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DESIGN STUDY (OF) ELECTRICAL COMPONENT
TECHNOLOGY FOR 0.25 TO 10.0 MEGAWATT
SPACE POWER SYSTEMS

Parametric Study of Inductor Alternators

By
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February 1968

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Westinghouse Electric Corporation
Lima, Ohio

UNITED STATES ATOMIC ENERGY COMMISSION • DIVISION OF TECHNICAL INFORMATION

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Aerospace Electrical Division
Westinghouse Electric Corporation

REPORT PARAMETRIC STUDY
OF
INDUCTOR ALTERNATORS

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ABSTRACT

Parametric studies to determine minimum system weight as a function of electrical rating, frequency, speed, voltage, and coolant temperature are presented on inductor alternators for ratings of 0.25 to 10.0 megawatts for space nuclear electric power plants utilizing the Rankine conversion cycle.

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1. INTRODUCTION

A parametric study of generator designs was performed to aid in the determination of optimum system parameters applicable to Rankine nuclear space power systems in the power range of 0.25 to 10.0 megawatts. The energy conversion equipment consists of a turbine-driven inductor generator with liquid potassium providing the cooling for the generator.

The study was performed by calculating a series of generator designs for various combinations of the five basic system parameters listed below:

generator rating	(megavolt-amperes)
frequency	(Hertz)
speed	(rpm)
output voltage	(volts line-to-neutral)
coolant temperature	(°F)

The evaluation was made by plotting the above parameters against the design conditions most important to the system designer, generator weight and the powerplant and cooling system weight penalties associated with generator losses (efficiency).

The parametric study was performed in two cycles. The first cycle was conducted to screen out parametric combinations which result in designs exceeding material limitations and to establish the feasibility of the initial material and parametric range choices. Results obtained from the first cycle designs were then used in the second cycle study to more specifically determine the best combinations of frequency, speed, and voltage for the specified ratings and coolant temperatures.

Following this, special studies were carried out on generators with advanced, high-temperature materials applicable to 1400°F coolant temperatures.

The range of parameters and/or specific parametric points chosen for the first cycle studies is presented in the following tables.

FIRST CYCLE-FREQUENCY PARAMETRIC STUDY

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Volts</u>	<u>RPM</u>	<u>Frequency Points (Hz)</u>
0.4	500,800,1100	1000	24 000	800,1200,2000,2800,3200
1.2		1000	15 000	500,1000,2000,2500,3000
6.0		1000	10 000	500,1000,2000,2500,3000
12.0		1000	8 000	400,1200,2000,2800,3200

FIRST CYCLE-SPEED PARAMETRIC STUDY

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Volts</u>	<u>Frequency</u>	<u>Speed Points (RPM) **</u>
0.4	500,800,1100	1000	*	12 000 to 30 000
1.2		1000	*	8 000 to 20 000
6.0		1000	*	4 000 to 15 000
12.0		1000	*	2 000 to 12 000

* As chosen from preceding frequency study.

** The exact speed points depend on the frequencies selected.
Five speed points were chosen from the speed ranges shown.

FIRST CYCLE-VOLTAGE PARAMETRIC STUDY

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Frequency</u>	<u>RPM</u>	<u>Voltage Points Line-to-Neutral</u>
0.4	500,800,1100	+	+	500,1000,1500,2140
1.2		+	+	500,1000,1500,2140
6.0		+	+	500,1000,1500,2140
12.0		+	+	500,1000,1500,2140

+ Selected from the preceding studies.

2. BACKGROUND DATA

Before starting the parametric study, certain generator design factors, system design factors, material choices, and overall limitations were established. A review of the data obtained and the conclusions reached on previous space, electric power generator programs and materials programs proved useful in establishing material choices, electrical and mechanical design parameters, and the parametric study approach and procedures. The previous generator programs were the SNAP-50/SPUR Generator Studies (1)* (USAF Program 675A), the NASA Generator Parametric Study (2) (NAS 5-1234), and in-house construction and test of a 50 kW inductor generator model. The materials programs were Contracts NAS 3-4162(3), (4) and NAS 3-6465(5).

The SNAP-50/SPUR generator studies dealt in detail with electrical, mechanical, and thermodynamic circuit design, materials, fabrication techniques, and anticipated performance of a 467 kVA inductor generator with a 600°F liquid-metal coolant. Data utilized from this program included (1) choice of internal electrical parameters, (2) rotor materials, (3) manufacturing tolerances, (4) cooling circuit design techniques, and (5) calculation of pole face losses.

The NASA parametric study involved calculation of some 400 liquid-metal cooled inductor generator designs covering a range of ratings from 250 kW to 10 MW. Since this generator parametric study was completed in December 1962, a number of modifications and refinements have been made to improve inductor generator design calculation techniques. More precise flux plotting methods have been adopted for use and additional leakage fluxes have been taken into account. Recently verified pole face loss calculation techniques and rotor stress and cooling design calculations were added to improve routines relative to rotor design.

Results of the NAS 5-1234 study provided a basis for choosing internal electrical parameters and for establishing the parametric study approach and procedures for the present program.

*Superscript numbers in parentheses refer to references listed at end of report.

The Westinghouse in-house program for construction and test of a 50 kW inductor generator model⁽⁶⁾ served to verify the inductor generator design program as well as provide experience in actual inductor generator construction.

The topical reports from NASA Materials Program, NAS 3-4162, provided the required information for selecting generator conductor, insulation, magnetic, and structural materials as discussed in detail in following sections of this report. Information gathered on a current electrical materials program, NAS 3-6465, was also utilized as it became available.

2.1 MATERIALS

The following materials were selected for use during the design study.

2.1.1 Stator Steel

Magnetic Materials Report NASA-CR-54091⁽³⁾, shows HIPERCO 27 and CUBEX to be suitable for the temperature range to be considered in this study. Since fabrication difficulties have been encountered with CUBEX, HIPERCO 27 was chosen for the stator punchings and the forged frame. Punchings 0.004-inch thick were used to minimize iron losses in the stator stacks. Magnetization curves determined from the data on pages 303 and 305 of NASA-CR-54091 were used, as shown by figures 1 and 2.

The core loss curves, figures 3 and 4, were prepared from data published in Materials Report NASA-CR-54091, pages 318-323. Core loss values from the report were multiplied by three, as past design experience has shown this to be necessary to allow for harmonics, stray losses, stacking, and other fabrication effects.

A stacking factor of 0.90 was used for the 0.004-inch stator punchings* and a stator steel and frame steel density of 0.285 lbs/in³ was used for size and weight calculations.

2.1.2 Rotor Steel

H-11 alloy was chosen for 500° and 800°F coolant temperatures and NIVCO was chosen for 800°F (initially) and 1100°F coolant temperatures based on a preliminary rotor materials comparison study which compared the combined magnetic-physical

*This is in accordance with past experience with the 0.004-inch thick stator punchings used on the SNAP-50/SPUR experimental generator.

Figure 1. DC Magnetization Curves for Hiperco 27,
0.004-Inch Laminations for Stator Stacks

Figure 2. DC Magnetization Curves for Hiperco 27 Frame
Forgings

Figure 3. Inductor Generator Hipercro 27 Stator Tooth Loss
at 60 kL/in²*, 0.004-Inch Laminations

* Effective flux density for non-reversing teeth flux.

Figure 4. Inductor Generator Hipercro 27 Core Loss
at 70 kL/in², 0.004-Inch Laminations

properties of H-11, NIVCO, precipitation-hardened martensitic and cobalt alloys and dispersion-hardened cobalt and iron-cobalt alloys over a 500° to 1600°F temperature range. A final choice of H-11 alloy at 500° and 800°F and an experimental alloy 1-B-S-1* at 1100°F was made after preliminary rotor stress and temperature comparisons were completed. Magnetization curves, pages 431 and 482 of NASA-CR-54091, were used to obtain the curves shown in figure 5 for the H-11 and NIVCO alloys. The estimated magnetization curve for 1-B-S-1 alloy is also shown in figure 5.

A rotor steel density of 0.281, 0.312, and 0.315 lb/in³ were used for weight calculations for H-11, NIVCO, and 1-B-S-1 respectively.

Figure 5. DC Magnetization Curves for H-11, 1-B-S-1, and NIVCO Rotor Forgings

*Precipitation-hardened cobalt base alloy (alloy 1-B-S-1) being developed by Westinghouse Electric Corp. on Contract NAS 3-6465 for the NASA, Lewis Research Center; composition is Co-15Ni-5Fe-5Ta-0.2Ti-1.25Al.

Maximum rotor hot-spot temperature (at the pole faces) for the H-11, NIVCO, and 1-B-S-1 alloys are shown below. Limiting temperatures are based primarily on material strength and stability considerations.

<u>Material</u>	<u>Maximum Temperature</u>
H-11	950°F
NIVCO	1200°F
1-B-S-1	1300°F

2.1.3 AC Windings

Rectangular, Inconel 600-clad silver wire with Anadur insulation was used for coolant temperatures of 500° and 800°F. This insulation is composed of fiber glass, fusible glass frit, and refractory oxide powders retained on the conductor during winding by a resin binder. Subsequent firing after winding fuses the glass frit to serve as a rigid binder. An insulation thickness of 0.011-inch per conductor side was used for all 500° and 800°F coolant temperature designs. Report NASA-CR-54092, page 279 gave the following electric strengths for 0.0068-inch Anadur insulation at 3200 Hz:

<u>°F</u>	<u>Total Volts</u>
500	430
932	420
1112	410

A more recent work has shown that an Anadur thickness of 0.011 inch is better than 0.0068 inch from the standpoints of application and damage resistance. A conservative value of 400 volts electric strength will be assured for all generator designs where Anadur insulation is used.

Rectangular dispersion-strengthened copper conductors with alumina insulation plasma-arc-sprayed on preformed coils was used for coolant temperatures of 1100°F and above. Report NASA-CR-54092, page 469, shows high-purity alumina electric strengths as follows:

<u>°F</u>	<u>Volts/mil</u>
932	140
1600	46

A conservative value of 50 volts/mil was used on 1100°F coolant designs where winding temperatures in the order of 1400°F were anticipated.

A conductor insulation thickness of 0.008-inch per conductor side was assumed to provide an electric strength of 400 volts as assumed for the Anadur insulation used in lower temperature designs.

The major portion of the phase-to-phase and phase-to-ground insulation is accomplished by the slot liners and separators and by phase insulation inserted in the end turns where necessary. The conductor insulation, besides providing some phase-to-phase and phase-to-ground insulation, also (1) provides separation between parallel conductors in the same slot, thus allowing more conductor surface area to reduce skin effect at higher frequencies and (2) insulates between series conductors in the same slot where they are used.

Inconel 600-clad silver wire has a 75 percent average conductivity and dispersion-strengthened copper wire has a 91 percent average conductivity relative to copper over the temperature ranges to be considered.

2.1.4 Field Windings

Wire and wire insulation as described for the a-c windings was used.

2.1.5 Stator Slot Insulation

Two U-shaped slot cells and a wedge was used for each slot. Alumina, 98.5% pure, was used for these components for coolant temperatures of 500°, 800°, and 1100°F. The rigid alumina slot insulation does not completely isolate the conductors, because air gaps exist between the conductors and iron, between adjacent surfaces of the wedge and upper slot cell and between adjacent surfaces of the upper and lower slot cells. The wedge and cell thickness must therefore be chosen to provide the necessary electric leakage path length in the surrounding stator ambient. Choice of stator cavity pressure to allow a gradient of 50 volts/mil was assumed and the peak voltage to be insulated was calculated by: (1.414 times rms volts per phase minus 400 volts capability of conductor insulation) times 1.5 safety factor.

Following are the wedge and cell thicknesses which are required and those which were used. Since 0.018-inch is the minimum thickness practical from a fabrication and handling standpoint, the 500-volt insulation thickness will be greater than required electrically. A minimum practical wedge thickness of 0.04-inch was chosen.

Rated V/ph	Thickness to be used (inches)		Minimum Cell and Wedge Thickness Required (inches)
	Cell	Wedge	
500	0.018	0.04	0.0092
1000	0.030	0.04	0.0304
1500	0.052	0.052	0.0517
2140	0.079	0.079	0.0787

2.1.6 Stator Encapsulation

Aluminum orthophosphate plus zircon potting compound will be assumed for the 500°F designs to hold the windings in place and to provide better heat transfer. No potting compounds are presently available which would prove satisfactory on the 800° and 1100°F designs.

2.2 ELECTRICAL DESIGN PARAMETERS

Generator ratings of 0.4, 1.2, 6.0, and 12.0 MVA were chosen for the study. The 0.4 MVA rating was chosen because it allows comparison and utilization of previous SPUR generator and systems data. The 1.2 MVA rating was likewise chosen because of commonalities with previous studies (7), (8), (9). The 6 and 12 MVA ratings were chosen to complete coverage up to 12 MVA as required. The following electrical design parameters have been selected for use during the design study.

2.2.1 Flux Density in Stator Frame, Core, and Teeth

The following flux densities were used over an assumed 600° to 1500°F iron temperature range corresponding to the 500° to 1400°F coolants:

Frame	70 kL/in ²
Core	70 kL/in ²
Teeth	120 kL/in ²

Past work on the SNAP-50/SPUR generator design showed that 70 kL/in² for the core and 120 kL/in² for the teeth provide the best combination of low weight and high efficiency for a 600°F coolant temperature; a frame flux density of about 110 kL/in² was used for the SPUR generator to provide a balanced magnetic design. A frame density of 70 kL/in² was used for all of the first cycle studies and a major portion of the second cycle studies. This density value was used to complete the studies in order to provide continuity in the parametric trends. A frame density of 110 kL/in² was used for the conceptual designs in the Final Report (10).

The HIPERCO 27 magnetization curves of NASA-CR-54091, pages 303 and 305, show little variation in excitation requirements over a 600° to 1500°F temperature range for flux densities between 70 and 120 kL/in²; therefore, stator densities were not varied with coolant temperature.

2.2.2 Flux Density in Rotor Core

The following rotor core flux densities were used:

<u>Coolant Temp.</u>	<u>Assumed Iron Temp.</u>	<u>Material</u>	<u>kL/in²</u>
500°F	600°F	H-11	82
800°F	900°F	NIVCO	50
1100°F	1200°F	NIVCO	50

The 82 kL/in² for H-11 at 600°F was based on the value found to provide a satisfactory SPUR design. According to the curves of NASA-CR-54091, pages 431 and 482, the chosen flux densities for NIVCO at 800° and 1100°F provide approximately the same excitation requirements as those for H-11 at 600°F.

2.2.3 AC Winding Current Density Desired

A current density of 9000 amps/in² was chosen based on past SPUR studies.

2.2.4 Field Winding Current Density Desired

A current density of 4000 amps/in² was chosen based on the previous SPUR and NAS 5-1234 designs. No tests or optimizing calculations for field current density had been run as on the a-c windings.

2.2.5 Direct Axis Synchronous Reactance

The following values were used initially as a desired goal. A plus or minus 10 percent tolerance was allowed initially.

<u>Rating - MVA</u>	<u>Desired X_d (P.U.)</u>
0.4	1.0
1.2	1.2
6.0	1.3
12.0	1.3

These values were chosen on the basis of the 467-kVA SPUR design which had an X_d of 0.935 and on the following average X_d values of the best NAS 5-1234 designs.

<u>kW Rating</u>	<u>X_d</u>
200	1.19
600	1.23
1000	1.15 to 1.18
2000	1.33
5000	1.28
10000	1.29

The chosen X_d values will reasonably limit: (1) voltage unbalance for unbalanced loading conditions, (2) excitation power, (3) transient surges, and (4) increase of field power for overloads.

2.2.6 Parallel Paths in Field

One parallel path was chosen. This can be changed later, depending on the excitation power supply, without significantly affecting size or weight.

2.2.7 Maximum Desired Field Amps at Continuous Full Load

The following values were chosen:

<u>MVA Rating</u>	<u>Field Amps</u>
0.4	25
1.2	48
6.0	240
12.0	480

These were based on a value of 25 amperes used in the SPUR design and a value of 40 amps/MW used in the NAS 5-1234 study. Final designs can be varied by changing the number of field turns and standard wire size to correspond with the desired field amps.

2.2.8 Minimum Power Factor Angle

A power factor angle of -31.8°, corresponding to a lagging power factor of 0.85, was used.

2.2.9 Single Air Gap and Bore Seal Thickness

The following single air gaps were used:

<u>MVA Ratings</u>	<u>Bore Seal Wall Thickness (inches)</u>	<u>Single Air Gap (inches)</u>
0.4	0.065	0.1 (not dependent on bore seal)
1.2	0.080	0.125 (not dependent on bore seal)
6.0	0.129	0.229 (bore seal thickness and running clearance)
12.0	0.175	0.275 (bore seal thickness and running clearance)

Air gap width is dependent on desired electrical performance, allowable pole face losses, and required bore seal thickness. A preliminary study of NAS 5-1234 data showed a single air gap of 0.125 inches provided lowest weight and highest efficiency for 1-megawatt ratings. Two 1-MW conceptual designs of NAS 5-1234 had single air gaps of 0.125 inch with rotor outside diameters of 14.3 and 14.7 inches and one had a single air gap of 0.100 inch with a rotor outside diameter of 14.0 inches. The SPUR generator design had a single air gap of 0.10 inch, a rotor outside diameter of 11.35 inches, and a bore seal wall thickness of 0.070 inch. Data taken from NAS 5-1234 showed the following typical rotor diameters and contacts with bore seal manufacturers (Coors Porcelain and Western Gold Co.) indicate that the following minimum bore seal thicknesses are practical for these diameters from a manufacturing viewpoint.

<u>kW Rating</u>	<u>Avg. Rotor OD (inches)</u>	<u>Minimum Practical Bore Seal Thickness (inches)</u>
300	10.5	0.063
600	12.9	0.078
1000	14.3	0.086
1000	11.7	0.070
2000	14.9	0.090
5000	21.4	0.129
10000	28.8	0.175

2.2.10 AC Winding Temperature

An ac winding temperature 300°F above the average coolant temperature was assumed. The computer uses this only as a design starting point. Actual ac winding temperatures are calculated and used to calculate winding I^2R losses.

2.2.11 Field Winding Temperature

A field winding temperature of 300°F above the average coolant temperature was assumed for all designs. Since the computer does not calculate the actual field winding temperature, this assumed temperature is used to determine field losses. Actual field winding temperatures are calculated only after the field cooling configuration has been designed in detail.

2.2.12 Stator Slot Configuration

The slot configuration to be used is shown below compared to conventional configuration:

Configuration to be Used

Conventional Open Slot

Use of the proposed variation from the conventional configuration allows a smaller slot opening to reduce pole face losses. Further reduction of the slot opening would further reduce the pole face losses but would require the ac windings to be inserted axially and then joined on the turns. The proposed slot configuration was used on the SPUR generator design.

2.2.13 Pole Embrace

A pole embrace of 67 percent was used based on a previous SPUR study. The study showed that 55 percent pole embrace provides minimum weight, 80 percent provides maximum efficiency, and 40 percent provides lowest conductor temperatures and rotor tangential stress. It concludes that 67 percent provides the best combination of advantages.

2.2.14 Phase Belts

Phase belts of 60 degrees were used in all designs to provide the lightest weight designs. A 67 percent pole embrace and $2/3$ pitch were used to eliminate the third harmonic.

2.2.15 Parallel Paths in AC Windings

The number of parallel paths affects the following: (1) heat transfer of armature winding loss, (2) required conductor size, and (3) synchronous reactance. Based on design experience, the following values were used on initial designs.

<u>No. of Poles</u>	<u>Parallel Paths</u>
4,8	4
10,20	5
6,12,18,30,36,42	6
14	7
16,24,48	8

2.2.16 AC Conductors in Parallel

Four conductors in parallel in the same slot were used for initial designs.

The number of conductors in parallel determines (1) eddy current factor, and (2) the required conductor size. The 120-volt SPUR generator used four conductors in parallel in each slot.

This parameter in conjunction with the number of parallel paths were varied as required to obtain the best combination of weight, efficiency, and ease of winding.

2.2.17 Slot Skew

Slot skew in the stator stack to reduce slot harmonics was not used because of the difficulty of inserting rigid ceramic slot liners into skewed slots.

2.2.18 Clearances

The following clearances were used, based on generator fabrication experience and Westinghouse Manufacturing Engineering recommendations.

<u>Clearance</u>	<u>Inches</u>
Between end turns of ac windings	0.075
Slot width	
400 kVA and 1200 kVA ratings	0.018
6 MVA and 12 MVA ratings	0.026
Slot depth	0.02
Between top of field winding and frame	0.09
Between bottom of field and ac windings	0.03
Frame air gap to allow for fit tolerances between flux carry- ing sections of the frame	0.0015
Field winding between conductors	0.002
Field winding end clearance	0.014

2.2.19 Tolerances

The following tolerances were chosen to limit the computer time required for each design and to allow designs to be obtained using standard conductor sizes.

Flux densities	<u>±</u> 2%
Coil throw	<u>±</u> 10%
Direct axis synchronous reactance	<u>±</u> 10%

2.3 PERTINENT MECHANICAL DESIGN PARAMETERS

The calculated stresses and temperatures within the rotor were combined with creep data for the various rotor materials to determine the creep strain of the rotor. Typical creep data for H-11 and NIVCO is given in figure 6. Any creep strain over 0.2 percent near the outer rotor core was considered excessive; on the other hand, creep strains up to 0.5 percent near the central portion of the rotor were considered acceptable.

A rotor cooling configuration which has several coolant passages through the core of the rotor was used. The computer locates the passages in the core such that the maximum hot-spot in the rotor can be maintained at less than the specified limiting design temperatures.

a. 10,000-Hr. Creep Strain
H-11 Alloy Steel Bar

b. 40,000-Hr. Creep Strain
H-11 Alloy Steel Bar

c. 10,000-Hr. Creep Strain
Nivco Alloy Bar

d. 40,000-Hr. Creep Strain
Nivco Alloy Bar

Figure 6. Creep Strain Data

The stator coolant configuration was represented by a constant temperature sink on the armature stack outside diameter. Other cooling configurations such as cooling tubes in the laminations or armature slots were not considered because of their fabrication complexities and the added tooth and core losses which result from their use.

The ambient atmosphere in the stator was assumed to be an inert gas at more than 100 torr or less than 0.1 torr absolute pressure. Gas pressure at one atmosphere was not used due to the undesirability of having excessive pressure differential across the ceramic bore seal; intermediate vacuum between 0.1 and 100 torr was considered because of possible corona difficulties. Because the stator materials outgas over the operating life of the generator, the preferred pressure is 100 torr rather than 0.1 torr.

The ambient atmosphere in the rotor cavity is potassium vapor at a pressure compatible with the coolant temperature, rotor speed, stub shaft sizes, shaft seal losses and windage losses. Use of Holweck pump vapor shaft seals allow the optimum combination of these parameters to be chosen somewhat at will. Dynamic liquid shaft seals such as viscoseals or dynamic slingers were chosen according to their applicability to specific design conditions.

The rotor stub shaft sizes, bearing bore sizes, friction loss coefficients, etc., were chosen to match actual operating conditions and were in accordance with liquid metal bearing practices. Pivoted pad bearings were the type specified for determining configuration dependent parameters. Bearing load pressure was limited to 15 psi based on the static weight of the rotor for the initial parametric screening studies.

