HYDRODYNAMIC JOURNAL BEARING CHARACTERISTICS FOR SONGS 1150 MW TURBINE-GENERATOR

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HYDRODYNAMIC BEARING CHARACTERISTICS OF SONGS 1150 MW TURBINE-GENERATOR

1. Bearing Design, Parameters, and Static Loading

The SONGS 1150 MW turbine-generator has 11 fluid film journal bearings. The bearings are labeled starting with the outboard bearing at the high-pressure turbine, which will be referred to as bearing number 1, to the bearing number 11 which is the exciter bearing. Table 1 shows the bearing location for the 11 bearings and gives the nominal bearing diameter, length, diametral bearing clearance, and average bearing static loads. The average bearing diametral clearance used in the bearing calculations is shown in Table 1. The HP and LP turbine bearings, in general, are designed with approximately 2 mils/in diametral clearance.

In order to determine the bearing stiffness and damping characteristics, the nominal loads on the bearings must be determined. The determination of the bearing loads was an extensive project involving the assumption of the initial vertical alignment of the bearings and the nominal bearing stiffnesses. The MSC/PAL2 finite element program has the unique ability that the turbine-generator static bearing loads may be computed with assumed initial vertical bearing alignment specifications.

Table 2 shows the bearing reactions and maximum stresses for the various alignment configurations calculated. Table 2 represents four cases of various vertical bearing alignments specified for the turbine-generator. The first case is a theoretical alignment with zero vertical offset of the bearings. The second alignment configuration was specified by SCE as a proposed alignment. This alignment results in some bearings being overloaded and others having a negative or upload. The third alignment case is a catnary curve specified by SCE with the #6 bearing taken as the zero point. This current alignment gives a very uniform distribution of the bearing loads. However, the shaft stress at the exciter location is 15,840 lb/in^2. The fourth case is a theoretical cubic spline fit for optimal alignment to minimize the stresses throughout the exciter and turbine-

TABLE 1 BEARING DESIGN CHARACTERISTICS

FOR SONGS *1150 MW TURBINE-GENERATOR*

Bearing #	Location	D (In)	L (In)	Cd (Mil)	W Nominal (Lb)	P_{avg} Loading (PSI)
1	HP Outboard	20	16	38	60,000	188
2	HP Inboard	30	18	51	100,000	185
3	LP1Outboard	30	22	55	150,000	227
4	LP1 Inboard	30	22	49	175,000	265
5	LP2Outboard	30	22	58	175,000	265
6	LP2 Inboard	30	22	61	175,000	265
7	LP3Outboard	30	22	56	175,000	265
8	LP3 Inboard	30	22	48	175,000	265
9	Gen. Inb.	30	24	56	210,000	292
10	Gen. Outb.	30	24	44	210,000	292
11	Exciter	10	7	13	9,000	129

TABLE 2 BEARING STATIC LOADING DUE TO DISTRIBUTED WEIGHT FOR VARIOUS ASSUMED VERTICAL BEARING SUPPORT DISPLACEMENTS

FOUR CASES COMPUTED FOR SONGS 1150 MW TURBINE-GENERATOR

N =1800 RPM, μ = 1.5E-6 REYN.

By Using MSC/PAL2 Finite Element Program

Four Cases	Without Inclination σ_{max} = 5,090 lb/in^2 Ideal Alignment		With Pre-Inclination σ_{max} = 25,680 lb/in^2 Specified Alignment		With Pre-Inclination σ_{max} = 15,840 lb/in^2 Specified Alignment		With Cubic Spline Fit σ_{max} = 4,390 lb/in^2 Optimal Alignment	
Bearing Location	Brg Supt Displ.(in)	Brg Force (lb)	Brg Supt Displ.(in)	Brg Force (lb)	Brg Supt Displ.(in)	Brg Force (lb)	Brg Supt Displ.(in)	Brg Force (lb)
#1 HP Outboard	0.0	55,130	-1.4	79,190	0.782	59,340	-1.4	62,600
#2 HP Inboard	0.0	101,300	-1.3	159,700	0.404	107,000	-1.0836	94,090
#3 LP1 Outboard	0.0	154,200	-1.3	-8,254	0.335	147,400	-0.925	151,800
#4 LP1 Inboard	0.0	175,200	-0.4	394,000	0.068	173,900	-0.4	175,000
#5 LP2 Outboard	0.0	173,100	-0.4	-4,729	0.014	171,400	-0.1912	173,800
#6 LP2 Inboard	0.0	169,200	0.4	347,400	0.0	170,000	0.4242	168,500
#7 LP3 Outboard	0.0	166,800	0.4	-31,470	0.047	173,000	0.6548	166,100
#8 LP3 Inboard	0.0	180,800	1.3	388,400	0.310	153,400	1.3	179,400
#9 Gen.Inborad	0.0	207,900	1.3	47,280	0.494	221,800	1.5203	214,100
#10 Gen.Outboard	0.0	208,500	2.5	214,800	1.452	211,200	2.5	206,900
#11 Exciter	0.0	5,774	3.5	11,330	2.163	9,265	2.7707	5,432
	Total Brg Load	1,597,904	Total Brg Load	1,597,647	Total Brg Load	1,597,705	Total Brg Load	1,597,722

generator. This alignment results in a maximum static stress of only 4,390 lb/in^2. Based on the alignment study, the nominal bearing loads were computed for the various bearings for use in the bearing coefficient calculations.

