K}_s + K_b)^2 + (\omega C_b)^2} \qquad \text{(A16)}$$

The occurrence of the term ωC_b in the denominator of Eq. (A16) can lead to a substantial reduction in the effective damping in the system. The resulting equation for effective damping in particular demonstrates the profound effect of foundation flexibility on the reduction of the overall effective damping acting at a bearing. It should also be observed that the damping decreases with speed and vanishes when the support is in resonance when

$$\hat{K}_s = K_s - M_s\omega^2 = 0$$

Therefore, a cracked pedestal or foundation support which places the supporting structure in resonance at running speed can lead to excessive vibrations in the turbine-generator. The resonance in the supporting structure causes a loss of damping at that bearing location. The unit will have high vibrations and will be extremely difficult to balance because of the lack of damping at that location. The foundation must be welded or repaired in order to successfully balance the rotor. A change out in the bearing type will also prove to be useless.