Rotor pole faces slotted with 0.005-inch wide by 0.062-inch minimum deep slots were assumed.

Ceramic bore-seals were assumed in the air gap to protect the stator components from the potassium vapor in the rotor cavity; the thicknesses used have been presented previously. Metal bore-seals, while having attributes relative to fabricability and reliability, have prohibitively high can losses in high speed machines. These losses may be calculated with the equation

$$w = \frac{15.5 \left((Bac) (RPS) (SL) \right)^2 (D^3) (t) 10^{-16}}{2\rho (SL + \frac{LBS}{2})} K_S \quad (\text{watts})$$

where

$$B_{ac} = \frac{(P)(E_g)10^8}{8(FF)(f)(N_{ph})(D)(SL)} \quad (\text{lines/in}^2)$$

B_{ac} = air gap flux density
 RPS = revolutions per second
 D = stator ID, inches
 SL = single stack length, inches
 t = can thickness, inches
 ρ = resistivity of can material, ohm-inch
 LBS = length between stacks
 P = number of generator poles
 f = frequency, Hz
 N_{ph} = series turns per phase
 E_g = air gap voltage per phase
 FF = form factor
 K_s = loss reduction factor

For illustrative purposes, the losses in a metal can for the SNAP-50/SPUR 467 kVA experimental generator would have been 228 kW for a 0.010-inch-thick can of Inconel X-750 or 135 kW for a 0.010-inch-thick can of titanium 12% aluminum alloy. The titanium-aluminum alloy has one of the highest resistivities of commercially available materials.

The can loss cannot be arbitrarily reduced by operating at lower frequencies if the turbo-generator is to remain a high-speed unit for minimum weight and maximum efficiency since can loss is proportional to rotor peripheral speed. This is illustrated below.

Let

peripheral speed = $v = (RPS)(\pi D)$, inches/sec
 can volume = $Vol = \pi(D)(t)(SL)$, cubic inches
 assume $LBS \approx SL$

Then

$$w \approx \frac{15.5(B_{ac})^2(v)^2(Vol)10^{-16}}{1.5\pi^3\rho} \quad (\text{watts})$$

which shows that can losses are proportional to the square of the rotor peripheral speed, the can volume and the inverse resistivity. Since rotor stress is proportional to rotor peripheral speed too, v will be high for all highly stressed generator rotors. This indicates that, in general, losses in metal

bore seals will be very high for turbine-driven space power generators.

To illustrate the reduction in can loss that can be achieved by lowering the peripheral speed, a low stress design for a 316 kVA generator was chosen from the NAS 5-1234 reports for analysis of can losses. Pertinent design information is listed in table 1 along with that for SNAP-50/SPUR for comparison.

Table 1. Comparative Can Loss Data

<u>Item</u>	<u>Low Stress Configuration</u>	<u>SNAP-50/SPUR Configuration</u>
Generator Rating, kVA	316	467
Voltage	1000/1730	120/208
Frequency, Hz	1000	3200
Speed, rpm	15000	24000
Diameter, inches	11.2	11.6
Number poles	8	16
Eg, Volts	1150	148
N _{ph}	70	5
FF	0.855	0.885
SL, inches	2.26	3.3
LBS, inches	4.28	2.04
Loss reduction factor (due to can extension beyond stack)	0.87	0.97
Can Mtl. resistivity, microhm-inch	106, Ti - 12% Al Alloy	
Rotor Peripheral Speed, inch/sec	8800	14600
Average Stress, psi	21000	57000
Bac, kilolines/in ²	71.4	55
Can Loss - kW per 10 mil can thickness	33	134

As shown in table 1, the losses can be reduced considerably; however, they are still too high for practical space power generator requirements.

Included in the generator computer design program are calculations for the bearing, shaft-seal and windage losses. Since these losses are included in the generator efficiency calculations, the bearings and seal designs specified must be based on realistic operating conditions and requirements rather than on arbitrarily selected designs. A dilemma occurs, however, because the bearing and seal designs cannot be completed until such information as the rotor weight and speed are specified in the generator designs; thus, a first-cut trial using

Figure 7. Generator Rotor Weight at Various Design Speeds

arbitrary bearing and seal designs is usually required to approximate the generator design from which more definitive bearing and seal designs may then be made. On a parametric study such as this, such cut and try methods are not practical and it becomes necessary to specify generalized generator design configurations upon which approximate bearing and seal designs may be based.

Since the generator design computer outputs from the previous 1 to 10 Megawatt Study(2) were still available, a set of general design parameters (namely rotor weight and speed) could be specified for the parametric bearing and seal designs of this study. A composite of the rotor weights from the previous study are shown as a function of speed and electrical rating on figure 7. Deviation between the generalized rotor weight and speed curves with parametric study results, while certainly having an effect on the final selected bearing designs, will not have a significant effect on the calculated generator efficiency and, therefore, will not affect the overall conclusions drawn from the generator parametric study.

Four important computer input parameters which were calculated using the preliminary data on figure 7 are:

Bearing bore (BB or D)
 Stub shaft (seal) diameter (SD)
 Friction coefficient (bearings and seals)
 Density of gas in rotor cavity

The analyses to approximate these input parameters are summarized in the following sections.

2.3.1 Bearing Bore and Friction

The bearings are pivoted pad journal bearings that are sized by considering the operating ranges and limits of certain dimensionless bearing characteristic numbers over the range of parameters applicable to the design study. Specifically, they were the:

- a) Bearing characteristic number (S_p) which is equivalent to the more popular Sommerfield number with added parameters to account for variations in pad configurations.

$$S_p \equiv k^2 C_w \left(\frac{R}{C} \right)^2 \frac{\mu n}{P}$$

k = fraction of bearing circumference filled with pads
 C_w = pad configuration aspect factor
 R = bearing radius, inches
 C = mean radial clearance, inches
 μ = fluid absolute viscosity, lb sec/in²
 n = shaft speed, rps (N = rpm)
 P = bearing pressure = W/LD , psi
 W = load on bearing, lbs
 L = bearing length, inches
 $D = 2R = BB$, inches

- b) Taylor ratio which is a measure of the turbulence and consequently the losses in the bearing and is the ratio of the mean Reynolds number to the critical Taylor number; that is,

$$T_r \equiv Re/Ta = \frac{\rho V C}{\mu g_c} \bigg/ \frac{41.1}{\sqrt{C/R}}$$

where

V = velocity, in/sec
 ρ = density, lb/in³

c) The eccentricity ratio which is defined as

$$\epsilon = \frac{e}{c}$$

where e is the radial runout between the shaft centerline and the bearing centerline.

d) The pad aspect ratio B/L where

$$B/L = \frac{\pi Dk/n'}{L} = \frac{\pi k}{(L/D)n'}$$

and $n' =$ number of pads.

The following assumptions and constants were selected to establish the basis for approximating the bearing designs.

- a) Assume a 6-pad bearing with $k = 0.85$.
- b) Use $L/D = 0.75$ for 0.4 and 1.2 MVA rating designs; use $L/D = 1.0$ for 6.0 and 12.0 MVA ratings.*
- c) Consider the load to be directly over a pad - the condition of least load capacity - so that results per assumption d below are conservative.
- d) Assume dynamic loading corresponding to synchronous whirl about a radius e at all speeds, a worst case condition. Because of this assumption, magnetic loading was assumed to be negligible as well as unbalance loading.
- e) Limit ϵ to 0.95 maximum at 1/5 rated speed (startup film).
- f) Limit T_r to 10 or less at rated speed.

According to assumptions e and c and the design chart of figure 8, the following limit is established.

$$S_p \geq 0.75 \text{ @ } 1/5 \text{ rated } N$$

Thus

$$S_p \geq 3.75 \text{ @ rated } N \text{ which results in** } \epsilon \leq 0.885$$

*Selected on the basis of preliminary design calculations.

**It was later recognized that a better assumption here might have been to limit ϵ to 0.9 at 1/5 speed which would have resulted in $\epsilon \leq 0.77$. Since the results did not have a significant effect on the generator performance, the "improved" assumption was not applied to the parametric study input data.

Figure 8. Chart for Determining Eccentricity Ratio

According to assumptions a and b and the design charts of figure 9, the following constants are established.

$$k^2 C_W = 162 \text{ at } L/D = 1.0$$

$$k^2 C_W = 180 \text{ at } L/D = 0.75$$

According to assumption d, the maximum bearing load is:

$$\begin{aligned} W &= \frac{W_{ROT} + W_{dynamic}}{2} = \frac{W_{ROT}}{2} + \frac{W_{ROT}}{2g_c} e (2\pi n)^2 \\ &= \frac{W_{ROT}}{2} \left(1 + \frac{\epsilon c (2\pi n)^2}{g_c} \right) \end{aligned}$$

Combining the above bearing load with the bearing characteristic number gives:

$$S_P = \frac{k^2 C_W \mu n^2 L D}{W_{ROT} \left(1 + \frac{\epsilon c (2\pi n)^2}{g_c} \right)} = \frac{4k^2 C_W (R/c)^3 \mu n c (L/D)}{W_{ROT} \left(1 + \frac{\epsilon c (2\pi n)^2}{g_c} \right)}$$

Figure 9. Charts for Determining Aspect Factors

or, since $S_p \geq 3.75$,

$$(R/c)^3 \geq \frac{(3.75) (WROT) \left(1 + \frac{\epsilon c (2\pi n)^2}{9c}\right)}{4k^2 C_w \mu n c (L/D)} \quad (1)$$

Now,

$$T_r = \frac{\rho V c}{\mu g_c} \frac{\sqrt{c/R}}{41.1} \leq T_r \text{ max.}$$

Thus,

$$R/c \leq \left(\frac{41.1 T_r \text{ max } \mu g_c}{\rho c^2 2 \pi n} \right)^2 \quad (2)$$

The two equations, 1 and 2, thus define the limits of radius-to-clearance ratio for the design study and represent two equations with two unknowns, R and c . By equating the two equations and solving graphically (with T_r nominally equal to 6.3) the minimum bearing bores (BB) and minimum R/c ratios were determined; the results are shown on figures 10 and 11. Note that for the 6.0 and 12.0 MVA ratings, the resulting values of R/c are higher than what would normally be considered an acceptable value (i.e., clearance is too small). Some adjustments would have to be made in any final designs to lower the ratios to the order of 1000 to 1500 as on the 0.4 and 1.2 MVA ratings.

The bearing configurations were not analyzed relative to optimizing the damping coefficients and spring coefficients. As a result, conclusions were not drawn as to the stability behavior, critical damping, etc. of the rotor-bearing systems. It will be shown later in this section that the stability problems are not too severe for the generator alone since the operating speed is usually well below the first rotor-bending critical speed. Thus, the only critical conditions that must be considered in reaching the operating speed are the resonances associated with the bearing film spring stiffness. Because of this, a separate bearing rotor dynamics parametric study was conducted by Mechanical Technology Inc. The results are indicated in Appendix A.

The bearing coefficient of friction was determined using the data of figure 12 and the bearing dimensions previously calculated. However, the bearing characteristic was calculated

Figure 10. Minimum Bearing Bore (BB) as a Function of Design Speed N

Figure 11. Minimum R/c Ratios as a Function of Design Speed N

Figure 12. Chart for Determining Coefficient of Friction

using W equal to the static load only in this case. This results in a more conservative answer for bearing losses.

2.3.2 Stub Shaft (Seal) Diameter and Seal Friction

The minimum stub shaft diameter (SD) was determined using the configuration shown on figure 13. The following assumptions were made to simplify the analysis.

$$L_b = 0.55L$$

$$SD \geq BB = D$$

$$\Delta \leq c$$

$$L_s = 1.2 \text{ SD for viscoseal configurations on the 0.4 and 1.2 MVA ratings.}$$

$$L_s = 3.5 \text{ inches for dynamic seal configurations on the 6.0 and 12.0 MVA ratings.}$$

Figure 13. Stub Shaft Configuration

The latter assumptions are based on preliminary analyses which lead to the conclusion that, for the higher speed smaller ratings, viscoseal shaft seal configurations would be appropriate and, for the lower speed larger ratings, dynamic slingers are more appropriate. In order to make the dynamic slinger losses competitive with viscoseals during the analysis, the slinger configuration was assumed to have several stages, each approximately 0.5 inch in length.

The deflection was first approximated for the special case of $SD = BB = D$. Thus,

$$\begin{aligned}
 c &\geq \Delta \approx \frac{W(L_s + L_b)^3}{3E \left(\frac{\pi D^4}{64} \right)} \\
 &= \frac{64W(1.2 + 0.55 L/D)^3}{3\pi ED} \quad \text{for 0.4 and 1.2 MVA ratings} \\
 &= \frac{64W(3.5 + 0.55L)^3}{3\pi ED^4} \quad \text{for 6.0 and 12.0 MVA ratings}
 \end{aligned}$$

Upon substitution of appropriate values and solving for the maximum speed that gives $\Delta \leq c$, it was concluded that there is sufficient stiffness for the special case where the stub shaft diameter (SD) equals the bearing bore (BB); thus, further analyses for $SD > BB$ were not made. The limiting speeds for this conclusion are shown on figure 10 and are generally higher than the maximum generator design speeds. Because of this and the critical speed result below, a general rule was established for the generator parametric study whereby $SD = BB = D$.

The first critical speeds of the rotor/stub shaft configurations defined above may be approximated by considering the rotor mass to be infinitely stiff compared to the stub shafts and that there is no damping. The first bending critical is then

$$\omega_f \approx \sqrt{\frac{k_s g c}{W}} \quad \text{where } k_s = \frac{W}{\Delta_{\text{static}}} \quad (\text{lb/in})$$

and ω_f is in radius per second.

Therefore,

$$N_f = \frac{187.7}{\sqrt{\Delta_{\text{static}}}}$$

where Δ static is calculated with the above equations for $W = 1/2W_{ROT}$. The results are shown on figure 14. Reference to this figure and the selected generator speeds of the previous section reveals that the generator will always be operating well below the first rotor-bending critical speed.

The friction losses for the seals were determined using methods of references 12, 13 and 14 for the viscoseals on the 0.4 and 1.2 MVA ratings and the method of reference 15 for the dynamic seals on the 6.0 and 12.0 MVA ratings. The following assumptions were used as guidelines for the analyses.

- a) Seal design pressure differential = 30 psi.
- b) The seal must be capable of sealing this pressure at 1/3-rated speed (startup condition).
- c) On the viscoseals, double threaded configurations* would be used with a physical radial clearance $c' = 0.002$ SD; thus, the effective radial clearance $S = c'/2 = 0.001$ SD.

Figure 14. First Resonant Speeds for Selected Rotor/Bearing Stub Shaft Configurations

* Threads on Shaft and housing both

- d) On the dynamic slingers, the tip (Do) to hub (SD) ratio would be 1.4 and 7 stages* would be used. All stages would be utilized at 1/3 speed and only the last stage would be utilized at rated speed. The impeller configurations are smooth on the flooded side and vaned on the interface side. The pressure rise coefficients are 0.1 and 0.9 for the smooth and vaned sides respectively.

The losses in the viscoseal are a function of the pressure being sealed. The pressure head per inch of flooded seal is⁽¹³⁾,

$$\text{psi/inch} = 1.845 \times 10^{-4} \frac{\mu^2}{\rho s^3} \left(\frac{\rho V_s}{\mu g_c} \right)^{2.18}$$

The viscoseal losses are calculated in accordance with journal bearing theory with modifications to account for the threads. The resulting losses per inch of flooded seal are^{(12), (14)}

$$h_p/\text{inch} = 9.63 \times 10^{-11} \frac{\mu D^3 N^2}{s} \left(\frac{\rho V_s}{\mu g_c} \right)^{0.57}$$

or in terms of the pressure being sealed,

$$h_p/\text{psi} = 3850 \times 10^{-6} \frac{D^2 s N}{\left(\frac{\rho V_s}{\mu g_c} \right)^{0.61}} \quad (3)$$

The losses in the dynamic slinger seal may also be calculated as a function of the pressure head being sealed. The sealed pressure head in the last stage is, for the fluid vapor interface radius at the recommended 0.8 of the tip radius⁽¹⁵⁾,

$$\begin{aligned} \text{psi} = & \{0.9 [(1.4D/2)^2 - (<0.8><1.4D/2>)^2] - 0.1 [(1.4D/2)^2 \\ & - (D/2)^2] \} \frac{\rho w^2}{2g_c} = 0.136 D^2 \frac{\rho w^2}{2g_c} \end{aligned}$$

*Preliminary analysis indicated that, with this combination of parameters a reasonably compact, low power seal could be obtained. It also indicated that little improvement could be obtained by going beyond 7 to 9 stages.

The losses at this condition are(15),

$$h_p = \frac{2\pi tN}{396000} = \frac{2\pi N}{396000} \frac{Cm}{2g_c} (1.4 D/2)^5$$

so that

$$h_p/psi = 19.6 \times 10^{-6} D^3 N Cm$$

where Cm is the torque coefficient and at an interface ratio of 0.8 is approximately,

$$Cm \approx 1.063 \left(\frac{\rho V_o D_o}{2\mu g_c} \right)^{-0.25} = \frac{1.07}{\left(\frac{\rho V D}{\mu g_c} \right)^{0.25}}$$

Thus,

$$h_p/psi = 21 \times 10^{-6} \frac{D^3 N}{\left(\frac{\rho V D}{\mu g_c} \right)^{0.25}} \quad (4)$$

Comparison of the two equations, 3 and 4, reveals the inherent power advantage the viscoseal holds over the dynamic slinger; that is

$$\begin{aligned} \frac{(h_p/psi)_{dynamic}}{(h_p/psi)_{viscoseal}} &= \frac{21 \times 10^{-6} D^3 N \left(\frac{\rho V s}{\mu g_c} \right)^{0.61}}{3850 \times 10^{-6} D^2 s N \left(\frac{\rho V D}{\mu g_c} \right)^{0.25}} \\ &= 0.00545 \frac{D^{0.75}}{s^{0.39}} \left(\frac{\rho V}{\mu g_c} \right)^{0.39} = 0.081 D^{0.36} \left(\frac{\rho V}{\mu g_c} \right)^{0.39} \end{aligned}$$

$$\text{for } s = 0.001 D$$

which is a number always much greater than one (50 to 100) at the conditions being considered.

The seal losses are included in the generator computer design by inputting the friction coefficient corresponding to the above losses using the relations (12), (14),

$$\text{friction coefficient} = \frac{0.082}{Re^{0.43}}$$

and,

$$\frac{2 \text{ (torque)}}{\pi^3 \rho N^2 D^4 L} = \frac{1}{Re}$$

where

$$\text{torque} = \frac{(396000) \text{ (hp)}}{2\pi N} \quad (\text{in-lb})$$

The resulting bearing losses, seal losses and windage losses for the optimum first cycle designs are usually of the order of 0.2 to 2 percent of the electrical rating for the 0.4 to 1.2 MVA and 6 to 12 MVA sizes respectively.

2.3.3 Density in Rotor Cavity

In order to calculate the windage losses in the rotor cavity, it is necessary (among other things) to specify the static density of the gas surrounding the rotor. Since Holweck type molecular vacuum pumps can be utilized to reduce the pressure (and hence the density and windage) it becomes necessary to establish some generalized pump designs and corresponding performance upon which the reduced density may be based. The basic shaft seal configuration utilized is based on past SNAP-50/SPUR designs and is shown on figure 15. The configuration is a viscoseal with an overhung Holweck pump. The static rotor cavity density may be determined as a function of the static saturated vapor properties and the pressure differential across the pump. That is,

$$\rho_{\text{static}} = P_{\text{static}} (\rho/P)_{\text{base}}$$

$$= \left(P_{\text{base}} + \frac{\rho_{\text{base}} v^2}{2g_c} \right) (\rho/P)_{\text{fore}}$$

Figure 15. Basic Shaft Seal Configuration

(cont'd)

$$= \rho_{\text{sat}} \left(1 - \frac{\Delta P_H}{P_{\text{sat}}} \left\{ 1 + \frac{0.0074}{T} \left(\frac{D_{HN}}{1000} \right)^2 \right\} \right)$$

where

ΔP_H = pressure difference across pump, psi
 D_H = Holweck pump diameter, inches
 T = Absolute temperature, °R

Thus, we see that it is convenient to describe the generalized Holweck Pump performance (ΔP_H) in terms of pump diameter, shaft speed and coolant temperature.

The 500°F coolant condition is comparable to the coolant temperature on previous SNAP-50/SPUR studies and during these studies it was found that a 10:1 pressure ratio could easily be maintained across the pump. Thus, at 500°F, it was assumed that:

$$\Delta P_H \approx 0.9 P_{\text{sat}} \text{ at all speeds.}$$

At 800° and 1100°F, the saturated vapor pressure increases, the pumps are not as efficient and pressure ratios less than 10:1 are more likely. In order to assess the performance at 800° and 1100°F, typical Holweck pump designs were made and their performance calculated⁽¹²⁾; the results are shown on figures 16 and 17. With these data, the rotor cavity densities as shown on figure 18 were calculated.

There is very little variation in the 1100°F cases because the saturated vapor pressure is sufficient to prevent the pump from achieving any significant pressure ratios. In fact, most were 1.05:1 or less for the designs used; with design refinements, this could be improved but probably not much more than 2:1. At 800°F, the pressure ratios are of the order of 2:1 for the approximating designs used.

Figure 16. Typical Holweck Pump Performance in 800°F Potassium

Figure 17. Typical Holweck Pump Performance in 1100°F Potassium

Figure 18. Density of Potassium Vapor Surrounding Rotor

3. FIRST CYCLE GENERATOR DESIGN STUDY RESULTS

During the first cycle, both H-11 and NIVCO rotor alloys were used in designs using 800°F coolant to compare the properties of H-11 and NIVCO, e.g. figure 19. The designs at each temperature and rating were compared on the basis of generator electrical weight and on the basis of generator electrical weight plus estimated power plant and cooling system weight penalty. Weight penalty is determined from the generator efficiency and the specific weights of the power plant and cooling systems that supply and dissipate the losses. The first cycle study specific weights were preliminary estimates.

3.1 FREQUENCY STUDY

Figures 20 through 23 show curves of generator weight (with and without the weight penalties) versus frequency. Tables 2 and 3 and figures 20 and 21 for 400 kVA and 1200 kVA ratings show that generator electrical weight tends to be lowest in the 2000 to 3000 Hz range while generator weight plus system weight penalty is lowest in the 1000 to 2000 Hz range. Based on this data, a frequency of 2000 Hz was chosen for use in 400 and 1200 kVA designs in the subsequent speed and voltage studies.

Tables 4 and 5 and figures 22 and 23 for 6 MVA and 12 MVA ratings show a generator weight advantage in the 1000 to 2000 Hz range and a system weight advantage in the 1000 to 1400 Hz range. Therefore, a frequency of 1200 Hz was chosen for use in 6 and 12 MVA designs in the subsequent speed and voltage studies.