2. High Pressure Turbine Bearing Characteristics

Table 1 shows that the inboard and outboard bearings of the high-pressure (HP) turbine are of different sizes. The outboard bearing, for example, has a diameter of 20 in. and a length of 16 in. The bearing L/D ratio is 0.8 for the outboard bearing. The outboard bearing nominal diametral clearance Cd is 38 mils. This represents a bearing design clearance ratio of 1.9 mils/in. Typical bearing clearance designs are 1 to 2 mils diametral clearance per inch of shaft diameter. Table 1 shows that the calculated nominal loading on the HP outboard bearing is 60,000 lb over a projected surface of 320 in^2. This represents a nominal design loading of 188 psi.

Table 3 represents the bearing Sommerfeld number S, bearing dimensionless eccentricity ratio and bearing coefficients for various values of bearing loading. The bearing coefficients were computed with the DYROBES-BECOEF finite element program. In Table 3, it is seen that the HP outboard bearing is operating at an eccentricity ratio of 74% of the bearing clearance at 1,800 RPM with a 50,000 lb bearing load. Operation above 70% eccentricity ratio represents a heavily loaded bearing. In Table 3, for the bearing stiffness and damping coefficients, the X direction is the direction of the applied load. The maximum stiffness of the bearing in the vertical direction is over 12E6 lb/in. With a bearing stiffness value this high, the foundation effects must be taken into consideration since foundation stiffnesses are less than 10E6 lb/in. The maximum damping in the X direction, C_{xx}, is 59,000 lb-sec/in., and C_{yy} is 17,400 lb-sec/in. for the damping in the horizontal direction. The inclusion of foundation flexibility will also greatly reduce these values.

TABLE 3 HP OUTBOARD BEARING STIFFNESS AND DAMPING COEFFICIENTS FOR VARIOUS LOADS AT 1,800 RPM

D = 20 in, L = 16 in, Cd = 38 mils, μ = 1.5 microreyns

Load (lb)	S (DIM)	ε (DIM)	Kxx* (lb/in)	Kxy* (lb/in)	Kyx* (lb/in)	Kyy* (lb/in)	Cxx* (lb/in)	Cxy* (lb/in)	Cyx* (lb/in)	Cyy* (lb/in)	P (psi) Orifice
5,000	.798	.220	0.369	1.35	-0.835	0.543	14.1	2.89	2.89	9.54	29.6
10,000	.399	.375	0.940	1.92	-0.837	1.04	19.3	5.51	5.51	11.2	61.0
20,000	.199	.547	2.69	3.47	-0.646	2.00	33.5	10.6	10.6	14.1	130.0
35,000	.114	.671	6.71	6.27	-0.03	3.36	59.1	17.5	17.5	17.4	257.0
50,000	.080	.737	12.4	9.40	0.979	4.61	86.1	22.6	22.6	19.0	380.5
80,000	.050	.811	28.0	15.9	3.73	6.84	142.0	30.6	30.6	21.3	676.0
110,000	.036	.850	49.0	23.7	7.26	9.25	204.0	37.8	37.8	23.2	966.0

K* = K X E-6, C* = C X E-3

TABLE 4 HP INBOARD BEARING STIFFNESS AND DAMPING COEFFICIENTS FOR VARIOUS LOADS AT 1,800 RPM

D = 30 in, L = 18 in, Cd = 51 mils, μ = 1.5 microreyns

Load (lb)	S (DIM)	ε (DIM)	Kxx* (lb/in)	Kxy* (lb/in)	Kyx* (lb/in)	Kyy* (lb/in)	Cxx* (lb/in)	Cxy* (lb/in)	Cyx* (lb/in)	Cyy* (lb/in)	P (psi) Orifice
5,000	1.75	0.17	0.280	1.28	-0.884	0.439	13.3	2.31	2.31	9.81	17.9
10,000	0.875	0.30	0.692	1.68	-0.869	0.835	16.8	4.26	4.26	10.7	36.7
20,000	0.437	0.464	1.84	2.83	-0.824	1.63	27.1	8.6	8.6	13.6	78.1
35,000	0.25	0.59	4.39	4.96	-0.552	2.75	46.2	14.7	14.7	17.0	147.0
50,000	0.175	0.66	7.87	7.23	-0.039	3.73	56.0	20.0	20.0	19.0	225.0
80,000	0.109	0.74	16.8	12.4	1.26	5.71	112.0	29.3	29.3	23.0	385.0
110,000	0.08	0.789	29.0	17.2	3.22	7.2	152.0	33.6	33.6	23.6	569.0

K* = K X E-6, C* = C X E-3

In Table 3 are also recorded the bearing fluid film pressures generated at the oil supply hole located at the bottom of the journal. These values are estimated from the relationship of the maximum bearing pressure to the pressure generated at the bottom along the vertical axis. The maximum oil film pressure is always displaced or offset from the vertical axis in the direction of rotation. Figure 1 represents the bearing pressure generated at 1,800 RPM with a load of 20,000 lb. In Fig. 1 is shown also the bearing characteristics and the critical journal weight for stability. If the effective mass acting at this location exceeds this value, then the bearing may become unstable.

Figure 2 represents the HP1 bearing with an external loading of 50,000 lb. In this case, the bearing eccentricity ratio is 0.74. For journal bearings that operate with eccentricity ratios exceeding 0.7, the bearing is completely stable and free from half-frequency whirling. The bearings used for the HP and LP turbines may go unstable (oil film whirl) in the event that the vertical bearing loading drops to a low value due to improper vertical bearing alignment.

Table 4 represents the high-pressure inboard bearing. This bearing has a larger shaft diameter of 20 in. and a length of 18 in. The nominal vertical bearing load is approximately 100,000 lb. The inboard HP projected pressure loading is 185 psi. This compares favorably with the projected pressure loading of the outboard bearing of 188 psi.