According to results shown in tables 2 through 5, the majority of 800°F designs using NIVCO rotor material exceeded the allowable stress limits while the 800°F designs using H-11 rotor material were satisfactory from a stress standpoint and also lighter in weight. The stress capabilities of NIVCO and H-11 are nearly equal at equal temperatures in the 800°F range (figure 19); however, the poorer magnetic properties of NIVCO require a larger rotor for a given rating resulting in greater stresses and a greater temperature drop in the rotor between the stressed portions and the coolant. For a given rating and coolant temperature, portions of a NIVCO rotor will operate at a higher temperature and stress than would be encountered using a smaller H-11 rotor, thus the stress capabilities of the NIVCO rotor will be lower than those of an equivalent H-11 rotor.

All 1100°F designs in the first frequency study exceeded the stress limits of the NIVCO rotor material at the chosen speeds. This prompted the choice of lower speed ranges for 1100°F designs in the subsequent speed study.

It was desirable to choose parameters for the preliminary designs which would limit the temperatures to ranges compatible with chosen materials.* Sixty-seven of the eighty designs in the first cycle frequency study equalled or exceeded the 1300°F winding temperature limit of Inconel 600-clad silver wire with Anadur insulation (used in 500° and 800°F designs) and the 1600°F winding temperature limit of dispersion-strengthened copper with alumina insulation (used on 1100°F designs). Therefore, the initially chosen ac current density of 9000 amperes per square inch was reduced to 7000 amps/in² for subsequent 500°F designs and to 6000 amps/in² for subsequent 800° and 1100°F designs.

The use of four ac conductors in parallel in the same slot resulted in the choice of a standard round wire by the computer design program for the smaller conductors required in the 400 and 1200 kVA designs. A single rectangular conductor was specified instead of four round conductors in subsequent 400 and 1200 kVA parametric designs to obtain better slot fill and better heat transfer characteristics.

A summary of the changes in chosen generator parameters and materials made as a result of the first cycle frequency study is shown in table 6.

3.2 SPEED STUDY

Following the frequency study, a speed study was conducted by calculating generator designs combining the following parameters and using the best frequencies from the frequency study.

*The computer program utilizes a fixed stator cooling configuration to analyze the winding temperatures. Although input data may be varied to circumvent normal routine and thereby account for variations in the cooling configuration, it was not done as part of the parametric screening studies in order to assess the relative severity of cooling problems associated with the various designs.

Figure 19. Creep Strength Versus Temperature For 0.2% Creep Strain in 10,000 Hours

Figure 20. 400 kVA Generator Frequency Study First Cycle Designs

Figure 21. 1200 kVA Generator Frequency Study
First Cycle Designs

Figure 22. 6 MVA Generator Frequency Study First Cycle Designs

Figure 23. 12 MVA Generator Frequency Study
First Cycle Designs

Table 2. Summary of 400 kVA Generator Designs, Frequency Study, First Cycle, 24000 RPM, 1000 Volts L-N

Design No.	Average Coolant Temp. (°F)	Frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
10501	500	800	761	89.2	2093	1611	0.94	H-11	22.68	15.15	105200	52122
02	↓	1200	530	89.4	1670	1450	0.96	↓	21.18	13.27	116400	46309
03		2000	432	88.7	1202	1432	0.89		20.42	12.12	122200	45032
04		2800	419	85.3	1322	1809	0.99		20.57	11.90	124000	46494
05	↓	3200	439	82.4	1321	2218	0.94	↓	20.97	11.91	124400	48665
10801	800	800	776	88.6	2882	1606	0.96	H-11	22.84	15.33	69100	53708
02	↓	1200	565	89.2	2042	1345	0.89	↓	21.64	13.58	76500	48203
03		2000	450	88.0	1602	1330	1.14		20.90	12.88	80200	44115
04		2800	472	84.5	1567	1672	0.90		21.48	12.50	81500	47794
05	↓	3200	462	83.3	1642	1762	0.98	↓	21.51	12.16	81600	49397
10806	800	800	866	85.0	4402	2016	1.05	Nivco	25.64	14.51	69100	82718
07	↓	1200	597	86.9	2782	1577	1.02	↓	22.94	12.82	76500	71064
08		2000	741	87.9	1436	1631	0.88		24.71	12.38	80200	91568
09		2800	621	82.9	1267	1961	1.17		24.01	12.19	81500	81175
10	↓	3200	670	82.3	1417	2050	0.92	↓	24.55	11.76	81600	88790
11101	1100	800	840	82.3	3952	2190	1.04	Nivco	25.26	14.06	14320	83040
02	↓	1200	569	85.6	2597	1619	1.03	↓	22.29	12.28	16380	71426
03		2000	490	86.0	1922	1494	0.93		21.97	10.76	16620	72235
04		2800	574	81.0	1817	2024	1.03		23.09	10.73	16930	85113
05	↓	3200	569	79.2	1727	2219	1.13	↓	23.22	11.20	16990	81482
* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.												

Table 3. Summary of 1200 kVA Generator Designs, Frequency Study, First Cycle 15000 RPM,
1000 Volts L-N

Design No.	Average Coolant Temp. (°F)	Pre-quency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
20501	500	500	3182	82.9	7252	7732	1.24	H-11	33.69	24.71	105200	50192
02	↓	1000	1494	91.8	2252	3394	1.18	↓	29.05	17.04	120300	37506
03	↓	2000	1326	90.6	1302	3479	1.13	↓	29.06	15.72	124400	37193
04	↓	2500	1361	89.9	1162	3660	0.99	↓	29.38	15.56	125000	38991
05	↓	3000	1390	87.6	1216	4839	1.09	↓	29.75	15.42	125000	40430
20801	800	500	3208	84.9	6282	6108	1.23	H-11	34.09	24.35	69100	51600
02	↓	1000	1538	90.3	2952	3288	1.22	↓	29.22	17.51	79000	38789
03	↓	2000	1327	90.8	1677	2967	1.13	↓	29.13	15.71	81600	37935
04	↓	2500	1275	89.9	1620	3075	0.99	↓	28.93	14.83	82000	39270
05	↓	3000	1411	88.0	1520	3611	1.04	↓	30.11	15.00	82000	42040
20806	800	500	3281	62.8	11840	>8000	1.28	Nivco	39.11	20.17	69100	80766
07	↓	1000	2111	90.3	3132	3861	0.88	↓	33.12	17.08	79000	66682
08	↓	2000	2418	90.1	1580	4198	0.97	↓	35.70	16.48	81600	79880
09	↓	2500	1198	41.5	-	>6000	1.24	↓	29.73	12.48	82000	54150
10	↓	3000	2172	86.2	1647	4772	1.06	↓	34.98	15.16	82000	80054
21101	1100	500	2996	49.8	11040	>8000	1.30	Nivco	38.80	18.76	14320	78599
02	↓	1000	2008	88.9	3072	3968	0.96	↓	32.51	16.88	16380	65243
03	↓	2000	1373	89.5	1932	3213	0.97	↓	30.21	14.00	16990	54953
04	↓	2500	1608	86.9	2022	3988	1.49	↓	31.71	13.98	17000	64437
05	↓	3000	2107	84.1	1822	5097	1.03	↓	34.43	14.85	17000	79736

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 4. Summary of 6 MVA Generator Designs, Frequency Study, First Cycle 10000 RPM,
1000 Volts L-N

Design No.	Average Coolant Temp. (°F)	Frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	Xd (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
30501	500	500	8722	83.7	7782	27222	1.20	H-11	54.29	23.16	116400	58803
02	↓	1000	5581	85.9	6012	20580	1.46	↓	45.80	22.16	123300	50783
03		2000	5878	86.4	3822	20277	1.13		47.46	21.80	125000	55295
04		2500	7378	85.9	2362	22377	0.83		48.67	26.48	125000	61279
05		3000	5418	78.3	-	33918	1.74	↓	45.40	30.33	125000	65890
30801	800	500	9106	84.2	9602	22605	1.17	H-11	56.29	22.43	76500	62415
02	↓	1000	5619	87.8	5842	15419	1.45	↓	46.17	21.86	81000	52124
03		2000	7890	89.5	2172	16189	0.78		48.72	28.56	82000	62236
04		2500	7733	88.7	2232	16732	0.85		49.05	27.47	82000	63809
05		3000	5380	83.6	-	19380	1.71	↓	45.57	28.45	82000	67337
30806	800	500	19460	79.2	5432	39460	1.09	Nivco	75.40	23.33	76500	150850
07	↓	1000	7935	86.5	5842	18935	1.17	↓	51.80	24.02	81000	86126
08		2000	8656	87.9	2597	18456	0.64		54.51	22.60	82000	98241
09		2500	11746	84.5	2502	24746	0.84		60.64	23.05	82000	125166
10		3000	12177	83.2	2492	26677	1.05	↓	61.18	23.44	82000	129484
31101	1100	500	14451	76.9	13570	35451	1.53	Nivco	68.93	21.79	15850	125608
02	↓	1000	7808	88.1	4782	17208	1.14	↓	51.88	23.28	16900	86812
03		2000	10186	85.1	3122	22186	1.19		60.85	19.02	17000	121320
04		2500	11759	84.0	2280	22759	0.82		61.07	22.43	17000	127501
05		3000	12354	82.1	2202	27354	1.16	↓	60.67	24.15	17000	129028

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 5. Summary of 12 MVA Generator Designs, Frequency Study, First Cycle, 8000 RPM, 1000 Volts L-N

Design No.	Average Coolant Temp. (°F)	Frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
40501	500	400	21355	83.0	6452	58855	1.56	H-11	77.60	26.06	116400	66680
02	↓	1200	13000	86.8	6652	39499	0.99	↓	59.55	29.14	124900	58713
03		2000	14375	84.1	7112	48874	0.83		63.38	27.22	125000	66422
04		2800	19211	1.1	-	-	5.86		71.45	26.50	125000	49357
05	↓	3200	17899	77.2	6232	77898	1.35	↓	63.67	35.20	125000	71483
40801	800	400	20298	85.8	6912	42298	1.39	H-11	76.09	25.61	76500	67814
02	↓	1200	13186	88.6	5432	30186	1.14	↓	59.16	30.29	81900	59444
03		2000	13050	84.1	6132	38050	1.59		58.45	31.62	82000	59401
04		2800	25638	84.9	-	49638	5.7		75.18	32.17	82000	49995
05	↓	3200	17535	83.1	4872	44535	1.31	↓	63.99	33.99	82000	73175
40806	800	400	50544	75.6	10560	>95000	1.21	Nivco	105.9	28.49	76500	185565
07	↓	1200	20917	84.7	6632	44917	1.28	↓	73.09	27.37	81900	116918
08		2000	26815	83.3	5632	53615	1.03		82.96	25.01	82000	151361
09		2800	11018	2.3	-	-	5.06		59.18	19.77	82000	96793
10	↓	3200		Design Failed				↓				
41101	1100	400	50997	1.7	11140	187072	1.19	Nivco	107.0	27.83	15850	187072
02	↓	1200	20346	86.9	5962	117895	1.09	↓	73.69	25.77	17000	117895
03		2000	13992	3.0	5602	96690	0.68		66.97	21.93	17000	96690
04		2800	10755	0.8	-	96756	5.1		58.81	20.40	17000	96756
05	↓	3200	15386	0.6	3422	101266	0.91	↓	67.66	24.10	17000	101266
* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.												

Table 6. Changes in Materials and Generator Parameters
Following First Cycle Frequency Study

Item	Initial Choice	Revised Choice	Reason for Change
Rotor material for 500°F and 800°F designs	H-11 or Nivco	H-11	Designs with Nivco rotors exceeded stress limits
Speed ranges to be studied for 1100°F designs:			
400 kVA	12000 to 30000 rpm	6000 to 15000 rpm	Stress limits or Nivco were exceeded at higher speeds.
1200 kVA	8000 to 20000 rpm	4000 to 10000 rpm	
6 MVA	4000 to 15000 rpm	2250 to 9000 rpm	
12 MVA	2000 to 12000 rpm	1000 to 6000 rpm	
Current Densities:			
500°F Designs	9000 a/in ²	7000 a/in ²	To reduce winding temperatures for compatibility with wire and insulation capabilities
800°F Designs	9000 a/in ²	6000 a/in ²	
1100°F Designs	9000 a/in ²	6000 a/in ²	
Conductors in parallel in same slot for 400 and 1200 kVA designs	4	1	To make feasible use of rectangular wire instead of round wire in the computer design iterations.

Generator Rating (MVA)	Coolant Temp. (°F)	Frequency (Hz)	Speed Range (rpm)
0.4	500	2000	12000 to 60000
0.4	800		12000 to 30000
0.4	1100		6000 to 15000
1.2	500		8000 to 40000
1.2	800		8000 to 20000
1.2	1100		4000 to 10000
6.0	500,800	1200	4500 to 18000
6.0	1100		2250 to 9000
12.0	500,800		2000 to 12000
12.0	1100		1000 to 6000

1000 volts line-to-neutral was used for all speed study designs.

The speed study designs were again compared on the basis of generator electrical weight and on the basis of generator electrical weight plus estimated power plant and cooling weight penalty.

Table 7 and figure 24 show that for 500° and 800°F coolant temperatures, 400 kVA generator weight and system weight penalties tend to decrease up to speeds of 30,000 rpm. Additional points calculated at speeds of 40,000 and 60,000 rpm for 500°F generators showed that little weight advantage was offered by these higher speeds; rotor material stress limits were exceeded at 60,000 rpm. For the 1100°F, 400 kVA designs, it was necessary to decrease the speed to 7500 rpm to maintain the rotor stress within the capabilities of NIVCO. No suitable 6000 rpm design was obtained because of saturation and excessive excitation requirements encountered using the standard fixed parameters chosen as constants throughout the study. Based on these results, a speed of 30,000 rpm for 500° and 800°F coolants and a speed of 7500 rpm for 1100°F coolant were chosen for the 400 kVA designs to be calculated in the voltage study.

Table 8 and figure 25 show that for 500°F, speeds in the 24,000 to 30,000 rpm range provide the lowest 1200 kVA generator and system weight with little advantage gained by operating above 30,000 rpm. Likewise, it appeared that little weight advantage could be gained above 20,000 rpm for the 800°F, 1200 kVA designs. All 1100°F, 1200 kVA designs failed stress above 5000 rpm and failed by only 10 percent at 5000 rpm. No suitable 4000 rpm, 1200 kVA design was obtained because of saturation and excessive excitation requirements encountered using the standard design inputs. Based on these results, a speed of 24,000 rpm for 500° and 800°F coolants and a speed of 4800 rpm for an 1100°F coolant were

chosen for the 1200 kVA designs to be calculated in the voltage study.

Table 9 and figure 26 show a 12,000 to 14,400 rpm speed range advantageous for 500° and 800°F, 6 MVA designs; however, the 800°F designs failed stress at 14,400 rpm. All 1100°F designs failed stress except those at the lower speed of 2250 rpm. A speed of 12,000 rpm for 500° and 800°F coolants and a speed of 2250 rpm for the 1100°F coolant were chosen for the 6 MVA voltage study designs.

Table 10 and figure 27 show the 8000 to 12,000 rpm range advantageous for 500° and 800°F, 12 MVA designs; however, the 800°F designs failed stress at 12,000 rpm. The three 1100°F designs which were satisfactorily calculated electrically, all failed stress. Complete 1100°F designs at 1000 and 2000 rpm were not obtained because of saturation and excessive excitation requirements encountered using the standard design inputs. A speed of 10,286 rpm (for 1200 Hz with 14 poles) was chosen for 500° and 800°F for the 12 MVA voltage study. A speed of 1000 rpm was arbitrarily chosen for further 1100°F, 12 MVA design trials to be run for the voltage study.

Whereas 67 of the 80 designs of the frequency study exceeded the temperature limits of the conductor insulations using an ac current density of 9000 amps/in² as previously discussed, only 24 of the 64 designs of the speed study exceeded the temperature limits with the ac current densities reduced to 7000 amps/in² for a 500°F coolant and 6000 amps/in² for 800° and 1100°F coolants. To further reduce the winding temperatures in subsequent designs, the assumption that stator slot voids are filled with a gas (with a thermal conductivity varying between 0.001 and 0.002 watts per inch-°C) was changed to the assumption that the stator slot voids will be filled with a high temperature filler material (with a thermal conductivity of not less than 0.015 watts per inch-°C) for all designs.

3.3 VOLTAGE STUDY

Following the frequency and speed studies, a voltage study was conducted by calculating designs combining the following parameters and using the frequency and speed combinations found best from the foregoing studies.

Design No.	Average Coolant Temp. (°F)	RPM	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
10510	500	12000	691	90.4	772	1501	0.89	H-11	23.49	12.99	125000	15880
09	↓	15000	563	88.6	920	1563	1.05	↓	22.37	12.30	124400	21330
08		20000	438	91.1	877	1178	1.01		20.27	11.84	123300	32644
07		24000	397	91.2	961	1137	0.91		19.50	11.62	122200	45016
06		30000	355	91.3	987	1085	0.93		18.66	11.50	120300	64091
11		40000	301	90.7	1172	1091	1.14		17.52	11.67	116400	98334
12	↓	60000	295	89.3	1559	1225	1.00	↓	16.86	11.98	105200	206829
10815	800	12000	635	89.8	1130	1375	1.05	H-11	23.17	12.04	82000	15714
14	↓	15000	563	88.9	1232	1363	1.04	↓	22.41	12.11	81600	21953
13		20000	443	91.1	1192	1073	1.00		20.41	11.80	81000	33213
12		24000	364	91.1	1267	994	1.09		19.06	11.36	80200	42930
11	↓	30000	363	91.4	1280	963	0.93	↓	18.85	11.51	79000	65456
11110	1100	6000	1134	Design Failed				Nivco				
09	↓	7500	1714	87.0	1282	2634	1.04	↓	32.07	13.67	17000	16999
08		10000	1229	88.0	1316	2079	1.04		29.10	12.75	17000	23651
07		12000	708	87.9	1390	1568	1.14		24.64	11.28	17000	23033
06	↓	15000	883	88.7	1362	1663	0.92	↓	26.08	12.17	16990	41559

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Design No.	Average Coolant Temp. (°F)	RPM	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
20510	500	8000	2433	91.9	842	4153	0.98	H-11	32.84	20.20	125000	16002
09	↓	10000	1801	91.3	893	3701	1.24	↓	30.80	17.70	125000	20535
08		12000	1752	92.1	952	3252	0.95		30.07	18.01	125000	29395
07		15000	1271	92.2	1002	2921	1.16		28.13	15.79	124400	37067
06		20000	982	92.2	1182	2632	1.39		25.72	15.08	123300	56509
11		30000	870	92.1	1390	2540	1.09		24.51	15.08	120300	117621
12	↓	40000	755	92.1	1637	2455	1.22	↓	23.18	14.73	116400	183007
20815	800	8000	2472	91.9	1137	3892	0.99	H-11	33.26	19.93	82000	16548
14	↓	10000	1836	91.3	1202	3376	1.24	↓	31.28	17.45	82000	21129
13		12000	1656	92.3	1199	2996	0.91		29.78	17.68	82000	28293
12		15000	1189	92.0	1371	2589	1.29		28.15	14.52	81600	39447
11	↓	20000	998	92.3	1472	2338	1.38	↓	25.94	14.94	81000	57732
21110	1100	4000	Design Failed					Nivco				
09	↓	5000	7045	89.8	1260	8805	1.14	↓	46.86	22.67	17000	18848
08		6000	5136	90.3	1287	6816	1.24		43.74	19.52	17000	22779
07		7500	3679	89.6	1326	5479	1.34		41.28	16.18	17000	31247
06	↓	10000	2887	90.4	1327	4537	1.29	↓	36.59	17.24	17000	41556

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 9. Summary of 6 MVA Generator Designs, Speed Study, First Cycle, 1200 Hz,
1000 Volts L-N

Design No.	Average Coolant Temp. (°F)	RPM	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
30510	500	4500	15746	89.9	1062	25346	1.12	H-11	58.06	38.15	125000	17667
09	↓	9000	8318	92.2	1389	15318	1.16	↓	48.39	30.83	124400	48183
08		12000	6113	92.7	1782	12613	1.47		44.61	27.48	123300	72503
07		14400	5379	92.3	2862	12279	1.07		44.20	24.07	122200	100960
06		18000	4289	90.4	5352	13289	1.27	↓	41.33	21.75	120300	141602
30815	800	4500	15962	90.4	1316	23542	1.13	H-11			82000	18250
14	↓	9000	8360	92.9	1551	13660	1.14	↓	49.00	30.21	81600	48932
13		12000	6140	93.2	2027	11340	1.45		45.09	26.96	81000	74080
12		14400	5919	93.0	2287	11119	1.14		45.43	25.22	80200	108687
11		18000	4457	91.6	4362	10957	1.30	↓	41.81	22.41	79000	146211
31110	1100	2250	66141	81.4	1347	81942	0.99	Nivco	94.71	47.03	17000	16800
09	↓	4500	19512	90.5	1421	26712	0.66	↓	67.10	31.23	17000	30938
08		6000	12036	90.9	1693	18836	1.41		58.86	26.75	17000	40329
07		7200	9701	92.1	1654	15501	1.24		55.73	24.57	17000	52446
06		9000	8361	92.1	1780	14161	1.36	↓	52.44	24.73	16990	70855

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 10. Summary of 12 MVA Generator Designs, Speed Study, First Cycle, 1200 Hz, 1000 Volts L-N

Design No.	Average Coolant Temp. (°F)	RPM	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)
40510	500	2000	51447	85.2	-	83447	2.18	H-11	90.3	43.74	125000	7690
09	↓	4000	28358	89.2	1844	48358	0.92	↓	74.20	38.79	125000	23366
08		6000	20229	90.4	1740	37829	1.46		63.39	41.15	125000	39232
07		8000	13554	90.6	3742	30554	1.03		59.80	30.50	124900	59484
06	↓	12000	9769	90.7	5782	26769	1.08	↓	54.53	26.86	123300	109985
40815	800	2000	73478	76.8	-	123478	2.22	H-11	104.5	43.98	82000	8207
14	↓	4000	32792	90.7	1787	46392	1.01	↓	77.32	41.20	82000	25974
13		6000	20195	92.0	1972	31694	1.44		63.97	40.21	82000	40144
12		8000	14064	93.0	2652	23864	1.06		60.32	31.45	81900	61386
11	↓	12000	10760	92.4	3882	21660	1.15	↓	56.16	28.02	81000	119116
41110	1100	1000	Design Failed					Nivco				
09	↓	2000	Design Failed					↓				
08		3000	73543	86.4	1532	93543	1.47		101.5	46.23	17000	34761
07		4000	42104	89.5	1652	57004	1.48	↓	88.93	35.59	17000	44939
06	↓	6000	Design Failed									
* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Figure 24. 400 kVA Generator Speed Study First Cycle Designs

Figure 25. 1200 kVA Generator Speed Study First Cycle Designs

Figure 26. 6 MVA Generator Speed Study Cycle Designs

Figure 27. 12 MVA Generator Speed Study Cycle Designs

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Frequency (Hz)</u>	<u>RPM</u>	<u>Voltage Points (Line-to- Neutral)</u>
0.4	500,800	2000	30000	
0.4	1100	2000	7500	500,1000,1500,&
1.2	500,800	2000	24000	2140 for all rat-
1.2	1100	2000	4800	ings & temperatures
6.0	500,800	1200	12000	plus additional
6.0	1100	1200	2250	250V designs for
12.0	500,800	1200	10286	6 MVA, 500° & 800°F
12.0	1100	1200	1000	

In this study, not only the slot cell thickness varied with voltage but also the number of conductor turns in series varied. This latter varying parameter in turn varied the direct-axis synchronous reactance (X_d) for different combinations of design parameters. The following values of X_d were initially chosen to reasonably limit (1) voltage unbalance for unbalanced loads, (2) excitation power, (3) transients, and (4) more use of field power for overloads.

<u>Rating (MVA)</u>	<u>Desired X_d (per unit)</u>
0.4	1.0
1.2	1.2
6.0	1.3
12.0	1.3

Some lightweight designs were obtained with considerably higher X_d values than those shown above and some high-weight designs were obtained with considerably lower X_d values than those shown above. In order to choose designs with comparable performance, designs with X_d values varying more than 10 percent from the chosen values were not used in choosing the voltages which provided the best generator and system weight advantages.

As shown in tables 11 through 13 and figures 28 through 30, 500 volts results in generator and/or system weight advantages for the following designs.

400 kVA, 500° and 800°F
1200 kVA, 1100°F
6 MVA, 500°, 800°, and 1100°F

The 400 kVA, 1100°F design showed a generator electrical weight advantage at 1000 volts and about equal system weights at

500 and 1000 volts due in part to the higher weight of the lower X_d , 500 volt design; the 1200 kVA, 500 volt, 500° and 800°F designs had low X_d 's and thus higher weights at 500 volts. Additional 250 volt designs calculated for 6 MVA, 500° and 800°F, confirmed the advantage of 500 volts. Based on the trends shown by the designs calculated, 500 volts line-to-neutral was chosen for further design studies on 0.4, 1.2, and 6.0 MVA designs.

As shown in table 14 and figure 31, the 12 MVA designs showed a system weight advantage at 1000 volts even though the low X_d , 1000 volt designs had higher electrical weight than did the 500 volt designs. A voltage of 1000 volts was chosen for further design studies on 12 MVA designs.

Twenty-one of fifty designs calculated for the voltage study exceeded the temperature limitations of the wire insulations; however, twelve of those twenty-one were 1500 or 2140 volt designs. The reduced current densities chosen following the frequency study and the increased thermal conductivity (for filler material in the slots) used following the voltage study appeared sufficient at this point in the program to maintain the temperatures at acceptable levels for 500 and 1000 volt designs in following studies.

3.4 SUMMARY OF FIRST CYCLE GENERATOR STUDIES

Table 15 summarizes the frequencies, speeds, and voltages found to offer system weight advantages. These values were used in the further refining design studies.

Changes in fixed parameters and choice of materials recommended for use in second cycle studies, based on improvements found in the first cycle study, were:

- a) Reduction of ac current densities to 7000 amps/in² for 500°F designs and to 6000 amps/in² for 800° and 1100°F designs instead of the 9000 amps/in² value initially chosen for all temperatures.
- b) Use of H-11 rotor material instead of NIVCO in designs for 800°F coolant temperatures.
- c) Reduction of speeds to 7500 rpm and lower in designs for 1100°F coolant temperatures to limit rotor stresses.
- d) Use of a single rectangular conductor instead of four round conductors in parallel in each stator slot of 400 and 1200 kVA designs to obtain better slot fill in the computer configurations.
- e) Use of alternate experimental rotor materials for trial 1100°F designs. The majority of 1100°F designs in the first cycle studies equalled or exceeded the stress limits of NIVCO rotor material.

Table 11. Summary of 400 kVA Generator Designs, Voltage Study, First Cycle, 2000 Hz

Design No.	Average Coolant Temp. (°F)	L-W Volts	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)	RPM
10513	500	500	346	91.7	822	1026	0.90	H-11	18.48	11.50	120300	65964	30000
14	↓	1000	355	91.5	837	1055	0.93	↓	18.66	11.50	↓	64091	↓
15	↓	1500	335	90.4	942	1145	1.09	↓	18.45	11.24	↓	60994	↓
16	↓	2140	408	87.8	1147	1508	1.01	↓	19.99	12.16	↓	63805	↓
10816	800	500	349	90.9	1167	999	0.89	H-11	18.56	11.42	79000	67000	30000
17	↓	1000	363	90.7	1184	1023	0.93	↓	18.85	11.51	↓	65456	↓
18	↓	1500	345	89.9	1290	1065	1.08	↓	18.68	11.39	↓	62119	↓
19	↓	2140	433	86.9	1522	1413	1.02	↓	20.45	12.54	↓	65560	↓
11111	1100	500	1879	88.5	1249	2689	0.88	Nivco	31.14	15.78	17000	17157	7500
12	↓	1000	1714	87.0	1280	2664	1.04	↓	32.07	13.67	↓	16999	↓
13	↓	1500	1770	84.9	1282	2870	1.02	↓	32.83	13.68	↓	17342	↓
14	↓	2140	1956	84.3	1302	3136	0.99	↓	33.73	14.44	↓	17935	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 12. Summary of 1200 kVA Generator Designs, Voltage Study, First Cycle, 2000 Hz

Design No.	Average Coolant Temp. (°F)	L-N Volts	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)	RPM
20513	500	500	1049	92.6	902	2599	0.86	H-11	25.82	16.08	122200	84300	24000
14	↓	1000	889	92.7	960	2529	1.33	↓	24.81	14.87	↓	76457	↓
15	↓	1500	917	92.5	1016	2517	1.24	↓	25.24	14.50	↓	75846	↓
16	↓	2140	906	91.3	1212	2806	1.22	↓	25.60	13.75	↓	74712	↓
20816	800	500	1060	92.6	1247	2360	0.85	H-11	26.11	15.83	80200	85988	24000
17	↓	1000	911	92.5	1292	2211	1.32	↓	25.07	14.80	↓	78072	↓
18	↓	1500	919	92.6	1370	2219	1.21	↓	25.35	14.37	↓	76602	↓
19	↓	2140	1030	91.6	1370	2530	1.20	↓	26.30	14.97	↓	80527	↓
21111	1100	500	7129	85.4	1312	9829	1.46	Nivco	45.01	24.16	17000	16671	4800
12	↓	1000	7531	89.3	1350	9411	0.98	↓	45.64	24.72	↓	16640	↓
13	↓	1500	8305	88.2	1270	10405	1.14	↓	47.41	25.25	↓	17590	↓
14	↓	2140	9096	87.2	1270	11396	1.19	↓	48.23	26.86	↓	18290	↓
* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.													

Table 13. Summary of 6 MVA Generator Designs, Voltage Study, First Cycle, 1200 Hz

Design No.	Average Coolant Temp. (°F)	L-N Volts	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)	RPM
30515	500	250	5561	77.6	6082	35661	0.83	H-11	47.64	18.75	123300	78629	12000
11	↓	500	5093	92.3	1861	12093	1.12	↓	44.60	21.44	↓	71587	↓
12	↓	1000	6713	92.8	1223	13113	1.27	↓	45.51	28.88	↓	76910	↓
13	↓	1500	6956	91.5	1647	14856	1.26	↓	46.66	28.29	↓	74487	↓
14	↓	2140	7241	92.2	1404	14240	1.23	↓	47.42	28.40	↓	76042	↓
30820	800	250	8606	82.8	3462	23607	1.23	H-11	53.73	22.44	81000	101969	12000
16	↓	500	5318	92.5	1852	11018	1.14	↓	45.20	22.04	↓	74571	↓
17	↓	1000	6685	93.3	1524	11685	1.24	↓	45.85	28.30	↓	77837	↓
18	↓	1500	7221	92.2	1757	13121	1.29	↓	47.35	28.65	↓	76953	↓
19	↓	2140	7310	92.8	1694	12510	1.21	↓	48.05	27.93	↓	77868	↓
31111	1100	500	19999	93.0	-	24999	1.84	Nivco	68.12	25.72	17000	13950	2250
12	↓	1000	66142	81.7	1337	81742	0.99	↓	94.71	47.03	↓	16800	↓
13	↓	1500	75023	79.2	1382	93022	1.18	↓	98.31	49.97	↓	17977	↓
14	↓	2140	68590	80.6	1397	85290	1.28	↓	101.5	41.89	↓	18880	↓
* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.													

Table 14. Summary of 12 MVA Generator Designs, Voltage Study, First Cycle, 1200 Hz

Design No.	Average Coolant Temp. (°F)	L-N Volts	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in)	Electrical Length (in)	Maximum Allowable Rotor Stress* (psi)	Average Maximum Rotor Stress (psi)	RPM
40511	500	500	9480	87.2	5122	34980	1.44	H-11	54.69	25.63	124000	82499	10286
12	↓	1000	11895	93.3	1817	23395	0.98	↓	56.87	30.15	↓	89426	↓
13	↓	1500	13783	93.1	1542	25583	1.23	↓	56.55	36.27	↓	89766	↓
14	↓	2140	16453	92.0	1726	30453	1.07	↓	59.27	38.94	↓	95072	↓
40816	800	500	9641	88.1	4532	28641	1.43	H-11	55.39	25.22	81500	85014	10286
17	↓	1000	12011	93.8	2012	20711	0.97	↓	57.53	27.64	↓	91690	↓
18	↓	1500	13981	93.5	1787	23181	1.21	↓	57.42	35.52	↓	92518	↓
19	↓	2140	16540	92.7	1942	26840	1.06	↓	60.01	38.16	↓	97556	↓
41111	1100	500	Design	Failed				Nivco			17000		1000
12	↓	1000	Design	Failed				↓			↓		↓
13	↓	1500	765891	59.2	1563	>800000	0.84	↓	223.7	82.62	↓	19936	↓
14	↓	2140	145005	56.9	-	>200000		↓	120.3	85.36	↓	5377	↓
* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.													

Table 15. Summary of Results from First Cycle
Generator Design Studies

Rating (MVA)	Coolant Temp. (°F)	Values Chosen for Subsequent Studies		
		Freq (Hz)	Speed (rpm)	Volts L-N
0.4	500 & 800	2000	30000	500
0.4	1100	2000	7500	500
1.2	500 & 800	2000	24000	500
1.2	1100	2000	4800	500
6.0	500 & 800	1200	12000	500
6.0	1100	1200	2250	500
12.0	500 & 800	1200	10000	1000
12.0	1100	1200	1000	1000

Figure 28. 400 kVA Generator Voltage Study First Cycle Designs

Figure 29. 1200 kVA Generator Voltage Study First Cycle Designs

Figure 30. 6 MVA Generator Voltage Study First Cycle Designs

Figure 31. 12 MVA Generator Voltage Study First Cycle Designs

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4. SECOND CYCLE GENERATOR DESIGN STUDY

Results from the first cycle study indicated the majority of 1100°F coolant temperature designs exceeded the stress limits of NIVCO rotor material at all but unreasonably low speeds. For this reason, a series of trial designs were performed for the 1100°F coolant condition using an experimental alloy 1-B-S-1.* The experimental alloy proved to be superior to NIVCO for the 1100°F coolant temperature with regard to high temperature strength and magnetic properties. Results of the trial designs and first cycle speed studies were combined to arrive at speed selections with the 1-B-S-1 rotor alloy for use in the second cycle 1100°F coolant temperature frequency study.

The specific parametric points for the second cycle frequency study are presented in the the following table. The selections of voltage and speed were based on results of the first cycle parametric studies except for the 1100°F speed points as mentioned previously.

Generator Rating (MVA)	Coolant Temp. (°F)	Volts	RPM	Frequency Points (Hz)
0.4	500	500	30000	1000,1500,2000,2500,3000
1.2	&	500	24000	800,1200,1600,2000,2800
6.0	800	500	12000	800,1200,1600,2000,2800
12.0		1000	10000	833,1000,1165,1500,2000
0.4	1100	500	12000	800,1000,2000,2800,3200
1.2	1100	500	10000	500,1000,2000,2500,3000
6.0	1100	500	8000	400, 800,1200,1600,2000
12.0	1100	1000	6000	400, 800,1200,1600,2000

Following the frequency study, a speed study was conducted combining the following parameters and using the best frequencies from the frequency study.

* Refer to page 7 for material composition.

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Volts</u>	<u>Freq. (Hz)</u>	<u>Speed Range (rpm) **</u>
0.4	500	500	*	15000 to 40000
1.2	500	500	*	12000 to 30000
6.0	500	500	*	9000 to 18000
12.0	500	1000	*	6000 to 15000
0.4	800	500	*	15000 to 40000
1.2	800	500	*	12000 to 32000
6.0	800	500	*	9000 to 18000
12.0	800	1000	*	4000 to 10000
0.4	1100	500	*	10000 to 18000
1.2	1100	500	*	8000 to 15000
6.0	1100	500	*	5000 to 10000
12.0	1100	1000	*	3000 to 6000

*Chosen from preceding frequency study.

**The exact speed points will depend on selected frequencies.

Following the frequency and speed studies, a voltage study was conducted by calculating designs and combining the following parameters, using the frequency and speed combinations found best from the foregoing studies.

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Freq. (Hz)</u>	<u>Speed (rpm)</u>	<u>Volts</u>
0.4	500,800,1100	*	*	500,1000,1500,2140
1.2	500,800,1100	*	*	500,1000,1500,2140
6.0	500,800,1100	*	*	500,1000,1500,2140
12.0	500,800,1100	*	*	500,1000,1500,2140

*Chosen from preceding frequency and speed studies.

5. SECOND CYCLE GENERATOR DESIGN STUDY RESULTS

The designs at each temperature and rating are compared on the basis of generator electrical weight and on the basis of generator electrical weight plus a power plant and cooling systems weight penalty. Power plant and cooling system weight penalty was determined from the specific weight of the systems supplying shaft power and rejecting electrical losses(10).

5.1 FREQUENCY STUDY

Figures 32 through 35 shows curves of generator electrical weight and generator electrical weight plus system weight penalty as a function of frequency. Figure 32 and table 16 show the trends for the 0.4 MVA rating. Generator electrical weight tends to be lowest in the 2000 to 3000 Hz range for the 500° and 800°F coolant temperatures while generator weight plus system weight penalty is minimum at a frequency of 2000 Hz. For the 1100°F coolant temperature, minimum generator weight plus weight penalty occurs at 1500 Hz. Based on these results, a frequency of 2000 Hz was chosen for use in the 0.4 MVA, 500° and 800°F speed and voltage studies, and a frequency of 1500 Hz was chosen for the 1100°F speed and voltage studies.

Figure 33 and table 17 show the trends for the 1.2 MVA rating. For the 500° and 800°F coolant temperature, generator weight plus system weight penalty is lowest in the 1200 Hz to 2000 Hz frequency range. For the 1100°F coolant temperature, minimum generator weight plus system weight penalty occurs at a frequency of 1200 Hz. A frequency of 1600 Hz was chosen for the subsequent 1.2 MVA speed and voltage studies at coolant temperatures of 500° and 800°F, and a frequency of 1200 Hz was chosen for the 1100°F coolant temperature.

Figure 34 and tables 18 and 19 show the trends for the 6.0 and 12.0 MVA ratings respectively. For the 500° and 800°F coolant temperatures, generator weight plus system weight penalty is lowest in the 1200 Hz to 1600 Hz frequency range. For the 1100°F coolant temperature, minimum generator weight plus system weight penalty occurs at a frequency of 800 Hz. A frequency of 1200 Hz was chosen for the subsequent 6.0 and 12.0 MVA speed and voltage studies at coolant temperatures of 500° and 800°F and a frequency of 800 Hz was chosen for the 1100°F coolant temperature.

A summary of optimum generator frequencies resulting from the second cycle frequency study is presented below.

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Frequency (Hz)</u>
0.4	500	2000
1.2	800	1600
6.0	800	1200
12.0		1200
0.4	1100	1500
1.2	1100	1200
6.0	1100	800
12.0	1100	800

5.2 SPEED STUDY

Following the frequency study, a speed study was conducted combining the following parameters and using the best frequencies from the frequency study.

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Volts</u>	<u>Freq. (Hz)</u>	<u>RPM</u>
0.4	500	500	2000	15000, 20000, 24000, 30000, 40000
1.2	500	500	1600	12000, 16000, 19200, 24000, 30000
6.0	500	500	1200	9000, 10286, 12000, 14400, 18000
12.0	500	1000	1200	6000, 8000, 10286, 12000, 14400
0.4	800	500	2000	15000, 20000, 24000, 30000, 40000
1.2	800	500	1600	12000, 16000, 19200, 24000, 32000
6.0	800	500	1200	9000, 10286, 12000, 14400, 18000
12.0	800	1000	1200	4000, 6000, 8000, 9000, 10286
0.4	1100	500	1500	10000, 11250, 12857, 15000, 18000
1.2	1100	500	1200	8000, 9000, 10286, 12000, 14400
6.0	1100	500	800	5333, 6000, 6857, 8000, 9600
12.0	1100	1000	800	3200, 4000, 4800, 5333, 6000

Figure 36 and table 20 show that for the 0.4 MVA rating, generator weight and generator weight plus system weight penalty decrease with increasing speed. The approximate maximum allowable speed based on rotor stress considerations is 42,000 rpm for the 500°F designs, 34,000 rpm for the 800°F designs, and 24,000

Figure 32. 0.4 MVA Generator Frequency Study - Second Cycle

Figure 33. 1.2 MVA Generator Frequency Study - Second Cycle

Figure 34. 6 MVA Generator Frequency Study - Second Cycle

Figure 35. 12 MVA Generator Frequency Study - Second Cycle

Figure 36. 0.4 MVA Generator Speed Study - Second Cycle

Table 16. Summary of 400 kVA Generator Designs, Frequency Study, Second Cycle, 500 Volts, L-N

Design No.	Average Coolant Temp. (°F)	Pre-frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM
10520	500	1000	582	91.3	1172	1272	1.05	H-11	20.7	15.2	105200	73900	30000
21	↓	1500	427	91.7	892	1087	0.98	↓	19.5	12.9	116400	67869	↓
22	↓	2000	346	91.7	814	1006	0.90	↓	18.5	11.5	120300	65964	↓
23	↓	2500	315	90.8	804	1055	0.94	↓	18.1	10.9	122200	65402	↓
24	↓	3000	288	89.5	812	1138	0.90	↓	17.8	10.2	123300	65096	↓
10820	800	1000	598	91.3	1512	1003	1.08	H-11	20.7	15.8	69100	75673	30000
21	↓	1500	416	91.8	-----	821	0.97	↓	19.3	12.8	76500	68334	↓
22	↓	2000	349	91.6	1142	764	0.89	↓	18.6	11.4	79000	67000	↓
23	↓	2500	322	90.5	1140	797	0.92	↓	18.3	10.9	80200	66369	↓
24	↓	3000	303	89.2	1135	853	0.97	↓	18.1	10.6	81000	66033	↓
11120	1100	800	1062	92.4	1422	1372	0.81	1-B-S-1	24.9	16.3	57750	23822	12000
21	↓	1000	831	92.4	1373	1141	0.86	↓	24.2	13.9	58600	21716	↓
22	↓	2000	755	91.2	1290	1115	0.74	↓	24.0	12.4	60000	21858	↓
23	↓	2800	851	87.8	1254	1351	0.81	↓	24.4	13.2	60000	23568	↓
24	↓	3200	911	85.2	1259	1561	0.86	↓	24.8	13.4	60000	24861	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 17. Summary of 1.2 MVA Generator Designs, Frequency Study, Second Cycle, 500 Volts, L-N

Design No.	Average Coolant Temp. (°F)	Pre-frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM
20520	500	800	1330	68.2	4452	11330	1.28	H-11	30.3	13.2	105200	80715	24000
21	↓	1200	1179	93.4	1142	2629	1.34	↓	26.0	17.8	116400	82940	↓
22	↓	1600	1008	92.3	1017	2708	1.20	↓	25.5	16.1	120300	81050	↓
23	↓	2000	964	92.0	974	2764	1.03	↓	25.3	15.6	122200	79653	↓
24	↓	2800	794	90.7	939	2894	1.27	↓	24.3	13.7	124000	79402	↓
20820	800	800	2169	77.5	3272	5869	1.28	H-11	34.0	15.0	69100	123879	24000
21	↓	1200	1353	93.2	1422	2298	1.12	↓	26.9	18.9	76500	91920	↓
22	↓	1600	1013	91.6	1402	2203	1.18	↓	25.7	15.8	79000	82156	↓
23	↓	2000	917	90.0	1582	2337	1.22	↓	25.1	15.1	80200	77694	↓
24	↓	2800	818	89.4	1322	2318	1.27	↓	24.6	13.6	81500	81205	↓
21120	1100	500	3857	90.3	2032	4987	1.17	1-B-S-1	35.7	25.3	55900	37104	10000
21	↓	1000	2192	93.4	1464	2942	1.09	↓	32.0	18.8	59600	28376	↓
22	↓	2000	3468	91.3	1282	4468	1.96	↓	36.0	21.3	60000	40874	↓
23	↓	2500	3171	85.7	1392	4921	1.24	↓	35.5	19.8	60000	39440	↓
24	↓	3000	3233	82.8	1414	5433	1.13	↓	36.3	19.0	60000	41167	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 18. Summary of 6 MVA Generator Designs, Frequency Study, Second Cycle, 500 Volts, L-N

Design No.	Average Coolant Temp. (°F)	Frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM
30520	500	800	5397	86.5	4892	19397	1.41	H-11	45.6	21.0	120300	71516	12000
21	↓	1200	5094	92.2	1874	12794	1.12	↓	44.6	21.4	123300	71587	↓
22	↓	1600	5782	92.3	1262	13382	1.06	↓	46.5	23.0	124400	79253	↓
23	↓	2000	4791	88.4	2052	16791	1.23	↓	44.5	20.4	125000	73297	↓
24	↓	2800	5564	90.0	1217	15564	1.21	↓	46.1	22.6	125000	79185	↓
30822	800	800	5551	87.0	3872	13351	1.42	H-11	46.1	21.4	79000	73969	12000
23	↓	1200	5318	92.4	1872	9718	1.14	↓	45.2	22.0	81000	74571	↓
24	↓	1600	6283	92.4	1492	10683	1.12	↓	47.5	24.0	81600	84138	↓
25	↓	2000	5373	90.0	1932	11173	1.27	↓	46.4	20.7	82000	79770	↓
26	↓	2800	10076	86.9	1516	18076	0.84	↓	56.5	23.7	82000	117908	↓
31120	1100	400	14920	84.1	5532	22920	1.35	1-B-S-1	64.2	26.4	55900	67952	8000
21	↓	800	9115	90.9	1980	13415	1.56	↓	52.1	25.9	59600	53820	↓
22	↓	1200	9861	91.7	1542	13761	1.36	↓	52.0	28.7	59900	55822	↓
23	↓	1600	10618	91.3	1480	14718	1.26	↓	52.3	30.6	60000	57541	↓
24	↓	2000	11613	90.5	1450	16064	1.19	↓	54.0	30.8	60000	61564	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Design No.	Average Coolant Temp. (°F)	Pre-Frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM
40520	500	833	12361	92.0	3542	28361	1.16	H-11	58.3	28.4	122200	86497	10000
21	↓	1000	11781	93.0	2492	25781	1.01	↓	57.3	28.6	123300	84102	↓
22	↓	1165	12238	93.2	1712	25538	0.97	↓	57.3	30.4	124000	85780	↓
23	↓	1500	10750	92.6	1767	25250	1.13	↓	56.0	28.4	124900	82196	↓
24	↓	2000	18255	80.1	--	58755	1.11	↓	68.6	28.2	125000	116475	↓
24A	↓	2000	11518	91.8	1572	27518	0.71	↓	58.7	26.5	125000	87424	↓
40820	800	833	11250	92.6	3212	19450	1.51	H-11	55.7	29.8	80200	81039	10000
21	↓	1000	11324	93.3	2197	18624	1.94	↓	53.8	33.9	81000	78314	↓
22	↓	1165	12358	93.8	1987	19158	0.96	↓	58.0	29.9	81500	87972	↓
23	↓	1500	11394	93.2	1816	18794	1.17	↓	56.8	29.5	81900	85792	↓
24	↓	2000	13086	93.2	1460	20486	0.76	↓	59.5	30.3	82000	94282	↓
41120	1100	400	46539	79.3	6802	68539	1.22	1-B-S-1	92.6	35.4	57750	91687	6000
21	↓	800	21585	92.9	2862	28085	1.28	↓	67.3	35.7	59900	53993	↓
22	↓	1200	23257	92.4	1722	30157	1.26	↓	70.4	34.7	6000	59617	↓
23	↓	1600	32701	90.0	1692	42001	1.08	↓	78.6	36.7	6000	76685	↓
24	↓	2000	25137	--	--	--	1.09	↓	79.5	26.3	6000	68819	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 20. Summary of 400 kVA Generator Design, Speed Study, Second Cycle, 500 Volts, L-N