Table 4 represents bearing stiffness and damping characteristics of the inboard HP bearing for various loads at 1,800 RPM. The inboard bearing is operating at the high eccentricity of approximately .78 at a loading of 110,000 lb. Operation at this high eccentricity levels results in a perfectly stable bearing. Figure 3 represents the pressure profile for the HP2 inboard bearing with a vertical load of 80,000 lb. This figure shows that the maximum film pressure is 466 psi and that the friction horsepower loss at 18,00 RPM is 166 HP. Figure 4 represents the bearing pressure profile for 110,000 lb vertical loading. The bearing is operating at an eccentricity of .79 as shown in Fig. 4. The total

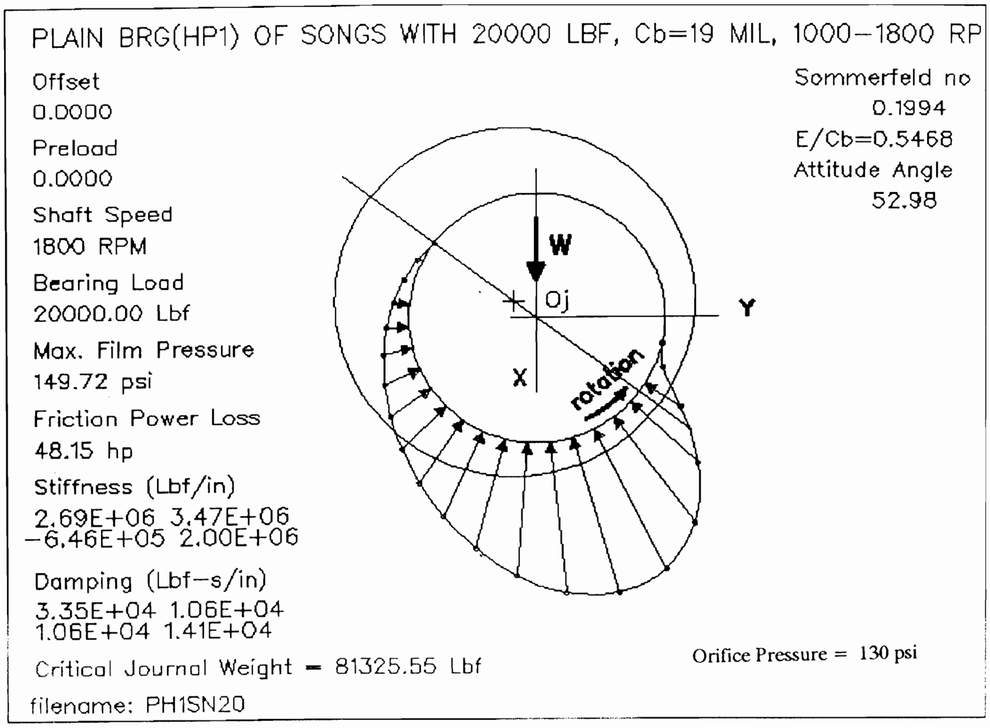

Figure 1 HP Outboard Bearing Pressure Profile for 20,000 lb Vertical Loading
L = 16 in, D = 20 in, Cd = 38 mil

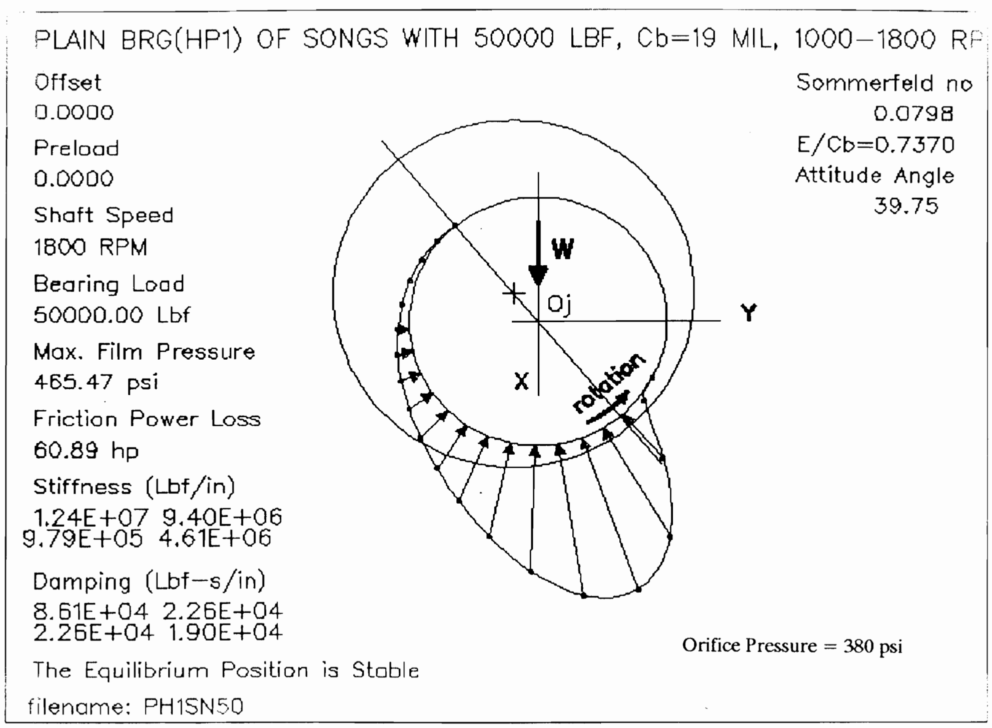

Figure 2 HP Outboard Bearing Pressure Profile for 50,000 lb Vertical Loading
L = 16 in, D = 20 in, Cd = 38 mil