Design No.	Average Coolant Temp. (°F)	RPM	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	Frequency (Hz)
10525	500	40000	294	91.3	939	984	1.09	H-11	17.3	11.5	116400	105131	2000
26	↓	30000	346	91.7	--	1006	0.90	↓	18.5	11.5	120300	65964	↓
27	↓	24000	357	91.5	809	1027	1.13	↓	18.8	11.4	122200	43375	↓
28	↓	20000	393	91.3	779	1083	1.17	↓	19.5	11.5	123300	31923	↓
29	↓	15000	525	91.5	694	1195	0.92	↓	21.6	12.1	124400	21654	↓
10825	800	40000	314	91.5	1272	734	1.04	H-11	17.7	12.0	76500	105938	2000
26	↓	30000	329	91.2	--	769	1.09	↓	18.2	11.4	79000	64236	↓
27	↓	24000	363	91.3	1145	798	1.12	↓	18.9	11.3	80200	44245	↓
28	↓	20000	430	91.4	1085	860	0.96	↓	20.1	11.8	81000	33142	↓
29	↓	15000	528	91.3	1012	963	0.91	↓	21.7	12.0	81600	21935	↓
11125A	1100	22500	426	90.4	1544	826	1.26	1-B-S-1	20.0	11.1	57750	54861	1500
25	↓	18000	536	91.8	1400	871	0.93	↓	21.5	11.9	38600	38378	↓
26	↓	15000	561	91.6	1394	906	0.97	↓	22.1	11.3	59600	28221	↓
27	↓	12857	659	91.9	1340	989	0.91	↓	23.1	12.1	59800	22379	↓
28	↓	11250	705	91.2	1346	1065	1.02	↓	23.8	11.9	59900	18188	↓
29	↓	10000	769	90.7	1308	1154	0.95	↓	24.7	11.5	59900	19855	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

rpm for the 1100°F coolant temperature designs. Speeds above 20,000 rpm for the 1100°F designs offer little weight advantage. Based on these results, speeds of 40,000 rpm, 30,000 rpm, and 18,000 rpm were chosen for coolant temperatures of 500°, 800°, and 1100°F, respectively.

Figure 37 and table 21 show the trends for the 1.2 MVA ratings. Generator weight decreases with increasing speed for all coolant temperature designs. Generator weight plus system weight penalty decreases with increasing speed for the 500° and 800°F designs but increases beyond a speed of 15,000 rpm for the 1100°F coolant temperature due to decreasing efficiency. The approximate maximum allowable speeds based on rotor stress considerations is 30,000, 21,500, and 15,500 rpm for the 500°, 800°, and 1100°F designs, respectively. Speeds chosen for the subsequent voltage study based on these results are 24,000, 19,200, and 12,000 rpm for the 500°, 800°, and 1100°F coolant temperatures, respectively.

Speed ranges of 12,000 rpm to 14,400 rpm and 10,286 rpm to 14,400 rpm, provides the lowest generator weight plus system weight penalty for 500° and 800°F, 6 MVA generator ratings, respectively, as shown by figure 38 and table 22. Maximum allowable rotor speed based on stress considerations is approximately 17,000 rpm for the 500°F designs and 12,800 rpm for the 800°F designs. Maximum speed for the 1100°F designs is approximately 8400 rpm. Based on these results, a rotor speed of 12,000 rpm was chosen for coolant temperatures of 500° and 800°F and a speed of 6857 rpm (for 800 Hz with 14 poles) was chosen for the 1100°F coolant temperature.

Figure 39 and table 23 show the trends for the 12 MVA generator rating. For coolant temperatures of 500°, 800°, and 1100°F, the speed ranges providing minimum generator weight plus system weight penalty are 10,000 rpm to 14,400 rpm, 8000 rpm to 10,286 rpm, and 5000 rpm to 6000 rpm, respectively. Maximum allowable rotor speed is approximately 12,900 rpm for the 500°F coolant, 10,200 rpm for the 800°F coolant, and 6200 rpm for the 1100°F coolant temperature.

Based on these results, speeds of 12,000, 8000, and 5333 rpm were chosen for coolant temperatures of 500°, 800°, and 1100°F, respectively.

A summary of optimum generator speeds resulting from the second cycle speed studies is presented on the following page.

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>RPM</u>
0.4	500	40000
1.2	500	24000
6.0	500	12000
12.0	500	12000
0.4	800	30000
1.2	800	19200
6.0	800	12000
12.0	800	8000
0.4	1100	18000
1.2	1100	12000
6.0	1100	6857
12.0	1100	5333

5.3 VOLTAGE STUDY

Following the frequency and speed studies, a voltage study was conducted by calculating designs and combining the following parameters, using the frequency and speed combinations found best from the foregoing studies.

<u>Generator Rating (MVA)</u>	<u>Coolant Temp. (°F)</u>	<u>Freq. (Hz)</u>	<u>RPM</u>	<u>Volts</u>
0.4	500	2000	40000	500,1000,1500,2140
1.2	500	1600	24000	500,1000,1500,2140
6.0	500	1200	12000	500,1000,1500,2140
12.0	500	1200	12000	500,1000,1500,2140
0.4	800	2000	30000	500,1000,1500,2140
1.2	800	1600	19200	500,1000,1500,2140
6.0	800	1200	12000	500,1000,1500,2140
12.0	800	1200	8000	500,1000,1500,2140
0.4	1100	1500	18000	500,1000,1500,2140
1.2	1100	1200	12000	500,1000,1500,2140
6.0	1100	800	6857	500,1000,1500,2140
12.0	1100	800	5333	500,1000,1500,2140

Figures 40 through 43 and tables 24 through 27 show the trends for generator weight and generator weight plus system

Figure 37. 1.2 MVA Generator Speed Study - Second Cycle

Figure 38. 6 MVA Generator Speed Study - Second Cycle

Figure 39. 12 MVA Generator Speed Study - Second Cycle

Figure 40. 0.4 MVA Generator Voltage Study - Second Cycle

Figure 41. 1.2 MVA Generator Voltage Study - Second Cycle

Figure 42. 6 MVA Generator Voltage Study - Second Cycle

Figure 43. 12 MVA Generator Voltage Study - Second Cycle

Table 21. Summary of 1.2 MVA Generator Designs, Speed Study, Second Cycle 500 Volts, L-N

Design No.	Average Coolant Temp. (°F)	RPM	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	Frequency (Hz)
20525	500	12000	1507	91.7	847	3357	1.32	H-11	28.5	17.9	124400	25802	1600
26	↓	16000	1364	92.0	889	3144	1.00	↓	27.5	17.8	123300	41862	↓
27	↓	19200	1148	90.7	977	3248	1.17	↓	26.3	16.8	122200	55453	↓
28	↓	24000	1008	92.3	1025	2708	1.20	↓	25.5	16.1	120300	81050	↓
29	↓	32000	947	92.8	1180	2547	1.13	↓	24.8	16.2	116400	133642	↓
20825	800	12000	1552	90.8	1227	2852	1.32	H-11	29.1	17.6	81600	26455	1600
26	↓	16000	1221	92.6	1192	2241	1.23	↓	27.4	16.2	81000	41920	↓
27	↓	19200	1151	92.1	1347	2251	1.14	↓	26.6	16.5	80200	55708	↓
28	↓	24000	1012	92.1	1392	2112	1.18	↓	25.7	15.8	79000	84483	↓
29	↓	32000	994	92.8	1482	1994	1.20	↓	25.0	16.9	76500	139710	↓
21125A	1100	18000	1543	89.4	1628	2793	1.10	1-B-S-1	29.0	17.2	57750	76298	1200
25	↓	14400	1664	92.6	1553	2514	1.22	↓	29.7	17.3	58600	50093	↓
26	↓	12000	1749	92.4	1538	2619	1.13	↓	30.2	17.2	59600	42859	↓
27	↓	10286	1880	93.2	1418	2660	1.18	↓	31.3	16.9	59800	28608	↓
28	↓	9000	2114	92.5	1376	2974	1.16	↓	32.5	17.3	59900	24048	↓
29	↓	8000	2093	91.5	1410	3043	1.33	↓	33.0	16.3	59900	19411	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 23. Summary of 12 MVA Generator Designs, Speed Study, Second Cycle, 1000 Volts, L-N

Design No.	Average Coolant Temp. (°F)	RPM	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	Frequency (Hz)
40525	500	6000	20229	92.9	1152	35529	1.46	H-11	63.4	41.2	125000	39232	1200
26	↓	8000	13554	93.1	1637	27054	1.03	↓	59.8	30.5	124900	59484	↓
27	↓	10286	11895	93.3	1727	24895	0.98	↓	56.9	30.2	124000	89426	↓
28	↓	12000	9769	93.0	2614	23469	1.08	↓	54.5	26.9	123300	109985	↓
29	↓	14400	8112	91.5	4302	24912	1.25	↓	51.5	25.0	122200	143803	↓
40825	800	4000	32792	90.4	1282	43392	1.01	H-11	77.3	41.2	82000	25974	1200
26	↓	6000	20195	92.6	1474	28395	1.44	↓	64.0	40.2	82000	40144	↓
27	↓	8000	14064	93.7	1692	20964	1.06	↓	60.3	31.5	81900	61386	↓
28	↓	9000	11482	91.8	2952	20482	1.27	↓	60.0	24.6	81600	73017	↓
29	↓	10286	12011	93.9	1982	18711	0.97	↓	57.5	29.6	81500	91690	↓
41125	1100	6000	25977	93.1	1734	32277	1.2	1-B-S-1	68.7	41.3	59900	56399	800
26	↓	5333	26654	92.9	1752	33154	1.32	↓	69.8	41.0	59900	45112	↓
27	↓	4800	31104	92.5	1904	38004	1.22	↓	73.1	43.0	60000	40588	↓
28	↓	4000	32339	92.4	1614	39289	1.38	↓	75.5	41.1	60000	30105	↓
29	↓	3200	43802	91.6	1534	51505	1.17	↓	82.4	45.4	60000	23314	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

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Design No.	Average Coolant Temp. (°F)	Frequency (Hz)	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM	Volts L-N
10530	500	2000	294	91.3	1007	984	1.09	H-11	17.3	11.5	116400	105131	40000	500
31	↓	↓	301	90.9	1112	1031	1.14	↓	17.5	11.7	↓	99834	↓	1000
32	↓	↓	336	88.9	1331	1246	1.02	↓	18.4	11.8	↓	100790	↓	1500
33	↓	↓	380	87.2	1347	1460	0.99	↓	19.3	12.4	↓	104745	↓	2140
10830	800	2000	329	91.2	1200	769	1.09	H-11	18.2	11.4	79000	64236	30000	500
31	↓	↓	363	91.5	1158	783	0.93	↓	18.8	11.5	↓	65456	↓	1000
32	↓	↓	345	90.7	1260	810	1.08	↓	18.7	11.4	↓	62119	↓	1500
33	↓	↓	433	88.1	1454	1053	1.02	↓	20.4	12.5	↓	65560	↓	2140
11130	1100	1500	536	91.8	1400	871	0.93	1-B-S-1	21.5	11.9	58600	38378	18000	500
31	↓	↓	555	91.6	1412	900	0.99	↓	21.8	12.0	↓	37903	↓	1000
32	↓	↓	554	90.1	1425	969	1.01	↓	21.9	11.2	↓	38554	↓	1500
33	↓	↓	584	89.5	1545	1024	1.10	↓	22.4		↓	38163	↓	2140

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 25. Summary of 1.2 MVA Generator Designs, Voltage Study, Second Cycle

Design No.	Average Coolant Temp. (°F)	Volts L-N	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM	Frequency (Hz)
20530	500	500	1008	92.3	1031	2708	1.20	H-11	25.4	16.0	120300	81050	24000	1600
31	↓	1000	987	92.9	1020	2587	1.18	↓	25.4	15.4	↓	78027	↓	↓
32	↓	1500	1037	92.6	1049	2657	1.28	↓	25.8	15.5	↓	77295	↓	↓
33	↓	2140	1137	92.3	1049	2837	1.16	↓	26.6	16.0	↓	80790	↓	↓
20830	800	500	1151	92.1	1350	2251	1.14	H-11	26.6	16.5	80200	55708	19200	1600
31	↓	1000	1166	93.0	1204	2136	1.26	↓	26.9	16.2	↓	56969	↓	↓
32	↓	1500	1253	93.1	1255	2203	1.08	↓	27.6	16.2	↓	57012	↓	↓
33	↓	2140	1283	92.5	1329	2323	1.24	↓	28.0	16.2	↓	56117	↓	↓
21130	1200	500	1749	92.4	1538	2629	1.13	1-B-S-1	30.2	17.2	59600	35763	12000	1200
31	↓	1000	1813	92.9	1532	2623	1.18	↓	30.9	17.1	↓	35692	↓	↓
32	↓	1500	1909	92.9	1535	2719	1.22	↓	32.0	16.7	↓	36443	↓	↓
33	↓	2140	2048	92.6	1574	2898	1.20	↓	32.9	16.9	↓	37627	↓	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 26. Summary of 6 MVA Generator Designs, Voltage Study, Second Cycle

Design No.	Average Coolant Temp. (°F)	Volts L-N	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM	Frequency (Hz)
30530	500	500	5257	91.9	1568	13257	1.15	H-11	44.8	22.3	123300	72754	12000	1200
31	↓	1000	6712	92.8	1249	13912	1.27	↓	45.5	28.9	↓	76910	↓	↓
32	↓	1500	7237	92.6	--	14637	1.10	↓	46.7	--	↓	76323	↓	↓
33	↓	2140	7241	92.2	1428	14941	1.23	↓	47.4	28.4	↓	76042	↓	↓
30832	800	500	5317	92.4	1868	9717	1.14	H-11	45.2	22.0	81000	74571	12000	1200
33	↓	1000	6733	93.4	1427	10533	1.24	↓	46.0	28.3	↓	77890	↓	↓
34	↓	1500	7383	93.1	--	11333	1.10	↓	47.4	--	↓	78770	↓	↓
35	↓	2140	7395	92.9	1596	11495	1.22	↓	48.4	28.0	↓	77995	↓	↓
31130	1100	500	10644	92.8	1769	13994	1.38	1-B-S-1	53.5	29.6	59800	34596	6857	800
31	↓	1000	11086	92.8	1689	14436	1.45	↓	54.2	30.1	↓	42571	↓	↓
32	↓	1500	11765	92.5	1652	15265	1.01	↓	56.7	28.1	↓	45603	↓	↓
33	↓	2140	12123	91.8	1941	15973	1.31	↓	58.8	28.1	↓	45975	↓	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 27. Summary of 12 MVA Generator Designs, Voltage Study, Second Cycle

Design No.	Average Coolant Temp. (°F)	Volts L-N	Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Rotor Material	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)	RPM	Frequency (Hz)
40530	500	500	9966	93.0	1988	23766	1.08	H-11	54.8	27.4	123300	111309	12000	1200
31	↓	1000	10704	92.8	1886	24704	1.17	↓	55.4	28.9	↓	115554	↓	↓
32	↓	1500	12135	93.2	1630	25335	1.44	↓	54.5	35.0	↓	114484	↓	↓
33	↓	2140	13600	92.5	1808	28600	1.18	↓	56.7	35.4	↓	117500	↓	↓
40830	800	500	16619	93.5	1371	23719	1.21	H-11	60.7	37.4	81900	63901	8000	1200
31	↓	1000	16037	92.9	1463	23837	1.37	↓	60.3	36.6	↓	63427	↓	↓
32	↓	1500	17223	93.6	1409	24223	1.20	↓	61.0	38.4	↓	63588	↓	↓
33	↓	2140	79364	92.1	1658	28064	1.15	↓	63.5	39.8	↓	65130	↓	↓
41130	1100	500	24906	93.2	1671	31106	1.13	1-B-S-1	70.2	37.6	59900	46531	5833	800
31	↓	1000	26654	92.9	1752	33154	1.32	↓	69.8	41.0	↓	45112	↓	↓
32	↓	1500	27471	92.9	1700	33971	1.35	↓	71.3	40.3	↓	46266	↓	↓
33	↓	2140	30411	92.5	1705	37211	1.13	↓	73.9	41.2	↓	49122	↓	↓

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

weight penalty as a function of generator line-to-neutral terminal voltage. For all ratings except the 1.2 MVA rating, generator weight plus system weight penalty increased with voltage for the range of 500 volts to 2140 volts. For the 1.2 MVA rating, system weight was minimum at 1000 volts with a slight increase in system weight at 500 volts. In general, system weight was not a strong function of voltage in the range of 500 volts to 1000 volts.

The establishment of a safe working voltage for inorganically-insulated magnet wire requires consideration of several important factors. These are insulation composition, material form, and operating temperature. As a result of testing and evaluation performed on NAS 3-4162 and NAS 3-6465, magnet wire insulation based upon served glass fiber with additional inorganic fillers and glassy binders has been selected as best for operation at temperatures of 900° to 1300°F. This insulation is known as Anadur and is produced by Anaconda Wire and Cable Company. The fully processed form of this insulation does not achieve complete density, but is porous. A designer must consider that the insulation serves as spacing rather than a complete insulating barrier. The glass-based binders in this composition begin to lose electrical insulation resistance at 1000°F and are very poor insulators above 1300°F. Thus, the electrical stresses of voltage potential, must be kept low if high current losses are to be avoided. The total electric strength of Anadur, 0.0068-inch thick, at 1100°F, is approximately 400 volts. This results in a typical value of about 60 volts per mil. This form of insulation should be used with supplemental ground and phase insulation composed of high-purity aluminum oxide. For efficient utilization of available conductor and insulation volume in a magnetic core, the ceramic supplemental insulation will be of minimum thickness, consistent with production and winding capabilities. The electric strength of aluminum oxide at 1100°F varies from about 60 to 150 volts per mil. For high purity alumina, about 0.018-inch thick, typical electric strength is about 100 volts per mil at 1100°F.

In order to assure a margin of safety to compensate for insulation defects created in processing, mechanical shock in operation, and transient overvoltages, an operating limit of 500 volts at temperature is recommended. Electrical current leakage with accompanying excessive losses may result if higher operational voltage level is attempted at conductor temperatures above 1000°F. In general, as voltage is increased, problems associated with electrical arcing and corona become more pronounced. Based on results of the voltage parametric study and the above considerations regarding voltage selection, a voltage of 500 volts line-to-neutral was chosen for the 0.4, 1.2, 6.0, and 12.0 MVA generator ratings.

6. GENERATOR DESIGNS WITH ADVANCED HIGH TEMPERATURE MATERIALS

Two generator designs which are representative of optimum designs were analyzed using advanced materials to determine possible weight and efficiency benefits. One design, representative of the optimum designs was analyzed using a 1400°F coolant temperature and advanced materials to determine possible weight and efficiency benefits.

The two designs selected for study with advanced materials were the 0.4 MVA, 30,000 rpm, 2000 Hz design and the 1.2 MVA, 19,200 rpm, 1600 Hz design, both designs having coolant temperatures of 800°F and H-11 rotor material. The 1100°F coolant temperature was not selected for this investigation because it was necessary to choose initially (or later change to) advanced materials in order to successfully complete the parametric study for the 1100°F coolant temperature. Therefore, the 1100°F designs are already representative of advanced high-temperature material designs.

The experimental alloy, 1-A-S-2* was selected to replace the H-11 rotor material to investigate the possibility of reduced system weight, by means of increasing the rotor speed. Material creep strength of the 1-A-S-2 rotor alloy is about 34 percent greater at the 800°F coolant temperature than the H-11 alloy and has a limiting rotor hot-spot temperature (occurring at the rotor pole faces) of about 1050°F compared to 950°F for H-11. The magnetic properties of 1-A-S-2 and H-11 alloys are shown in figure 44 for comparison.

Generator designs with the 1-A-S-2 rotor material are compared against the corresponding generator designs with H-11 rotor material in table 28. Generator weight plus system weight penalty for the 0.4 MVA, 30,000 rpm, 1-A-S-2 rotor design is slightly higher (5%) than for the corresponding H-11 rotor design but has the advantage of a higher stress safety factor than the H-11 rotor design at 30,000 rpm. At 40,000 rpm, the 1-A-S-2 rotor is slightly overstressed. The same trend is noted for the 1.2 MVA generator except that a speed increase from 19,200 rpm

*Martensitic precipitation-hardened iron base alloy (alloy 1-A-S-2) being developed by Westinghouse Electric Corp. on Contract NAS 3-6465 for the NASA, Lewis Research Center; composition is 53.2Fe-12.0Ni-30.0Co-1.0W-0.4Al-0.4Ti

to 24,000 rpm did not cause the 1-A-S-2 rotor to be overstressed. The speed increase resulted in a small (6.5%) system weight savings. The 1-A-S-2 rotor material does look attractive from the standpoint of increased material strength for a relatively small sacrifice in weight.

The design reference selected for the 1400°F coolant temperature analysis was the 1.2 MVA, 14,400 rpm, 1100°F coolant temperature design. The experimental alloy CBDS* was selected to replace the 1-B-S-1 rotor alloy which is also an experimental alloy but is not capable of operating at a coolant temperature of 1400°F. See figure 45 for estimated magnetic properties used for CBDS alloy. The HIPERCO 27 stator frame and armature stack material and the alumina-insulated dispersion-strengthened copper conductors of the 1100°F design were also selected for the 1400°F coolant temperature design, because these materials are representative of the most advanced materials at this point in time. Table 29 shows the generator design for the 1400°F coolant temperature compared to the design representative of the optimum 1100°F coolant temperature generator. Material strength of the CBDS alloy at the 1400°F coolant temperature is approximately 17 percent of the 1-B-S-1 alloy at the 1100°F coolant temperature. A reduction in speed (from 12,000 rpm to 4000 rpm) was necessary to reduce the rotor stress below the material strength limit of the CBDS rotor alloy. This resulted in a large increase in generator weight and system weight compared to the 1100°F coolant temperature design. Results of this analysis show that a feasible design can be arrived at for a 1400°F coolant temperature with the experimental rotor alloy CBDS, but there is no advantage in doing so with regard to system weight.