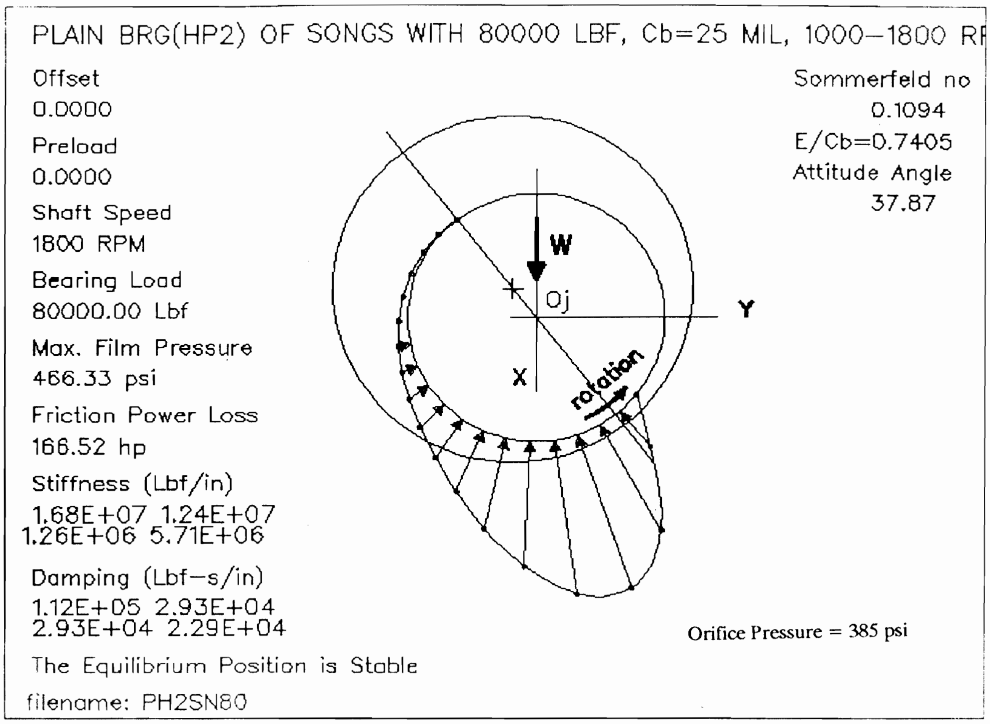

Figure 3 HP Inboard Bearing Pressure Profile for 80,000 lb Vertical Loading
L = 18 in, D = 30 in, Cd = 51 mil

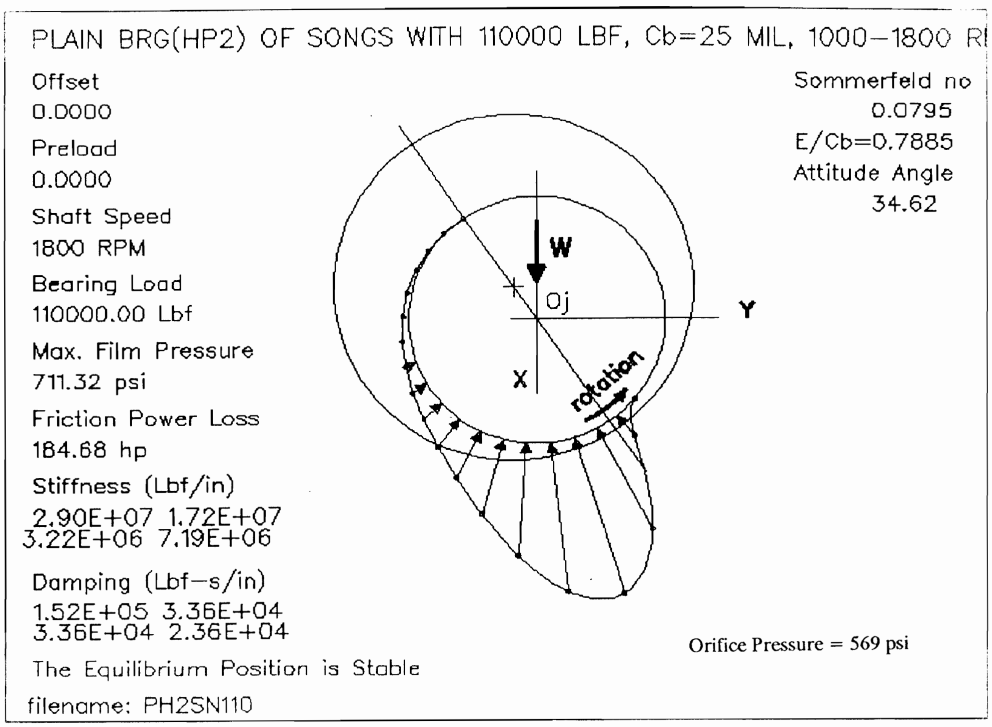

Figure 4 HP Inboard Bearing Pressure Profile for 110,000 Lb Vertical Loading
L = 18 in, D = 30 in, Cd = 51 mil

bearing load is being carried by a bearing arc of approximately 120 degrees. The maximum film pressure is 711 psi and the friction HP power loss has increased to 185 HP.

3. Low Pressure Bearing Characteristics

The design of the bearings for the three low-pressure turbines are very similar. The bearing diameters are 30 in and the lengths are 22 in. The measurement of the nominal clearances shows that there is a range of bearing clearance from a minimum of 48 mils diametral for the #8 bearing, which is the third inboard bearing, to 61 mils for bearing #6, which is the second LP turbine inboard bearing. The nominal bearing load for the LP bearings is 175,000 lb. Table 5 represents the low-pressure bearing stiffness and damping characteristics for a range of loads at 1,800 RPM. The estimated orifice pressure for each load is also shown in Table 5.

Figure 5 represents the pressure profile on the low-pressure turbine bearing at 1,800 RPM with a vertical bearing load of 170,000 lb. The specified bearing radial clearance Cb is 22 mils, which corresponds to a diametral bearing clearance of 44 mils. The maximum film pressure is 793 psi and the estimated orifice pressure is 648 psi. Figure 6 is similar to Fig. 5, except that the bearing loading has been increased to 200,000 lb. The bearing eccentricity increases from 0.75 to 0.77. The orifice pressure increases to 768 psi.

4. Generator Bearing Coefficients

The generator bearings have a diameter of 30 in. and a length of 24 in. The nominal bearing diametral clearance varies from 44 mils on the outboard bearing to 56 mils on the inboard bearing. The nominal bearing vertical loading is 210,000 lb which represents a projected bearing loading of 292 psi. The generator bearings are the most highly loaded bearings in the system. The characteristics of the generator bearings were computed for a range of loads and several clearance values. The values of generator bearing stiffness and damping are shown in Table 6.

TABLE 5 LOW PRESSURE BEARING STIFFNESS AND DAMPING COEFFICIENTS FOR VARIOUS LOADS AT 1,800 RPM

D = 30 in, L = 22 in, Cd = 44 mils, μ = 1.5 microreyns

Load (lb)	S (DIM)	ε (DIM)	Kxx* (lb/in)	Kxy* (lb/in)	Kyx* (lb/in)	Kyy* (lb/in)	Cxx* (lb/in)	Cxy* (lb/in)	Cyx* (lb/in)	Cyy* (lb/in)	P (psi) Orifice
10,000	1.38	0.15	0.581	3.16	-2.19	0.99	33.2	5.38	5.38	24.1	28.9
20,000	0.69	0.28	1.45	3.94	-2.16	1.85	40.0	9.6	9.6	26.0	58.0
35,000	0.39	0.41	3.18	5.62	-2.03	3.13	55.5	16.0	16.0	29.6	105.0
50,000	0.276	0.50	5.32	7.70	-1.96	4.47	74.6	23.8	23.8	35.2	157.0
80,000	0.173	0.60	11.3	12.3	-1.07	6.85	116.0	35.8	35.8	40.7	267.0
110,000	0.126	0.67	18.9	17.5	-0.09	9.25	164.0	49.0	49.0	47.8	389.0
140,000	0.099	0.72	28.6	22.8	1.69	11.3	209.0	56.7	56.7	49.4	514.0

K* = K X E-6, C* = C X E-3

TABLE 6 GENERATOR BEARING STIFFNESS AND DAMPING COEFFICIENTS FOR VARIOUS LOADS AT 1,800 RPM

D = 30 in, L = 24 in, Cd = 50 mils, μ = 1.5 microreyns

Load (lb)	S (DIM)	ε (DIM)	Kxx* (lb/in)	Kxy* (lb/in)	Kyx* (lb/in)	Kyy* (lb/in)	Cxx* (lb/in)	Cxy* (lb/in)	Cyx* (lb/in)	Cyy* (lb/in)	P (psi) Orifice
10,000	1.17	0.158	0.489	2.73	-1.88	0.867	28.7	4.82	4.82	20.7	26.4
20,000	0.583	0.285	1.24	3.41	-1.86	1.61	35.0	8.47	8.47	22.4	53.1
50,000	0.233	0.509	4.63	6.65	-1.64	3.87	65.0	20.7	20.7	30.1	140.0
80,000	0.146	0.619	9.98	10.6	-0.772	5.92	101.0	30.7	30.7	34.4	246.0
110,000	0.106	0.687	17.5	14.7	0.746	7.59	135.0	36.0	36.0	35.1	350.0
140,000	0.0833	0.729	25.3	19.9	1.73	9.90	184.0	49.8	49.8	42.6	468.0
170,000	0.0686	0.761	35.2	25.0	3.46	11.8	230.0	58.1	58.1	45.4	571.0
200,000	0.0583	0.787	46.6	30.1	5.32	13.6	275.0	66.0	66.0	48.3	710.0