*Cobalt-base dispersion-strengthened alloy (CBDS). Material data used in 1400°F parametric study was based on preliminary estimates. More up-to-date information is available in report NASA-CR-54361 entitled "High Temperature Magnetic Materials Topical Report" on contract NAS 3-6465.

Figure 44. D.C. Magnetization Curves

Figure 45. Estimated D.C. Magnetization Curve for CBDS Alloy at 1500°F

Table 28. Generator Design with Advanced High Temperature Materials

Rating (MVA)	RPM	Rotor Material	Generator Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	X _d (P.U.)	Frequency (Hz)	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)
0.4	30000	H-11	349	91.7	1054	759	0.89	2000	18.6	11.4	79000	67000
0.4	30000	1-A-S-2	356	91.2	1150	796	0.89	2000	18.7	11.3	106000	69824
0.4	40000	1-A-S-2	318	91.6	1287	738	1.03	2000	17.8	11.9	102700	109471
1.2	19200	H-11	1151	92.1	1346	2250	1.14	1600	26.6	16.5	80200	55708
1.2	19200	1-A-S-2	1163	92.0	1351	2273	1.13	1600	26.8	16.2	107500	57792
1.2	24000	1-A-S-2	1020	92.2	1271	2110	1.16	1600	25.8	15.6	106000	84868

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.

Table 29. Generator Design for 1400°F Coolant Temperature

Rating (MVA)	Coolant Temp. (°F)	RPM	Generator Electrical Weight (lbs)	% Efficiency	Average Winding Temp. (°F)	Electrical Weight & System Weight Penalty (lbs)	Rotor Material	Frequency (Hz)	Maximum O.D. (in.)	Electrical Length (in.)	Maximum Allowable Rotor Stress* (psi)	Average Tangential Stress (psi)
1.2	1400	4000	6631	91.7	1634	7571	CBDS	1200	41.3	29.3	10000	8610
1.2**	1100	12000	1749	92.4	1539	2619	1-B-S-1	1200	30.2	17.2	59600	42859

* Maximum Material Stress Corrected to Account for Stress Concentrations Created by Poles.
 ** Reference design.

7. SUMMARY AND CONCLUSIONS

A summary of the parametric combinations and corresponding generator electrical weight*, generator efficiency, and generator electrical weight plus system weight penalty resulting from the second cycle parametric studies is given in table 30. The best rotor material for coolant temperature of 500° and 800°F was found to be H-11 (AMS 6487) alloy. The use of NIVCO alloy at a coolant temperature of 800°F resulted in relatively heavy, low-speed designs.

At a coolant temperature of 1100°F, rotors of NIVCO alloy were overstressed at all but unreasonably low speeds. Feasible generator designs were obtained with the experimental rotor alloy 1-B-S-1 but the increased generator weight of these designs more than offset the reduced system weight penalty associated with the 1100°F coolant temperature.

Table 30. Summary of Parametric Combinations

Generator Rating (MVA)	Coolant Temp. (°F)	Frequency (Hz)	Speed (rpm)	Volts	Generator Electrical Weight (lbs)	Generator Efficiency (lbs)	Generator Weight Plus System Weight Penalty (lbs)
0.4	500	2000	40000	500	294	91.3	984
1.2	500	1600	24000	500	1008	92.3	2708
6.0	500	1200	12000	500	5257	91.9	13257
12.0	500	1200	12000	500	9966	93.0	23766
0.4	800	2000	30000	500	329	91.2	769
1.2	800	1600	19200	500	1151	92.1	2251
6.0	800	1200	12000	500	5317	92.4	9717
12.0	800	1200	8000	500	16619	93.5	23719
0.4	1100	1500	18000	500	536	91.8	871
1.2	1100	1200	12000	500	1749	92.4	2629
6.0	1100	800	6857	500	10644	92.8	13994
12.0	1100	800	5333	500	24906	93.2	31106

*Excludes weight of stub shafts, end brackets, shrouds, bearings, seals, terminals, etc., includes armature stacks, magnetic stator frame, rotor, armature and field copper weights.

Figure 46 shows the approximate maximum generator speed, based on rotor stress considerations, as a function of MVA rating for coolant temperatures of 500°, 800°, and 1100°F, respectively, as determined from the second cycle parametric study. The speeds selected for the generator designs, representative of the optimum designs, are shown for each MVA rating. Increasing the speed to the maximum allowable speed results in a relatively small increase in system weight savings, or in some cases, a system weight increase, at the expense of a large reduction in rotor stress safety factor.

Figure 47 shows generator electrical weight plus system weight penalty plotted as a function of coolant temperature for ratings of 0.4, 1.2, 6.0, and 12.0 MVA respectively. For all ratings, minimum system weight is achieved at the 800°F coolant temperature. Assuming, for the time, that the curve lines between points hold true (i.e., discontinuity would not occur if studies were performed at intermediate coolant temperatures) a range of coolant temperatures can be established which would result in optimum or near optimum system weights. The selected coolant temperature ranges based on the indicated trends would be 650° to 950°F for ratings of 0.4, 1.2, and 6.0 MVA and 500° to 800°F for the 12.0 MVA rating. In reality, a discontinuity does exist in the curves of figure 47, because of the approximate 800°F coolant temperature limit of H-11 alloy and the required change in rotor material for the 1100°F coolant temperature. Typically, the shape of the curves would be as shown by the solid line of figure 48 when the discontinuity is considered. The previously described optimum coolant temperature ranges, modified to recognize the temperature limit of the H-11 rotor alloy, are given in table 31.

Figure 46. Maximum Allowable Generator Speed

Figure 47. Effects of Coolant Temperature on System Weight

Figure 48. Discontinuity Curve

Table 31. Optimum Coolant Temperature Range

Rating (MVA)	Optimum Coolant Temperature Range (°F)	Indicated Optimum Coolant Temperature (°F)
0.4	650-800	800
1.2	650-800	800
6.0	650-800	800
12.0	500-800	650

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8. REFERENCES

1. Work on the SNAP-50/SPUR Generator covered a period from 1960 to 1966 and was carried out under U.S.A.F. Contracts AF33(616)-7379 and -8322, AF33(657)-8954 and -10922, and AF33(615)-1551 and -2326. Progress reports covering AF33(657)-8954, AF33(615)-1551, and AF33(615)-2326 were utilized extensively in selecting design parameters and background for this study.
2. Work on Contract NAS 5-1234 was reported in six volumes:
Vol. I, II, III and IV as Westinghouse Electric Corp., Aerospace Electrical Division, Lima, Ohio, Report No. D-709533, December 1962.

One volume as Westinghouse Electric Corp., Aerospace Electrical Division, Lima, Ohio, Report No. WAED 63.11E, December 1963.

One volume, unnumbered, by Westinghouse Electric Corp., Aerospace Electrical Division, Lima, Ohio, on 250 and 500 kW designs on an extension of NAS 5-1234.

Official NASA contractor report numbers were not applied to these reports.
3. Magnetic Materials Topical Report, NASA Lewis Research Center Report, NASA-CR-54091, September, 1964. Prepared by Westinghouse Electric Corp., Aerospace Electrical Division, Lima, Ohio.
4. Electrical Conductor and Electrical Insulation Materials Topical Report, NASA Lewis Research Center Report, NASA-CR-54092, October, 1964. Prepared by Westinghouse Electric Corp., Aerospace Electrical Division, Lima, Ohio.
5. "Contract for Development and Evaluating Magnetic and Electrical Materials Capable of Operating in the Temperature Range from 800° to 1600°F". Contract No. NAS 3-6465.
6. R. W. Briggs and F. X. Dobler, "Development of Potassium-Cooled Generators for Space Application". Presented at AIAA Summer Meeting, June 17-20, 1963.

7. Research on Spacecraft and Powerplant Integration Problems, Final Report, NASA Lewis Research Center Report, NASA-CR-54159, July, 1964. Prepared by General Electric Co.
8. Advanced Nuclear Electric Power Generator System Study, Rankine Cycle Nuclear Spare Powerplant, Pratt and Whitney Aircraft Co., Report No. PWA-2232, August, 1963.
9. General Electric Missile and Space Division, Research on Spacecraft and Powerplant Integration Problems, Third and Fourth Quarterly Report, Report 64SD700, Philadelphia, Pa., 26 April 1964. Contract NAS 3-2533.
10. Design Study Electrical Component Technology for 0.25 to 10 Megawatt Space Power Systems, Final Report, AEC Report SAN-679-8, February, 1968. Prepared by Westinghouse Electric Corp., Aerospace Electrical Division, Lima, Ohio.
11. Robinson, R.C., I. Rowe, and L. E. Donelan, "Calculations of Can Losses in Canned Motors," AIEE Transactions, Part III, Power Apparatus and Systems, pp. 312-315, June 1957.
12. "Screw Type Shaft Seals for Potassium Lubricated Generators" by A. E. King, Supplement to IEEE Transactions on Aerospace, Vol. AS-3, No. 2, June, 1965.
13. SNAP-50/SPUR 467 kVA Experimental Generator Design Summary, USAF Report AFAPL-TR-67-7, May 1967. Prepared by Westinghouse Electric Corp., Aerospace Electrical Division, Lima, Ohio.
14. Analysis of the Viscoseal, Part II, The Concentric Turbulent Case, NASA Report No. NASA-CR-76866, U. of Tenn., Dept. of Mechanical & Aerospace Engineering, June, 1966.
15. Centrifugal Types of Dynamic Shaft Seals, ASME Paper No. 63-WA-176, November, 1963.

APPENDIX A

ROTOR-BEARING PARAMETRIC SCREENING STUDY

Documentation of the rotor-bearing parametric screening studies by Mechanical Technology Inc., was originally completed under separate cover in an MTI report, MTI-67TR86. This report is included in its entirety in the following pages (Appendix A).

All references, figures, tables, etc. referred to in the report (Appendix A) are those associated with Appendix A and not those in the topical report of which Appendix A is a part.

$$\begin{array}{r} 114 \\ 150 \\ \hline 268 \end{array}$$

NO. MTI-67TR86

DATE: Oct. 20, 1967

TECHNICAL REPORT

Rotor-Bearing Parametric Screening Study
for Liquid Potassium Lubricated Generators

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REFERENCES

ABSTRACT

A rotor-bearing parametric study was made of liquid potassium lubricated generator rotors in the range of 0.25 to 10 megawatt space nuclear power systems. The following rotors were considered in this study.

Westinghouse Corporation	Electrical Rating (KW)	Design Speed (RPM)	Lubricant Temperature °F
10526	350	30,000	500
10826	350	30,000	800
10827	350	24,000	800
20528	1000	24,000	500
20827	1000	19,200	800
20826	1000	16,000	800
30829	5000	12,000	800
40829	10000	8,000	800

The journal bearings, bearing supports and rotor stub shafts were sized, and the steady state and dynamic bearing performance characteristics were calculated for each of these eight rotors. Critical speed and rotor-bearings dynamic analyses were made for three representative generator designs (Design Nos 10827, 20827, and 30829).

It was concluded that acceptable operation with good bearing performance and small unbalance response orbits can be realized. There are no flexural system critical speeds anywhere near the operating range of any of the generator rotors considered in the study. Two areas where new technology is needed to facilitate successful development of the bearings for high speed, liquid metal lubricated turbomachinery were identified as being: (a) development and quality control of bearing and journal surface coatings for good compatibility in liquid metal environment and, (b) extension of present turbulent lubrication theory to include fluid inertia effects. This latter is important principally for the larger machines with power rating exceeding 1000 KW.

FOREWORD - DEFINITION OF THE ROTOR-BEARINGS PARAMETRIC STUDY

The rotor-bearing parametric screening study of generator rotors for 0.25 to 10 megawatt Space Nuclear Power Systems reported herein was conducted by Mechanical Technology Incorporated for Westinghouse Corporation, Aerospace Division.

The purpose of this study, as defined in the Westinghouse Corporation Purchase Order, was to analyze the liquid metal lubricated journal bearing requirements that coincide with the optimum inductor alternator generator designs determined by Westinghouse Corporation under a separate task of their AEC Contract AT(04-3)-679. The rotor-bearings analysis was required to include (a) the selection of journal bearing type, (b) the specification of bearing configuration and dimensions and, (c) the determination of the rotor-bearing dynamics and stability characteristics of the selected configurations.

The specific objectives and results of the study were limited to the selection, specifications and tabulation of analytical parametric data, facts, designs, etc. and the statement of conclusions or recommendations, related to the analysis.

The intent of the results reported herein was that the data be of value to both the specification of generator design requirements and the specification of liquid metal lubricated journal bearing technology requirements to achieve operating bearings for large-rating turbo-alternators in Rankine Cycle Space power systems.

For the purpose of conducting this study, Westinghouse Corporation furnished MTI with:

- a. Rotor configurations including complete dimensional details, pole configuration, weight, materials, materials properties at operating temperatures, shaft seal stub shaft requirements, etc.
- b. Lubricant, operating temperatures and lubricant properties of interest.
 - (1) Lubricant shall be liquid potassium
 - (2) Specified bearing operating temperature for each rotor
 - (3) Potassium properties.

- c. Operating conditions shall include bearing reactions for gravity and zero-gravity at either no-load demagnetized, no-load magnetized or full-load magnetized electrical state, which ever is most pertinent to the rotor-bearing selection and analysis.
- d. Generalized pedestal configuration limitations.
- e. Proposed balance specification.

Specifically the work items to be accomplished in the study were as follows:

- a. Select best journal bearing configuration for each rotor-bearing system, e.g., pivoted-pad, hydrostatic, etc. Preference shall be given to 4- or 6-pad pivoted-pad bearings.
- b. Determine pertinent bearing dimensions for the selected bearings, e.g., length, diameter, clearance, pivot locations, preload, (consider the case of zero preload where applicable).
- c. Determine rotor-bearing system critical speeds (prefer generalized solution of rotor speed vs bearing stiffness on log-log scale).
- d. Determine dynamic response characteristics of each rotor-bearing system (e.g., amplitude vs speed, damping and spring coefficients). If a pivoted pad bearing is not selected, the stability of the bearing selected shall be substantiated in a suitable manner.
- e. Determine maximum allowable unbalance on a rigid foundation and make recommendations relative to proposed balance specification. Determine dynamic bearing loads (reactions) and maximum rotor deflections under maximum unbalance conditions.
- f. Determine bearing operating characteristics such as minimum film thickness, eccentricity ratio, bearing losses.
- g. Determine variable speed and/or start-up requirements relative to:
 - (1) minimum film thickness
 - (2) minimum operating speed for sustained operation at no-load magnetized and demagnetized conditions only (at gravity and zero-gravity condition).

The rotor and resulting bearing reactions were to be assumed to be symmetrical about the bearings. The bearings and pedestals were to be assumed to be massless with no damping in the pedestals.* The radial clearance were to be selected according to the requirements at the operating condition, assuming that thermal compensation would be provided to maintain the clearance at room temperature.

* The masses of the bearing pads and of the bearing housings were actually incorporated in the analysis, as this could be achieved with little difficulty and it improved the validity and accuracy of the dynamic model of the bearing supports.

SUMMARY AND CONCLUSIONS

A dynamic analysis has been performed to determine practical bearing, bearing support and rotor stub shaft dimensions for a range of sizes of radial gap generators, lubricated with liquid potassium. The designs were based upon information contained in the Westinghouse Corporation drawings EDSK-332561, which provided the rotor and pedestal configuration data for each of the generators. The generator body dimensions, rotor weight, design speeds, operating temperatures and other information required for the bearing design and rotor-bearing dynamic analysis were contained in these drawings.

The design analysis reported herein included for each generator:

- (a) selection and sizing the journal bearings, bearing supports and rotor stub shafts
- (b) calculation of the steady-state and dynamic performance characteristics of the journal bearings and supports including: eccentricity ratio, minimum pivot point film thickness, power loss, fluid film stiffness and damping as well as the stiffness of the bearing support.
- (c) preparation of the system undamped critical speed maps
- (d) detailed unbalance response calculations, which provided the rotor orbits and transmitted dynamic loads, for the specified maximum values of mass unbalance.

The bearing and the rotor-bearing dynamics analyses were conducted for three load conditions as follows: (a) the zero load condition corresponding to vertical operation or operation in a zero g field in the absence of magnetic loading, (b) the normal static load condition corresponding to horizontal operation in a one g field, with no or little magnetic loading and, (c) the maximum static load condition in which the journal bearings support the rotor weight as well as the maximum calculated magnetic loading. Conditions (a) and (c) are the limiting load conditions on the bearings, i.e. under condition (a) the net bearing load is zero and under condition (c) the bearing load is at its maximum value. The calculations given in the report showed that the maximum magnetic load is, in all cases, smaller than the gravity load due to the rotor weight. Thus, the fourth operating conditions of interest, i.e. where the rotor operates vertically (or in a zero g field) but with the maximum magnetic load, is intermediate between conditions (a) and (b) above, from the standpoints of bearing operation and rotor-bearing dynamics.

The analyses were made for a speed range that spanned the rigid body system critical speeds, the design speed and an overspeed range.

The bearing design and steady-state bearing performance are described in Section II of this report. The numerical results are given in Tables I through V. The principal conclusions of this section may be summarized as follows:

1. Self-acting bearings are capable of supporting the maximum loads at design speed.
2. Tilting pad bearings are preferred for reasons of inherent stability, inherent self alignment, superior (to other stable, self-acting bearing types) stiffness and damping, as well as the background of successful operation of this type of bearing in liquid metal ambients.
3. Four-pad tilting pad bearings are preferred over six-pad bearings in all cases, as they provide somewhat higher load capacity, stiffness and damping, at least in the range of L/D 's from 0.5 to 1, which was used for all the generators in this study.
4. In order to insure stability even under zero net bearing load conditions with low viscosity lubricants, preloading of the bearings is desirable. A preload factor of 0.3 was used in all cases.
5. A conservative bearing design was achieved insuring that the eccentricity ratio at design speed and under the maximum static load conditions does not exceed 0.5. This provides a good margin against off design conditions, including operation under moderately large acceleration loads, of the order of 2 g.
6. The bearing power loss was calculated and shown to be acceptably small, being in all cases smaller than or about 1% of the rated generator output, for the two bearings. This power loss, as well as all other steady-state and dynamic characteristics were obtained using turbulent flow bearing theory, as the bearings in all cases operate well above the point of transition from laminar flow.
7. The pivots of the pivoted pad bearings were designed and sized. The pivots were sized, such that the radius of curvature of the socket element is larger than that of the ball seat to insure that pivot motions are taken

by elastic deformation and rolling motions rather than by sliding motions, to prevent pivot damage in long term operation. The pivots were sized for low Hertzian contact stresses but high radial pivot stiffness.

8. The disk diaphragms that support the bearing housing were also sized for high radial stiffness and to insure that the lowest disk resonance is at least 35% higher than design speed. This can be easily raised if desired by increasing the disk thickness or by providing support ribs.
9. The rotor stub shaft dimensions were selected to satisfy the journal bearing sizes, to allow for axial length for bearing seals and to satisfy good rotor-dynamic performance.
10. The fluid film stiffness and damping coefficients required for the rotor-bearing dynamics analyses were calculated.

The rotor-bearing dynamics analysis is given in Section III of this report. The principal results are given in Figs. III through VI. The conclusions of the analysis may be summarized as follows:

1. The undamped system critical speed maps were obtained for each rotor. These indicated that the rotors will operate near but generally above the second, rigid body system critical speed.
2. There are no flexural critical speeds anywhere near the operating region. The lowest flexural critical speeds are, generally, above twice the design speed.
3. The unbalance response calculations that were made over the full speed range through the rigid body critical speeds, the design speed and the overspeed range, showed that acceptable operation with low vibration amplitudes can be realized in every case. At design speed, the dynamic rotor orbits under the highest specified values of mass unbalance are of the order of 5% of the bearing clearance.
4. The unbalance response was obtained both for:
 - (a) the residual mass unbalances in the two dynamic balance correction planes located in-phase with each other and,
 - (b) the mass unbalances located 180° out of phase with each other.

Since the rotors operate nearer to the second than to the first rigid body critical speed, and since this second critical speed was a conical mode, the largest dynamic orbits and transmitted force were obtained with the "out-of-phase" locations of the mass unbalance. The dynamic orbit sizes noted in Item 3 above are for this most severe distribution of the residual unbalance.

5. The unbalance response calculations also showed that the amplitude peaks of the rigid body resonances are low and well damped. The rotor orbits at these resonances were also of the order of 5% of the bearing clearance or less.
6. The maximum specified mass unbalances (given in the Westinghouse Corporation drawings EDSK 332561) were found to be within the limits achievable by "High Quality" and "Good Quality" balancing of rigid rotors. Accordingly, it was recommended that the specified maximum unbalances be adhered to. In view of the moderate dynamic orbit sizes and transmitted loads at these speeds, they allow some margin for possible deterioration of the residual unbalance in long term operation.

General purpose design data on 4 pad and 6 pad tilting pad bearings operating in laminar and turbulent flow regimes is given in this report, in Appendix C. This data was used in the design of the generator bearings.

Some areas where new technology is needed in connection with the development of the bearings and rotor-bearing systems for high speed, liquid metal lubricated turbomachinery for space power and other applications are identified in Section IV of this report. These include:

1. Development and quality control of ceramic and other coatings is desirable to achieve journal bearing surfaces that have good compatibility with each other in high temperature, liquid metal environment. The achievement of compatible bearing and journal surfaces by means of coating is particularly promising, as it then allows the journals and bearing pads to be constructed of normal, structural materials, thus avoiding the problem of differential thermal expansions that exist if solid carbide pads and sleeves are used.

2. With normal bearing clearance ratios* in the range of 0.001 "/" to 0.002 "/", the present turbulent lubrication theory is valid at Reynolds Number up to about 20,000. Above this, it is expected that fluid inertia effects become increasingly important. For the generators considered in this study, the present turbulent lubrication theory is entirely adequate for the smaller generators, up to the ones rated at 1000 KW. For the larger units, however, the bearing Reynolds Numbers are higher, in the range of 30,000 to 60,000, so that the calculation of the bearing characteristics, neglecting fluid inertia effects are approximate. Extension of the turbulent lubrication theory to include fluid inertia effects and its experimental verification is a technology area that is needed for development of large, liquid metal lubricated turbomachinery.

* The "bearing clearance ratio" is the ratio of the radial clearance in the bearing to the radius of the bearing.

I. INTRODUCTION

The successful development of high-speed, high-temperature rotating machinery frequently depends as much on bearing and seal design and on proper integration of the system from a rotor-dynamics standpoint, as it does on optimum design of its electrical or aerodynamic components. Frequently also, the mechanical design of the machinery can be simplified and its reliability improved by using the process fluid as the lubricant. This permits such advantages as (a) reduction of thermal gradients along the rotor by operating the bearings at system temperature, (b) elimination of complex oil seals, (c) prevention of oil contamination of critical system components and, (d) elimination of the cumbersome and sometimes unreliable auxiliary lube oil system.