K* = K X E-6, C* = C X E-3

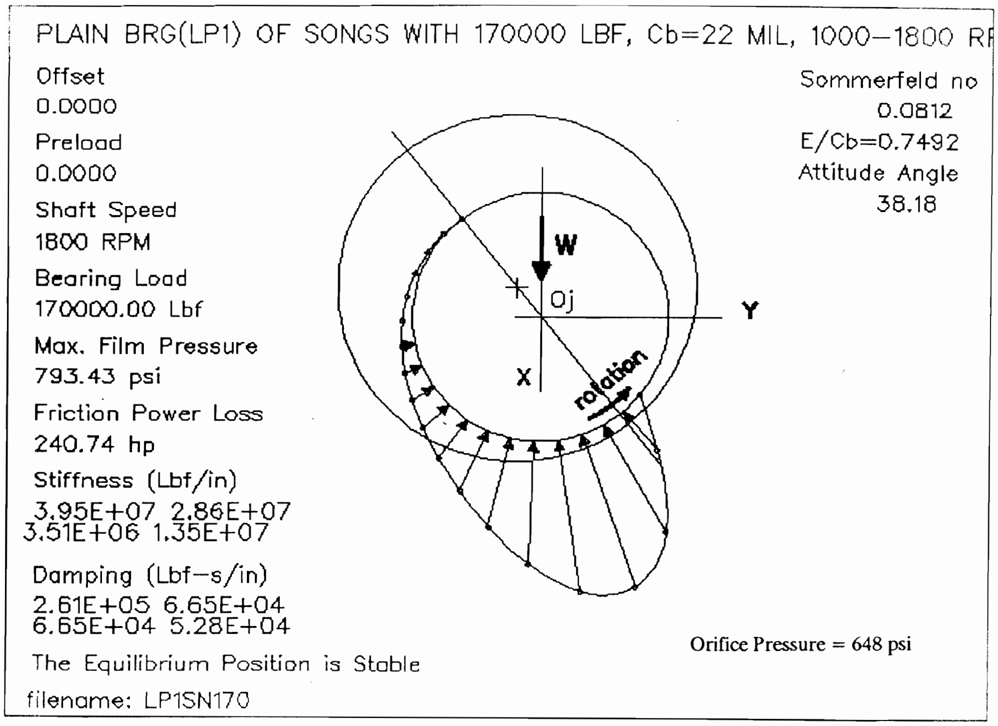

Figure 5 LP Bearing Pressure Profile for 170,000 lb Vertical Loading
L = 22 in, D = 30 in, Cd = 44 mil

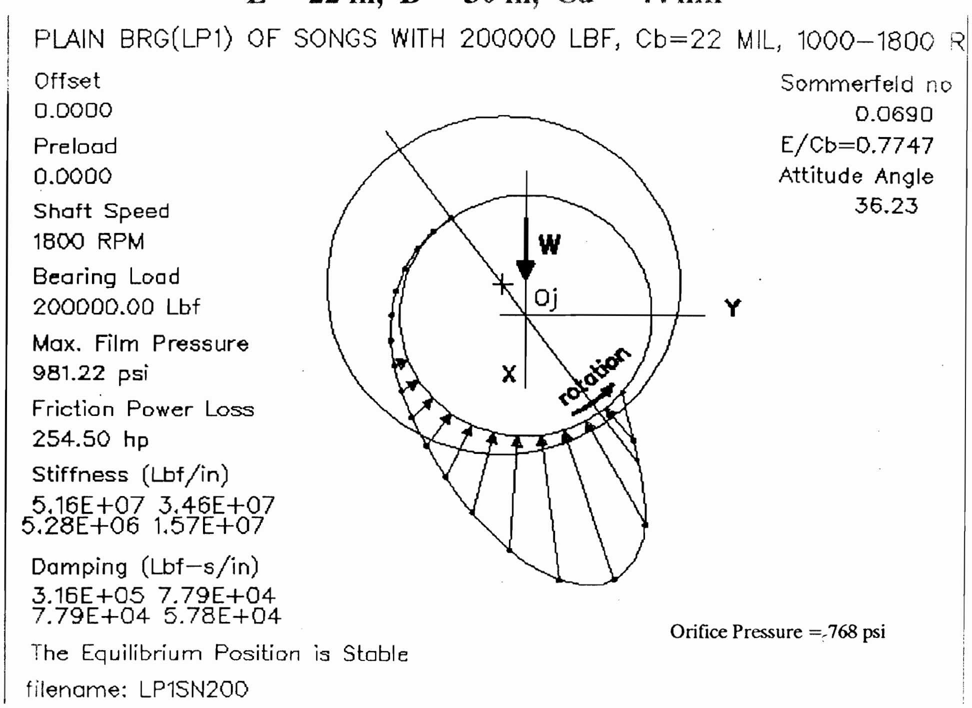

Figure 6 LP Bearing Pressure Profile for 200,000 lb Vertical Loading
L = 22 in, D = 30 in, Cd = 44 mil

5. Exciter Bearing Characteristics

The exciter bearing number 11 is the smallest bearing of the system. The bearing diameter is 10 inches, bearing length is 7 inches, and the nominal diametral bearing clearance is 13 mils. Table 7 represents the bearing stiffness and damping characteristics for a range of loads at 1,800 RPM. The design bearing load varies depending upon the amount of vertical alignment specified for the exciter end.

Table 7 shows a design load of 10,000 lb for the exciter bearing. This results in a bearing eccentricity of .6. The estimated bearing orifice pressure is 312 psi.

TABLE 7 EXCITER BEARING STIFFNESS AND DAMPING COEFFICIENTS FOR VARIOUS LOADS AT 1,800 RPM

D = 10 in, L = 7 in, Cd = 13 mils, μ = 1.5 microreyns

Load (lb)	S (DIM)	ε (DIM)	Kxx* (lb/in)	Kxy* (lb/in)	Kyx* (lb/in)	Kyy* (lb/in)	Cxx* (lb/in)	Cxy* (lb/in)	Cyx* (lb/in)	Cyy* (lb/in)	P (psi) Orifice
1,000	1.86	0.126	0.185	1.29	-0.937	0.347	13.6	1.93	1.93	10.2	27.0
2,000	0.932	0.234	0.455	1.52	-0.939	0.648	15.7	3.41	3.41	10.8	54.1
3,500	0.533	0.356	0.980	2.04	-0.908	1.09	20.3	5.61	5.61	12.0	98.3
5,000	0.373	0.443	1.62	2.70	-0.904	1.55	26.3	8.23	8.23	13.9	146.0
6,500	0.287	0.506	2.42	3.41	-0.822	1.98	32.8	10.5	10.5	15.2	191.0
8,000	0.233	0.555	3.48	4.11	-0.546	2.29	38.6	11.4	11.4	14.7	245.0
10,000	**0.186**	**0.604**	**4.81**	**5.28**	**-0.487**	**2.93**	**49.7**	**15.5**	**15.5**	**17.6**	**312.0**

K* = K X E-6, C* = C X E-3

6. Estimated Bearing Loading vs Orifice Pressure Measurements.

Experimental measurements have been obtained at the various orifice locations for the various bearings. From the computer predictions of the bearing pressure profile, an estimation of the orifice pressure at the bottom of the bearing can be made. For example, Fig. 7 represents a typical LP bearing oil pressure versus loading at 1,800 RPM. On Fig. 7 are shown the maximum film pressure and the estimated bottom oil pressure versus loading. It is of interest to note that the maximum pressure in a fluid film journal bearing does not correspond to the location of the maximum loading. The maximum oil film pressure in a fluid film journal bearing is always offset to the direction of the maximum loading. The angle of offset to the point of maximum film pressure is in the direction of rotation. For the heavily loaded bearings, the maximum oil film pressure is offset from the orifice pressure by an angular location of approximately 15 to 20 degrees measured in the direction of rotation.

Figure 8 represents the various HP, LP, generator and exciter estimated bearing loads as a function of orifice pressure. It is seen that for a given orifice pressure, the generator bearings carry the greatest loading. The exciter bearing load capacity is considerably lower than any of the other bearings. The measurement of low orifice pressure for any of the LP bearings then may indicate an unloaded or lightly loaded bearing. The bearing load capacity may be increased or reduced by changing the bearing vertical alignment.

Table 8 represents the influence of vertical bearing alignment and the corresponding bearing reactions estimated from the bearing orifice pressure measured. In Table 9, are shown the bearing displacements computed with the cubic spline fit and the bearing forces that would result from such an alignment. The next column shows the bearing orifice measured pressure as furnished by SCE. The bearing load was then estimated from the orifice pressure. It is of interest to note that the number 7 LP3 outboard bearing has an orifice pressure of only 200 psi and the number 8 LP3 inboard

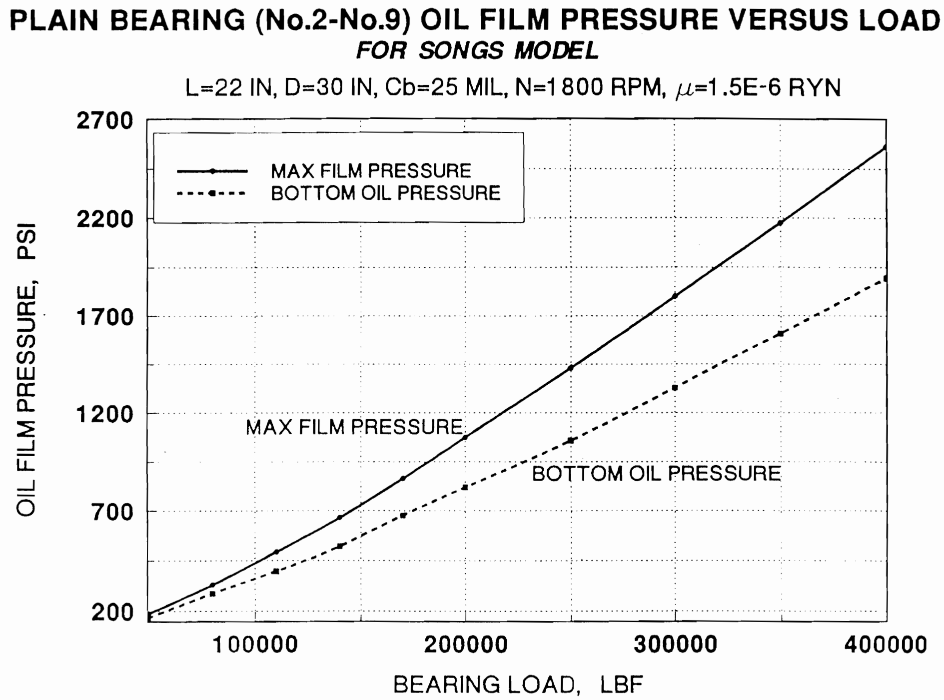

Figure 7 Maximum and Bottom Oil Film Pressure Vs Bearing Load for a Typical LP Bearing at 1,800 RPM

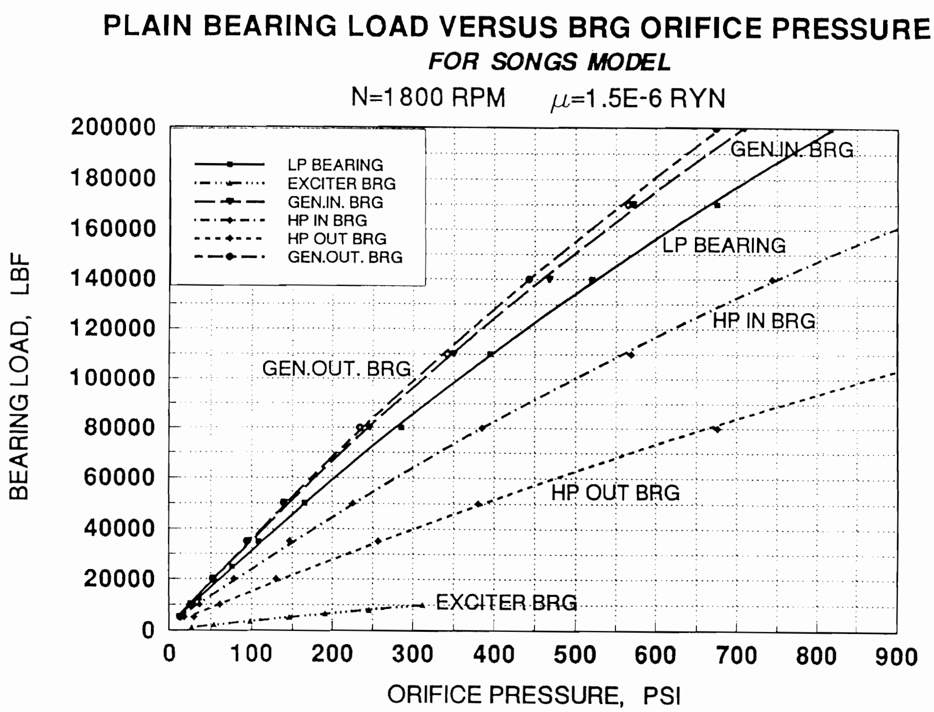

Figure 8 Estimated Bearing Loading Vs Orifice Pressure for Various Bearings at 1,800 RPM

Table 8

Vertical Bearing Alignment and Corresponding Bearing Reactions Estimated from Bearing Orifice Pressure Measured & Initial Bearing Support Displacements

Computed for SONGS Model

N=1800 RPM, μ=1.5E-6 REYN.