It must be recognized, however, when using process fluids such as liquid metals as lubricants, that they have low viscosity and lack the boundary lubrication property of conventional, hydrocarbon lube oils. This makes bearing selection and rotor-bearing dynamics design particularly critical. The stability threshold of the bearings must be well above the operating speed range and the bearing stiffness and damping must be accurately computed to permit analysis of the dynamic response of the rotor-bearing system. Conservative sizing of the bearings, to provide margin against off-design operating conditions is generally desirable to compensate in part against the absence of a boundary lubrication safeguard.

Continuous maintenance of the thin film between the journal and pads in a fluid film bearing requires careful investigation of the amplitudes of motion expected in the bearing system. The presence of unbalance may cause large motions as the rotor passes through the system critical speeds. Suppression of these motions is possible first through fine balancing, but mainly by effective action of system damping, which is almost entirely provided by the bearings. Failure to design the rotor-bearing system, so that excessive motions between the journal and the bearing are effectively prevented, will lead at best to annoying vibratory problems and at worst to failure of the machine.

The main objective of a rotor response analysis is the prediction of synchronous vibration amplitude at the design speed, where the rotor is intended to operate

for long life. Other objectives include more precise location of the system critical speeds and determination of maximum expected amplitudes of vibration as the rotor passes through these speeds.

The bearing designs and the rotor-bearing dynamics analyses conducted here for the generator rotors were made based on the above considerations.

The study was conducted in several steps as follows.

1. Accurate design data in laminar and turbulent flow regimes was obtained to cover the range of operating conditions of the various rotors. This data was provided in normalized form so that it constitutes technology information applicable to the rotors under consideration in the study, as well as other applications. This data is given in Appendix C.
2. Accurate dynamic models of the rotors and pedestals were generated from the Westinghouse Corporation rotor and support drawings for the dynamic analysis.
3. A preliminary investigation was made for each rotor bearing system, in which the bearings were sized, and the bearing and support stiffness and damping characteristics were calculated, and critical speed maps and rotor response analyses were made.
4. Based on these preliminary investigations, optimization studies were conducted for each rotor, varying the factors allowed in the rotor-bearing design study, such as bearing dimensions and span, to arrive at the final designs in which bearing load capacity and power loss were satisfactory and the dynamic amplitudes of the rotor under specified unbalance loads were everywhere acceptable.
5. Final bearing designs, rotor-bearing critical speed maps and rotor response were then prepared for each rotor.

II. BEARING DESIGN AND STEADY-STATE PERFORMANCE

1 Selection of Bearing Type

The two basic types of fluid film bearings are:

- (a) the externally pressurized (hydrostatic) bearing, where the load capacity is generated principally by introducing high pressure lubricant into several pockets or recesses in the bearing, through orifices or other restrictors. The load capacity of this type of bearing is, therefore, principally dependent on the available pressure difference between the high pressure lubricant source and the ambient pressure.
- (b) the self-acting (hydrodynamic) bearing where pressures are self-generated in the fluid film between the rotor and stator by shear of the thin, viscous, lubricant film. These self generated pressures serve to separate the rotor and stator elements. The load capacity of this type of bearing is, therefore, principally a function of the speed and lubricant viscosity. The lubricant supply to the bearing need not be pressurized, except to the extent required to overcome the losses in circulating the flow to lubricate and cool the bearings.

Calculation of the load capacity, as well as fluid film stiffness and damping, for all the rotors in the present study showed that the self generated bearing pressures were entirely adequate so that the journal bearings could in all cases, be of the self acting type. The self acting bearing type is preferred for space power units since it does not require a flow of high pressure liquid potassium. The special provision of a source of high pressure liquid potassium would impose a significant penalty on the system. Accordingly, self acting journal bearings were selected for the generator rotors.

The self-acting bearings for the space power generators have to satisfy the following criteria.

- 1 They must be inherently stable, even under zero net radial load, as in operation in a 0g field (This requirement rules out such bearings as the plain circular).

2. High load capacity, fluid film stiffness and fluid film damping are desirable for operation on a thick lubricant film, with margin for off design conditions, and for low vibration amplitudes under dynamic loads.
3. Self alignment provisions are desirable to tolerate thermally or otherwise induced misalignment. These are needed also so that when the dynamic rotor orbits are of a conical mode, the film thickness will remain essentially constant along the bearing length.
4. Low power loss, so as to minimize parasitic losses.

In view of these requirements, the tilting pad bearing which, properly designed, is inherently stable and self aligning, is the principal bearing choice. The tilting pad bearing also has somewhat higher fluid film stiffness and damping than other self acting bearing types. This is illustrated by a comparison recently made for four stable bearing types, intended for a potassium turbine rotor. The comparison was made on the following basis:

Rotor speed = 20,000 RPM

Lubricant = potassium at 950 F (viscosity = 2.34×10^{-3} lb.sec/in²
Density = 0.026 lbs/in³)

Bearing diameter = 2"

Bearing length = 2"

Bearing diametral clearance = 0.002"

At the concentric shaft position (i.e. with zero net load on the radial bearings), the calculated bearing stiffness and damping were:

Bearing Type	Radial Bearing Stiffness (lbs/inch)	Radial Bearing Damping (lb.sec/inch)
Tilting pad with 0.5 preload coefficient	12.4×10^4	71.4
Shrouded step	9.2×10^4	38.4
Herringbone grooved	7.6×10^4	46.2
Three lobe with 0.5 preload coefficient	3.2×10^4	51.9

Furthermore, tilting pad bearings have been successfully operated in liquid alkali metals, so that there exists a body of laboratory and field experience with these bearings that generally substantiates their calculated performance. NaK lubricated bearings are, for example, used in the NaK PMA pump of SNAP 8. Potassium lubricated tilting pad bearings have been tested to provide technology information under SNAP 50/SPUR. During the past several months, MTI has been conducting an AEC sponsored liquid metal bearings technology program in which a 50 lb. rotor supported on 2" diameter, NaK lubricated tilting pad bearings was operated over a speed range from 0 to 26,000 RPM. The steady load on the bearings was continuously varied from 0 to 80 psi, both with a balanced rotor and with 0.4 oz. inches of unbalance. Operation was entirely stable and the film thickness and dynamic orbit measurements that were made showed generally close agreement with the theoretically predicted performance.*

Both four and six pad tilting pad bearings were analyzed for the Westinghouse Corporation generator rotors, using the design data obtained and furnished here in Appendix C of this report. The four pad bearings proved to be superior from the standpoints of load capacity, fluid film stiffness and damping. The four pad bearings also have fewer elements than the six pad bearing and permit easier and more accurate assembly. The four pad bearing can also be designed to have nearly uniform stiffness and damping characteristics all around the bearing, which is advantageous from the standpoint of rotor-bearing dynamics.

2. Bearing Design Criteria

The design criteria listed below were adhered to in designing and sizing the tilting pad bearings for all the generators. These criteria are generally conservative and follow good design practice for liquid metal lubricated machinery over the size and speed ranges of the generators.

(a) Maximum, steady-state eccentricity ratio ≤ 0.5

The effective bearing eccentricity ratio, under the most adverse combination of gravity (lg) and magnetic loading was specified not to exceed 0.5. This provides adequate margin for dynamic loads and for off design conditions.

* The bearing design data given here in Appendix C is an extension of data obtained from this program, conducted under AEC contract AT(30-1)-3363.

(b) Maximum unit load at start up ≈ 50 psi

The bearings must be able to tolerate metal to metal, or boundary lubrication conditions at start-up and shut-down. An upper limit to the allowable unit pressures on the bearing pads at start-up as a function of materials combinations has not yet been established for liquid metal lubricated bearings. This is an important area of liquid metal bearings technology requiring attention in the near future. For the present study, a unit load of the order of 50 psi on the bearing (i.e. $50 \cos 45 = 35$ psi, based on projected area on each of the two pads under the shaft) was specified. This is a conservative value based on experience with low viscosity lubricants.

(c) Bearing preload and clearance ratio

Preloading^{*} of the bearing is necessary to insure stability and to prevent pad flutter even when the bearing carries no load (as in 0g environment). Accordingly bearing preloading was specified in this study, and a preload coefficient of 0.3 was used. Bearing clearance ratios ranging from 0.001 "/" to 0.0013 "/" were considered for this application and were found to be satisfactory from the standpoints of the dynamic bearing coefficients (fluid film stiffness and damping). The clearance ratio of 0.0013 "/" was selected as it provided the minimum power loss.

(d) Bearing dynamic characteristics

The bearing dynamic characteristics (i.e. fluid film stiffness and damping) had to be compatible with the stiffness characteristics of the rotor. The principal requirements that must be met are that (a) there are no flexural resonances at or near design speed and, (b) the fluid film damping (which is generally the only significant damping in the system) is sufficient so that the amplitudes of rotor vibration are small, when the critical speed regions are traversed. In addition, as is discussed later, the bearing support flexibilities must be selected so as to make maximum utilization of the fluid film damping.

* As described in Appendix C, tilting pad bearings can be preloaded by moving the pads in on each other at assembly, so that their centers of curvature form a small circle around the bearing center. In this way, all the individual bearing pads are positively loaded, when the shaft is rotating, even when the net bearing load is zero.

3. Steady-State Bearing Loads

The steady state bearing loads have two components:

- (a) the gravity force component (W_{grav}) due to the weight of the rotor in horizontal operation in a 1 g field.
- (b) the magnetic force component (W_{mag}) which is due to the magnetic stiffness at the pole faces and the eccentricity between the magnetic center and the center of rotation of the rotor at the poles.

The Westinghouse Corporation drawings series EDSK-332561 listed the rotor weights, exclusive of the stub shafts. The gravity load component on each journal bearing (W_{grav}) is half this quantity (since the rotors are symmetrical) plus the calculated weight of the stub shaft at each end of the rotor.

The Westinghouse drawings series EDSK-332561, also listed the magnetic stiffness (K_{mag}) for each generator rotor. Westinghouse Corporation also provided the following tolerance limits on manufacturing eccentricities (Total Indicator Reading) for the rotors and stators:

For the generators whose design identification number starts with the digits "10" or "20" (e.g. Generator Design No. 10827):

Maximum Stator Diametral Eccentricity: $2 \delta_S = 0.002''$

Maximum Rotor Diametral Eccentricity: $2 \delta_R = 0.0005''$

For generators whose design identification number starts with the digits "30" or "40" (e.g. Generator Design No. 30829):

Maximum Stator Diametral Eccentricity: $2 \delta_S = 0.004''$

Maximum Rotor Diametral Eccentricity: $2 \delta_R = 0.001''$

In addition to this there is the rotor to stator eccentricity at the pole faces due to (a) the eccentricity of the journals in the bearings and, (b) the rotor deflection at the pole faces, relative to its deflection at the journal bearings.

The maximum design value of the eccentricity at the bearings at design speed is $0.5 C_B$, where C_B is the radial bearing clearance. The calculation of rotor deflections, showed the rotors to be quite rigid at their design speeds and the rotor deflections at the pole faces, relative to their deflections at the journals were, in each case, less than 0.0005".

Finally, allowance should be made for tolerances between the centers of the bearing and the centers of the stators at assembly. These tolerances were set at:

$\Delta = 0.001"$ for the smaller rotors, i.e. those whose identification number starts with the digits "10" or "20".

and

$\Delta = 0.002"$ for the larger rotors, i.e. those whose identification number with the digits "30" or "40".

The maximum possible eccentricity of the pole faces is then the sum of absolute values of all the above eccentricity components, i.e. maximum pole face eccentricity = $\delta_S + \delta_r + 0.5 C_B + 0.0005 + \Delta$.

The upper limit of the magnetic load on each bearing, where the magnetic stiffness is K_{mag} , and the rotor is symmetrically supported in its two bearings is:

$$W_{mag} = \frac{1}{2} K_{mag} (\delta_S + \delta_r + 0.5 C_B + \Delta)$$

The highest value of the steady state load in the bearing occurs when the gravity and magnetic load components act along the same axis. Table I shows the maximum steady-state bearing load for each generator.

4. Calculated Bearing Performance Characteristics

The journal bearings were sized for each of the generators considered in this study, based on the design criteria and bearing loads noted above and using the general tilting pad bearing design data in Apperdix C.

The final design values are given for each generator bearing in Tables IIA through IIH. These tables show the operating conditions and bearing dimensions in each case.

The steady-state performance characteristics (effective eccentricity ratio, minimum pivot point film thickness and power loss) as well as the dynamic characteristics (fluid film stiffness and damping) are all tabulated as functions of speed, for three bearing load conditions. The speed range covered in every case spans the design speed and the bearing characteristics at the design speed are given. The three load conditions given in the tables are:

- (a) No Load - This corresponds to operation in vertical orientation, or in a zero g environment and in the absence of magnetic loading.
- (b) Normal Static Load - This is the normal operating condition of the rotor during its evaluation in horizontal orientation in a one g environment, and where the magnetic loads are zero, or small.
- (c) 160% Normal Static Load - This corresponds to the maximum steady-state load condition on the generator bearings. The load under this condition is the sum of the gravity (1g) and the magnetic loads, as obtained above and given in Table I. Review of the load data showed that the magnetic loading is generally equal to or less than 60% of the gravity load*. Accordingly, the bearing performance calculations for operation under maximum steady-state load were made at 160% of the normal static load.

Tables IIA through IIH show that, in each case, the design is very conservative from the standpoint of steady-state operation, the effective eccentricity ratio in all cases being less than 0.5 at design speed and under the maximum steady-state load. This allows substantial margin against off design conditions. It will also permit

* The only exception here, is Generator Design No. 30829. As shown in Table I, the maximum magnetic loading in this case is 785 lbs. and the normal static load is 1200 lbs., i.e. the maximum magnetic load amounts to 65.5% of the normal static load. However, the difference between the actual, maximum steady-state load on the bearings, $1200 + 785 = 1985$ lbs. and the value used in Table IIG, 1922 lbs., is small and will not significantly affect the calculated bearing performance data. In any case, as shown in Table IIH, the highest value of eccentricity ratio at design speed is 0.44, so that the design is very conservative.

operating the bearings at design speed for indefinite periods under moderate acceleration loads, of the order of 2g, provided adequate cooling flow is provided.

The bearing power loss is moderate and is acceptable in all cases. In the tabulation below, the power loss per bearing, given in Tables IIA through IIH for each rotor is expressed as a percentage of the rated generator output:

Generator Design No.	Calculated Power Loss per journal bearing at design speed and maximum load (KW)	Rated Generator Output (KW)	Calculated power loss per journal bearing, expressed as percentage of Rated Generator Output
10526	0.44	350	0.13%
10826	0.72	350	0.21%
10827	0.65	350	0.19%
20528	3.20	1000	0.32%
20827	4.51	1000	0.45%
20826	4.74	1000	0.47%
30829	13.80	5000	0.28%
40827	55.20	10000	0.55%

In all cases, therefore, the power loss in the two journal bearings is about 1% or less of the rated generator output. (This value does not include the bearing seal losses, since the seal design was not a part of the task reported herein. However, if viscous screw seals are used, the seal losses would be expected to be of the same order as the bearing losses listed above and would depend on the length, clearance, groove geometry and level of turbulence in the seals).

Tables IIA through IIH also show the calculated fluid film stiffness and damping coefficients. These stiffness and damping values were used in the final critical speed and unbalance response calculations given in Section III of this report.

5. Pivot and Support Disk Stiffnesses

In the critical speed and rotor response calculations, allowance has to be made for the finite stiffness of the pivots that support each pad and of the annular disk that supports each bearing assembly.

There are three basic pivot configurations as follows:

- (a) ball on flat
- (b) ball in cylindrical socket (axis of cylinder parallel to shaft axis)
- (c) ball in spherical socket

In pivot design, where a ball and socket arrangement is used, it is desirable for the socket radius to be larger than the radius of the ball, in order that the relative motions in the pivot be accommodated largely by small rolling motions, instead of sliding motions. (Clearly, if a fully conforming pivot is used, i.e. one where the ball and socket have identical radii, the relative motion is entirely accommodated by relative sliding. This design leads to pivot wear, aggravated by the fact that wear debris is continuously trapped in the pivot. Where, however, the socket radius is larger than the ball seat radius, the relative motions are accommodated partly by elastic deflections and partly by the ball seat rolling in the socket. Sliding motions and pivot wear are thereby minimized).

For operation at up to 800 F, as is the case for generators considered in this study, tool steel pivot elements hardened to about Rc 60 may be used. Alternately solid* carbide ball and socket elements may be used. For higher temperatures, say for the generators where the bearings operate at 1100 F, the solid tungsten carbide pivots should be used.

Another important design factor is the pivot stress. Since the pivot elements are in Hertzian contact, the compressive contact stresses will be quite high. Accordingly, the radii of curvature of the ball and socket elements have to be sized to maintain the contact stresses below a safe maximum. A conservative value of the maximum compressive stress for hardened tool steel pivots operating at temperatures up to about 800 F is 120,000 psi (i.e. a mean compressive stress, $s_{\text{mean}} = \frac{2}{3} s_{\text{max}} = 80,000 \text{ psi}$).

The pivot design selected here is that of a ball in a cylindrical socket. The radii of curvature of the pivots were sized to maintain a maximum compressive stress in the range of 60,000 to 120,000 psi under the highest static bearing load (i.e. $W_{\text{grav}} + W_{\text{mag}}$). The radius of curvature of the cylindrical socket was

*Carbide coatings of the pivot surfaces should not be used because of the likelihood of coating removal that may result from sliding friction under the high, Hertzian contact stresses.

made 2% larger than the radius of curvature of the ball seat to insure rolling, rather than sliding motions. Table III shows the diameters of curvature of the pivot elements, the total pivot stiffness of the bearing and the mean compressive stress in the most heavily loaded bearing pads, assuming tool steel pivots. The calculation procedure for pivot stiffness and maximum compressor stress are given in Appendix A.

As shown in Westinghouse Corporation drawing EDSK-332561, the journal bearing pads are supported in a housing ring which is itself supported in an annular disk. The dimensions and radial stiffness of this disk are given in Table IV, calculated as shown in Appendix A.

The disk thickness was selected so that its lowest natural frequency in the axial mode (which is its lowest resonance), is at least 35% higher than the design speed of the generator. This calculated pivot resonance frequency of the disk and bearing housing assembly is also given in Table IV and the calculation procedure is described in Appendix A. It may be noted here that this axial mode resonance frequency can readily be raised if desired by adding ribs to the annular disk.

6. Stub Shaft Dimensions

The stub shaft dimensions for each of the generator rotors are given in Table V. The stub shafts are identical at the two ends of each rotor. The stub shaft dimensions were arrived at, based on the calculated journal bearing diameters and lengths, and provide adequate axial length for centrifugal and viscous seals at each end of the bearings. Furthermore, as shown in the section on rotor-bearing dynamic response, the dynamic orbit amplitudes under the specified maximum unbalance values, are everywhere small.

III. ROTOR-BEARING DYNAMICS

The rotor-bearing dynamics analysis conducted here is a study of the response of the rotor-bearing system to rotating unbalance. The amplitudes of synchronous whirl of the rotor, in response to unbalance are quantitatively obtained to assess whether acceptable, (i.e., low synchronous vibration amplitude) operation can be realized at the design speed, in the permissible overspeed range and during passage through the system resonances that occur within the operating range.

The rotor-bearing dynamic analysis was conducted for three representative generators. The three representative generators, selected by Westinghouse Corporation for detailed rotor-bearing dynamics analysis, were generator design numbers 10827, 20827 and 30829. The rotor-bearings dynamics analysis was conducted as follows:

1. Preliminary critical speed maps were computed using the rotor data in the Westinghouse Corporation drawing EDSK 332561 and initial dimensions for the stub shaft, the bearings and the bearing supports.
2. The vibration response of the rotor in its bearings was calculated over a speed range from less than 50 percent of design speed to above 120 percent of design speed, assuming the highest unbalance value specified for each particular rotor in the Westinghouse Corporation drawing EDSK 332561.
3. Based on the data obtained in Steps 1 and 2 above, modifications were made to the stub shaft, bearing and bearing support dimensions, in order to reduce vibration amplitudes and to prevent peak amplitudes from occurring at the design speed. These steps were repeated until the final dimensions of the stub shaft, bearings and bearing supports were obtained which provided for acceptably low amplitudes of vibrations throughout the operating and overspeed ranges.

The final stub shaft and bearing support dimensions were earlier given in Tables II through V. In this section, the calculated critical speed maps and rotor-response data are presented and discussed. Before this, however, the dynamic model of the rotor-bearing support system is briefly described below.

1. Dynamic Model of the Rotor-Bearing System

The dynamic model of the bearing and bearing support is illustrated in Fig. I. The rotor is supported at each bearing, on the fluid film of the bearing. This fluid film is represented dynamically by a stiffness coefficient K_1 and a damping coefficient B_1 . The bearing pads, of total mass M_1 are in turn supported on the pivots which are represented by the stiffness coefficient, K_2 . The bearing housing ring of mass M_2 , which supports the pads through their pivots, is itself supported by the disk diaphragm which has a radial stiffness, K_3 . The disk diaphragm is rigidly connected to "ground" which is the generator casing.

The rotor itself is treated as a series of discrete masses and springs following the method of Prohl (Ref. 1). Twelve mass and stiffness stations were used in the generator rotor calculations located at the station indicated in Fig. II.

It is important to note there that the bearing fluid films are virtually the only sources of damping in rotor-bearing systems, such as the generators considered here. Accordingly, it is important in designing the bearing and support systems to maximize the fluid film bearing damping so as to suppress any rotor vibrations at the system critical speeds. To this end, it is desirable for the radial stiffness of the fluid film to be smaller than that of the bearing support. The radial deflections in the bearing and support system, in response to rotor vibrations, is then taken mainly by viscous squeeze film action in the fluid film rather than by elastic deformations of the bearing supports. For this reason, as may be seen from Tables II, III and IV, the bearing and support systems were sized such that the fluid film stiffness (K_1 of Fig. I) is smaller than the stiffnesses of the pivots and the disk diaphragm (K_2 and K_3 of Fig. I).

2. Undamped Critical Speed Maps

Quantitative assessment of the synchronous unbalance response of the generators from the standpoint of whether acceptable (i.e. low synchronous vibration amplitude) performance will be realized can only be obtained by performing response calculations that accurately account for the damping, as well as the stiffness elements of the system. From a practical standpoint, these unbalance response calculations can only be made using a digital computer. However, physical understanding of the unbalance responses, insofar as identifying which of the system vibration modes are participating, and what design changes may be made

to reduce unbalance amplitudes, is sometimes difficult to achieve from the response calculations alone. A useful prelude to the unbalance response calculations is the construction of the undamped critical speed maps which serve to approximately locate the system resonances and to identify, from the mode shapes, whether these are of the "rigid body" or of the "flexural" (bent shaft) type*.

The undamped critical speed map for each of the generators are shown in Figs. IIIA through IIIC. These were obtained using the dynamic modeling of the rotor, bearing and support systems discussed above except that the damping is neglected. A digital computer program was used which follows the procedure described in Ref.2. In addition, due to the relatively simple rotor configurations of the generator rotors, the first two critical speeds were also computed by hand calculations, as described in Appendix B.