By Using **MSC/Pal2** Finite Element Program

July 27, 1993

	With Cubic Spline Fit σ_{max}=4,390 Lb/In^2 Optimal Alignment		Bearing Orifice Pressure	Bearing Load Estim. From Orifice Pressu.	Brg Support Initial Displ. IterationBegins	Estimation Close To Pressure Measured σ_{max}=7,203 Lb/In^2 (After 70 Iterations)	
Bearing Location	Brg Supt. Displ.(In)	Brg Force(Lb)	(PSI)	(Lbs)	(In)	Brg Supt. Displ.(In)	Brg Force(Lb)
#1 HP Outboard	-1.4	62,600	500	62,600	-1.4	-1.35	65,140
#2 HP Inboard	-1.0836	94,090	400	94,090	-1.0835	-0.99	79,990
#3 LP1 Outboard	-0.925	151,800	700	255,400	-0.8	-0.80	148,100
#4 LP1 Inboard	-0.4	175,000	550	208,500	-0.3	-0.23	205,200
#5 LP2 Outboard	-0.1912	173,800	500	192,300	-0.15	-0.088	177,200
#6 LP2 Inboard	0.4242	168,500	500	192,300	0.3	0.274	199,700
#7 LP3 Outboard	0.6548	166,100	200	86,800	0.8	0.38	87,440
#8 LP3 Inboard	1.3	179,400	100	48,300	1.45	1.12	237,000
#9 Gen. Inboard	1.5203	214,100	400	179,000	1.6	1.31	183,000
#10 Gen. Outboard	2.5	206,900	450	207,600	2.5	2.43	208,200
#11 Exciter	2.7707	5,432	200	6,700	2.77	2.90	6,806
	Total Brg Load	1597722		1,534,000		Total Brg Load	1597776

bearing pressure is only 100 psi. This would indicate very lightly loaded bearings with loads of only 86,800 lb and 48,000 lb, respectively. The measurement of only 100 psi for the number 8 LP3 inboard bearing should be confirmed as this indicates an unloaded bearing.

In order to match the assumed bearing loads, the finite element program was used with various alignment procedures. A total of over 70 iterations was used to try to match the load capacity as shown in the estimated bearing loads. Excellent agreement was obtained for the bearings except the number 8 LP3 inboard, which had only 48,000 lb loading. The instrument measuring the LP3 inboard bearing number 8 may have been in error. From these calculations, it is seen that the static orifice pressure may be used to estimate the bearing loading on the various bearings and hence can also be used to indicate whether the proper initial vertical alignment of the bearings has been performed.

7. Bearing Synchronous Stiffness and Damping Characteristics Including Foundation Effects

It was found that the predominant influence on the bearing stiffness and damping characteristics for the turbine-generator bearing characteristics was the influence of foundation effects. An extensive number of calculations of the bearing coefficients with various foundation flexibility conditions was performed in order to determine the influence of foundation flexibility on the bearing stiffness and damping coefficients. In general, the influence of foundation flexibility had a profound effect on reducing the effective bearing damping in particular. Details of this are given in Appendix B. In Appendix B are also shown plots of the bearing synchronous stiffness and damping with and without foundation effects. For example, if the bearing stiffness is 20E6 and the foundation stiffness is only 4E6, then the maximum effective bearing stiffness must be less than 4E6.

An even more significant reduction in bearing damping is obtained. Although the bearing effective stiffness may drop by a factor of 5, it is possible for the effective bearing damping to drop by a factor of 50. Table 9 represents typical values of effective stiffness and damping assuming foundation stiffness values of 4E6 and 2E6 in vertical and horizontal directions.

In summary, various changes in bearing stiffness and damping characteristics in the bearings due to various clearance differences have little influence in the overall effective bearing stiffness and damping characteristics when foundation flexibility effects are included.

The influence of the foundation on the bearing effective stiffness and damping characteristics was computed using the RODYN BRGCOEF bearing lookup program. In this program the Y direction is in the vertical direction and the X direction is in the horizontal direction. This is opposite from the convention used in the DYROBES finite element bearing program. Excellent agreement was obtained between the RODYN BRGCOEF bearing program and the DYROBES program.

Table 9 Stiffness and Damping Coefficients of 11 Bearings of the 1150 MW Turbine-Generator Considering Foundation Flexibility

BRG Characteris-tics	HP outboard BRG	HP inboard BRG	LP outboard & inboard BRGs	Generator inboard BRG	Generator outboard BRG	Exciter BRG
Kyy (lb/in)	1.8E6	1.7E6	1.82E6	1.82E6	1.82E6	1.0E6
Cyy (lb-s/in)	1600	3000	1200	1000	900	5600
Kzz (lb/in)	3.6E6	3.5E6	3.66E6	3.62E6	3.63E6	2.9E6
Czz (lb-s/in)	500	600	350	500	550	1600
Bearing Load (lbs)	1E5	1E5	2E5	2E5	2E5	1E4