Each of the undamped critical speed maps, Figs. IIIA through IIIC, were obtained as follows:

1. For the given rotor, bearing and support configurations, calculations were made for a range of values of the bearing fluid film stiffness. In these calculations a large speed range was scanned to establish the first, second and, in some cases, the third and fourth critical speeds. These were then plotted as functions of the fluid film stiffness.
2. The plots of the fluid film stiffness of the bearing as a function of speed for given load conditions, were superposed on the critical speed plots obtained in Step 1 above. The intersections of the critical speed vs fluid film stiffness plots and the fluid film stiffness vs speed curves denote the undamped system critical system speeds.

For example, referring to Fig. IIIA, which applies to generator design No. 10827, the first and second system critical speeds under normal static load conditions (i.e. when the steady-state bearing load is that corresponding to the rotor weight) occur at 8200 and 17,500 RPM respectively. For operation under 160% of normal static load (i.e. when the steady-state load in the bearings is the sum of the

* It should be cautioned that bearing damping will influence the frequencies at which the synchronous rotor vibrations (induced by rotor unbalance) reach maximum values, as well as the magnitudes of these vibrations. The true system resonant frequencies, as well as the rotor vibration amplitudes have to be obtained from the unbalance response calculations, which rigorously allow for the effects of damping.

rotor weight and at the highest value of magnetic load imposed at the poles), the first two system critical speeds occur at 10,000 and 21,000 RPM. For this particular rotor, therefore, operating in a horizontal orientation in a 1g field, the first undamped system critical speed will occur at some speed equal to or below 10,000 RPM, while the second undamped system critical speed will occur at some speed equal to or below 21,000 RPM, depending on the extent of magnetic loading on the bearings.

Under no load on the bearings (which corresponds to vertical operation or to operation in a zero g field and in the absence of magnetic loading), the first and second undamped system critical speeds will occur at very low speeds.*

Figures IIIA through IIIC show that in all cases the design speeds of the generator rotors are near or above the second undamped system critical speeds, the location of the critical speed being a function of the bearing load. In all cases, however, the mode shapes at the first two undamped system critical speeds, as well as the unbalance response calculations which are given later in this section, showed these to be "rigid body" critical speeds. At rigid body critical speeds, the rotor will oscillate essentially as a rigid body (i.e., with little or no bending deflections of the rotor axis), on the fluid film bearing "springs." With the relatively high damping that is provided by the fluid film of the tilting pad bearings, the amplitudes of the synchronous rotor vibrations at the rigid body critical speeds are small and do not exhibit any sharp resonant peaks, as shown by the unbalance response calculations given a little later in this section of the report.

The third and higher system critical speeds, on the other hand, constitute flexural

*In all the critical speed calculations shown in Figure III, the magnetic stiffness was introduced as a negative spring rate at the pole locations. Thus, when the bearing stiffness is very small, equal in magnitude but opposite in sign to the magnetic stiffness, both the first and second critical speed vs. film stiffness plots approach zero, as K_{film} approaches $(-K_{\text{magnetic}})$. This corresponds to the "free-free" support. Under similar calculations conducted neglecting any magnetic stiffness, the 1st and 2nd critical speed vs. film stiffness plots would have smaller slopes at the low film stiffness end of the range. These curves would then reach zero at $K_{\text{film}} = 0$. They would intersect the "no-load" film stiffness vs. speed curve, at small values of speed of the order of 1000 to 4000 RPM in the case of generator design No. 10827.

(i.e., bending) vibration modes of the rotor. At these resonances, rotors will generally exhibit large vibration amplitude peaks, with appreciable bending deflections of the rotor axis. Since the rotors generally do not possess any appreciable damping, except that provided at the bearing locations, it is generally desirable to design the rotors so that the flexural resonant frequencies are far removed from the normal operating speeds. As shown in Figures IIIA through IIIC, this is the case for all the generators considered here. For example, in the case of generator design No. 10827, the critical speed calculations were carried out for a speed range up to 50,000 RPM without encountering the third critical speed. In the case of generator design No. 20827, the third critical speed is shown in Figure IIIB to occur at 48,000 RPM while the design speed is 19,200 RPM. Similarly, in the case of generator design No. 30829, the third critical speed is shown in Figure IIIC to occur at 35,000 RPM, whereas the design speed is 12,000 RPM.

3. Unbalance Response Calculations

The forced vibration response of each of the generator rotors was calculated using an existing computer program, following the method described in Ref. 2. This program rigorously allows for the influence of the damping provided by the fluid film of the journal bearings.

The mass unbalances assigned to the rotor in each case were the maximum allowable unbalances specified in the Westinghouse Corporation drawings series EDSK 332561. The axial locations of the mass unbalances, were at the dynamic balance correction planes specified in the drawings series EDSK 332561, i.e., at the outer faces of the poles.

The circumferential locations of the residual mass unbalances, in the two dynamic balance correction planes are, of course, arbitrary. Accordingly, there are two limiting cases that have to be considered. These are:

(a) "In-phase" unbalance

In this case, mass unbalances in the two planes are assumed to be at the same locations circumferentially. This distribution of the mass unbalances tends to excite the translatory vibration mode of the rotor.

(b) "Out-of-Phase unbalance"

In this case, the circumferential location of the mass unbalance in one plane is assumed to be 180° away from the location of the mass unbalance in the second plane. This distribution of the mass unbalance tends to excite the conical vibration mode of the rotor.

The unbalance response calculations were conducted for each generator, for both "in-phase" and "out-of-phase" mass unbalance conditions.

Figures IVA through IVC show the results of the unbalance response calculations at the design speed for each generator. The figures show the amplitudes of vibration at each point along the rotors. They indicate that, with the "in-phase" locations of the mass unbalances, the vibration amplitudes are very small and nearly equal along the full length of the rotor. The "out-of-phase" location of the mass unbalances causes larger but still relatively low amplitudes of vibration and the mode shape is conical, i.e. the vibration amplitudes vary almost linearly along the rotor length. The dynamic orbits are maximum at the ends of the rotors and zero at the mid-planes of the rotors.

The amplitudes denoted in Figs. IVA through IVC, are the radii of the synchronous, vibration orbits. Thus, Fig. IVA which applies to generator design No. 10827, shows that, at each journal bearing, the rotor at design speed and under the maximum mass unbalance conditions will perform a synchronous dynamic orbit whose radius is about 70 microinches. Since the radial clearance of the bearing is 1.3 mils, the maximum dynamic orbit at the journal bearing is only about 5.4% of the clearance. This is very small and operation under this condition is entirely acceptable. The vibration orbits are still smaller at the generator poles, the radius of the synchronous dynamic orbits at these locations being about 25 microinches. The maximum orbits occur at the ends of the rotor, though here again they are quite small, about 100 microinches in radius.

The rotating mass unbalances also cause a dynamic force to be transmitted through the bearing to the generator casing. These dynamic, transmitted forces are also listed in Figs. IVA through IVC. For the case of generator design No. 10827, Fig. IVC shows the transmitted force to be 3.5 lbs. for the in-phase mass unbalance condition and 15.8 lbs. for the out-of-phase mass unbalance condition.

These are again acceptably small, the higher one being 23.6% of the normal static load (67 lbs. per bearing, as shown in Table IIA).

Equally acceptable rotor-bearing dynamic operation at design speed is shown for each of the other rotors in Figs. IVB and IVC. In the case of rotor 20827, for example, Fig. IVB shows the highest synchronous response orbit radii to be 100, 70 and 25 microinches respectively at the rotor ends, at the bearing centerlines and at the pole face centerlines, while the highest transmitted force is 35.2 lbs. per bearing. In the case of generator No. 30829, Fig. IVC shows the highest synchronous response orbits to be 280, 160 and 70 microinches respectively at the rotor ends, at the bearing centerlines and at the pole face centerlines. The bearing orbits in this case are about 4.85% of the radial clearance. The maximum transmitted load is 85.7 lbs. i.e. about 7.5% of the normal static load (1202 lbs. per Table IIC).

The data in Figs. IVA through IVC was obtained for the bearing stiffness and damping values corresponding to the highest bearing load (160% of normal static load), as this condition was found to give the highest values of dynamic orbits and transmitted loads.

Figures IVA through IVC therefore show that, for the selected design values of the stub shaft, bearing and bearing support dimensions given earlier in this report, the rotor-bearing dynamic operation is entirely satisfactory at design speed. It is necessary, in addition, to study the performance of the rotors over the complete speed range, including the speeds at which the first and second system resonances are traversed, and at the overspeed conditions. These were also computed and the results are given in Figs. VA through VC. These figures show the radius of the dynamic, synchronous orbits at the bearings at all speeds up to and above the design speed. The two curves in each figure represent the in-phase and out-of-phase mass unbalance locations, as noted. The upper envelope marked by these two curves therefore shows the highest bearing orbit amplitudes over the full speed range. It will be noted from these figures that the in-phase mass unbalance locations produce a peak in the vibration amplitudes at the first system critical speed, while the out-of-phase unbalance excites a peak at the second system critical speed. This is due the fact that, because of the relatively long span between

bearings, the mode shape at the first critical speed is translatory, whereas the mode shape at the second critical speed is conical. The vibration amplitudes at these rigid body critical speeds are, however, in all cases small, due to the relatively high damping in the bearings. The data in Figs. VA through VC was again obtained for the maximum steady-state bearing load condition, as this gave the highest vibration amplitudes. The specified maximum values of mass unbalance were used in computing the data.

It will be noted by comparing the frequencies at which the amplitudes are maximum in Figs. VA through VC with the undamped system critical speeds noted in Figs. IIIA through IIIC that the frequencies at which the amplitudes of vibration reach their maxima are somewhat higher than the calculated, undamped system critical speeds. This is due to the damping in the fluid film bearings which is rigorously included in the unbalance response calculations but neglected in the undamped critical speed calculations.

One additional set of unbalance response calculations was made. This set of calculations was made at the design speed of each generator rotor, assuming maximum mass unbalances, located 180° out of phase, and varying the fluid film stiffness and damping values over a range that extended from below the no-load bearing stiffness and damping values to above the maximum-load bearing stiffness and damping values. The results of these calculations shown in Figs. VIA through VIC, are plots of the synchronous vibration amplitudes at the bearings, as functions of the fluid film stiffness.

Figures VIA through VIC show that the vibration amplitudes are acceptably small throughout the complete range of bearing operation from zero load to 160% of normal static load. It shows also that small differences between actual bearing stiffness and damping and the design values of bearing stiffness and damping, as may result from the tolerances on bearing clearances, will not significantly influence the satisfactory operation of the rotor-bearings systems.

4. Discussion of Maximum Allowable Unbalance

The degree of residual unbalance to which high speed turbomachinery rotors can be balanced has been the subject of much investigation in the past. Several

formulas and balancing specification tables have been prepared, as functions of rotor weight, speed, number of balancing planes available, rotor rigidity etc. References 3 and 4 provide some of these balancing specifications.

All the generator rotors considered here have essentially rigid rotors at these design speeds. Based on the data in References 3 and 4, the appropriate balancing designations and achievable residual unbalance are as follows:

"High Quality Balancing" - used for high speed - 0.015 to 0.075 oz.in.per
 electric motors, 100 lbs of rotor weight
 turbogenerators,
 steam turbines,
 gas turbines, etc.

"Good Quality Balancing" - used for commercial - 0.030 to 0.150 oz.in.per
 electric motors, fans, 100 lbs. of rotor weight.
 blowers, centrifugal
 pumps, etc.

The unbalance limits specified for the generators in the Westinghouse Corporation drawing series EDSK-332561 are as follows:

Generator Design Number	Rotor Weight (including stub shaft) lbs.	Maximum Unbalance in each tolerance plane oz.in.	Maximum Unbalance in oz.in. per 100 lbs. of rotor weight
10526	116	0.012	0.021
10826	120	0.012	0.020
10827	134	0.017	0.026
20528	402	0.051	0.026
20827	496	0.082	0.033
20826	562	0.120	0.043
30829	2404	0.960	0.080
40827	7574	5.20	0.138

The specified values of maximum residual unbalance are, therefore, within the obtainable limits for "High Quality" and "Good Quality" balancing.

The unbalance response calculations described in Part 3 of this section, showed that within the specified unbalance limits, good rotor dynamic operation could be realized, the dynamic orbits in the bearings at design speed being generally of the order of 5% or less of the clearance.

It is clearly desirable to maintain the residual unbalance and, hence, the dynamic vibration orbits and the force transmitted to the generator casing as small as possible, and thus provide a wide margin of safety against off design operation. Accordingly, it is recommended that the unbalance limits currently specified in the Westinghouse Corporation series EDSK 332561 drawings be adhered to.

For rigid rotor operation and where the dynamic orbits are small compared to the bearing clearance (say up to about 20% of bearing clearance), the orbit size and the transmitted force are directly proportional to the magnitude of the mass unbalance. Thus, if the unbalance limits were raised or if they showed modest deterioration in the course of operation, the previously given values of the orbit sizes and transmittal force would increase in proportion with the increase of unbalance. Insofar as the bearings and pivots are concerned, rotor orbits up to about 10% of the clearance and dynamic loads transmitted through the pivots up to about 40% of the static load can be deemed acceptable. Based on this and the calculated orbits and transmitted force, it appears that deterioration of the unbalance in the course of operation up to about 50% of the currently specified values would be very safely tolerated over the life of the machines.

IV. ADDITIONAL CONSIDERATIONS

The steady-state bearing performance characteristics and the rotor-bearings dynamic analyses of the space power generators described in Sections II and III of this report completed the task requirements, as defined in the Foreword of the report. It was concluded that acceptable steady-state and dynamic performance with margin for off-design operation can be realized for each of the generators considered. The designs of the stub shafts, the journal bearings and the bearing supports that will permit acceptable steady-state and dynamic performance of the rotor-bearings were defined.

In this section of the report, some other components that bear in the mechanical performance of the generators, but which were outside the scope of the task reported herein are briefly noted. In addition some important areas where technology is needed for sound mechanical design of the generators are identified.

1. Thrust Bearing, Journal Bearing Seals and Drive Coupling

For horizontal operation and assuming that the drive coupling does not transmit any significant thrust loads from the drive turbine, the axial load in the radial gap generators will be very small. A positioning thrust bearing of nominal size would then suffice for each generator rotor. If, however, it is necessary to operate the generators vertically or under high acceleration loads a larger thrust bearing would be required. The addition of a thrust collar could then have some influence on the rotor dynamics of the generator rotors, depending on the size and location of the thrust collar. Accordingly, some of the critical speed and unbalance response computations given in Section III above should be reviewed and where necessary revised, to allow for the added thrust collar mass after the size and location of the thrust collar have been established.

Assuming the generator cavity to be vented to the system condenser, the seals at each end of the journal bearings may comprise a step or centrifugal seal, in series with a viscous (screw) seal. In the stub shaft dimensions, given in Table V, adequate free axial length was allowed on each side of the journal bearings for such seals. If, however, changes in the stub shaft diameters or lengths in the seal regions are made at the time when the seals are designed,

the effect of such changes on the rotor stiffness and, hence, on the critical speeds and unbalance response of the system should be reviewed.

No allowance was made in the rotor-bearings dynamic analysis for any radial loads or moment restraints introduced by the coupling that will connect each generator rotor to its drive turbine. These couplings should be flexible, and capable of operating at the system temperatures and not require lubrication. Suitable flexible couplings for this application include the flexible diaphragm and other types. Such couplings will transmit the drive torque without imposing significant radial loads on moment restraints. This should, however, be established at the time of design and selection of the coupling. It should be noted here that the unbalance response calculations given in Section III, showed that the calculated dynamic orbits at the free ends of the generator rotor, under the specified unbalance loads, range from less than 0.001" diameter for the smaller generators, up to about 0.002" diameter for the larger units. These deflections are small and well within the capacity of well designed flexible couplings.

2. Bearing Materials

The selection of bearing materials for good compatibility between the rotor and stator faces in the liquid potassium environment herein. Previous studies (e.g. Ref. 6) have generally confirmed that tungsten carbide rotor and stator faces have good compatibility in liquid metal environments. The use of carbide journals and bearing pads, however, poses design problems due to the difference in the thermal expansion of carbides and that of the rotor and housing structural materials. While carbide journal sleeves can be designed for attachment to steel rotors, such that differential thermal expansions are accommodated by relative growths between the journal and sleeve, these generally add complexity to the design and in many cases introduce additional undesirable flexibility in the journal bearing. An alternative approach is the use of thin coatings on the journal and bearing pad surfaces. The development and quality control of ceramic coated bearings remains an important technology area for liquid metal space power turbomachinery.

3. Alternative Bearing Configurations

As noted in Section II of the report, tilting pad journal bearings are the pre-

ferred because of their inherent stability, self-alignment and superior fluid film stiffness and damping properties. Furthermore, there is more successful high speed, liquid metal experience with these than with any other self acting bearings. Nevertheless, there are other known, stable bearing configurations that can also be used in high speed turbomachinery applications lubricated with low viscosity fluids such as liquid metals. Such bearing types include the shrouded step and the herringbone grooved bearing types. These have merit of one piece construction and the absence of Hertzian contact pivots. These bearings also offer the possibility of incorporating part of the seal in the bearing. Additional technology effort is desirable to optimize such bearings and evaluate them in liquid metal environment, so that a sound comparison may be made of their advantages and disadvantages, relative to the tilting pad bearings in high speed, liquid metal lubricated space power turbomachinery.

4. Turbulent Lubrication Theory Extension

The calculated bearing performance data given in Tables II, shows that the mean fluid film Reynolds Number at design speed, based on the machined pad-to-shaft clearance (as defined in Appendix C), is in the range of 6000 to 30,000 for the smaller generators, i.e. those whose rated output is up to 1000 KW. Since the bearings are preloaded, with a preload coefficient, $m = 0.3$, the actual bearing clearance is 70% of the machined pad-to-shaft clearance. The effective Reynolds Number in the film is therefore $0.7 \times 30,000 = 21,000$.

For the tilting pad bearings, with normal values of bearing clearance in the range of 0.001 "/" to 0.002 "/", the present turbulent lubrication theory is entirely adequate for design purposes up to Reynolds Numbers of the order of 20,000. It has been experimentally verified at Reynolds Numbers up to 15000.

At large Reynolds Numbers ($>20,000$ for $C/R \leq 0.002$ "/") however, it is expected (Ref. 7) that fluid inertia effects will become increasingly important. These fluid inertia effects (which basically deal with the energy required to accelerate the fluid in the film and the partial recovery of the velocity heads generated in the film) are not now included in the theory. Accordingly, the calculated data for the larger generators (power rating > 1000 KW), where the bearing Reynolds Number is in the range of 30,000 to 60,000, are approximate. An important bearing

technology area for such large machines lubricated with low viscosity fluids,
is the extension of the turbulent lubrication theory to include fluid inertia
effects and the experimental verification of such new theory.

TABLE I
STEADY STATE LOADS ON THE GENERATOR BEARINGS

Generator Design No.	Rotor Weight exclusive of stub shafts (lbs)	Gravity load per bearing (W_{grav} , lbs)	Magnetic Stiffness (K_{mag} , lbs)	Maximum Total eccentricity at pole faces ($\delta_s + \delta_r + \Delta + 0.5C_B + 0.0005$)*	Maximum Magnetic load per bearing (W_{mag} , lb)	Maximum Steady state load per bearing ($W_{grav} + W_{mag}$) lbs.
10526	112	58	21,000	0.00324	34	92
10826	112	60	21,000	0.00334	35	95
10827	124	67	23,000	0.00340	39	106
20528	375	201	55,000	0.00366	101	302
20827	442	248	61,000	0.00389	119	367
20826	480	280	55,000	0.00405	111	391
30829	2128	1200	236,000	0.00665	785	1985
40829	6480	3780	450,000	0.00793	1785	5565

* The values of the radial bearing clearances C_B are given in Tables II for each rotor.

TABLE IIA
CALCULATED BEARING PERFORMANCE CHARACTERISTICS

Generator Design No. 10827

Design Speed = 24,000 RPM

Lubricant = potassium at 800F

Viscosity = 0.2758×10^{-7} lbsec/in²

Mass Density = 0.6896×10^{-4} lbsec²/in⁴

Bearing Diameter, D = 2.0 in.

Bearing Length, L = 1.0 in.

Bearing Clearance Ratio, $C_p/R = 1.3 \times 10^{-3}$ in/in₃

Pad to Shaft Clearance Ratio, $C_p/R = 1.857 \times 10^{-3}$ in/in

Bearing Preload Coefficient, $m = 0.3$

Four Pad Bearings: $\beta = 80^\circ$; $\theta_p/\beta = 0.55$

Steady-State Load Direction is midway between pivots

Unit load at start up, $W_{\text{grav}}/LD = 34$ psi

Load Condition	Steady State Load W, lbs.	Speed (60N) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness $(h_p)_{\text{min}}$, in.	Power Loss per Brg.	Fluid Film Stiffness $K_x = K_y$ lbs/in.	Fluid Film Damping $B_x = B_y$ lbsec/in
No Load	0	10,000	4,869	0	1.300×10^{-3}	0.075	19,000	28.8
	0	20,000	9,738	0	1.300×10^{-3}	0.489	56,500	43.5
	0	24,000	11,686	0	1.300×10^{-3}	0.815	75,500	49.0
	0	30,000	14,600	0	1.300×10^{-3}	1.425	109,800	56.8
Normal	67	10,000	4,869	0.62	0.495×10^{-3}	0.086	143,000	101.2
Static	67	20,000	9,738	0.37	0.820×10^{-3}	0.505	112,000	65.2
Load	67	24,000	11,686	0.30	0.900×10^{-3}	0.855	117,000	64.0
	67	30,000	14,600	0.23	1.000×10^{-3}	1.49	141,300	64.5
150%	107	10,000	4,869	0.72	0.364×10^{-3}	0.100	302,000	115.4
Normal	107	20,000	9,738	0.47	0.690×10^{-3}	0.502	163,000	70.9
Static	107	24,000	11,686	0.40	0.770×10^{-3}	0.875	168,000	64.2
Load	107	30,000	14,600	0.33	0.870×10^{-3}	1.501	191,000	58.9

TABLE IIB
CALUCLATED BEARING PERFORMANCE CHARACTERISTICS

Generator Design No. 20827

Design Speed = 19,200

Lubricant = potassium at 7800 F

Viscosity = 0.2758×10^{-7} lbsec/in²

Mass Density = 0.6869×10^{-4} lbsec²/in⁴

Bearing Diameter, D = 3.5 in.

Bearing Length, L = 1.75 in.

Bearing Clearance Ratio, $C_B/R = 1.3 \times 10^{-3}$ in/in³

Pad to Shaft Clearance Ratio, $C_P/R = 1.857 \times 10^{-3}$ in/in

Preload Coefficient, m = 0.3

Four Pad Bearing: $\beta = 80^\circ$; $\theta_p/\beta = 0.55$

Steady-State Load Direction is midway between pivots

Unit Load at start up, $W_{grav}/LD = 41$ psi

Load Condition	Steady State Load W, Lbs.	Speed (60N) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness $(h_p)_{min}$, in.	Power Loss per Brg.	Fluid Film Stiffness $K_x = K_y$ lbs/in	Fluid Film Damping $B_x = B_y$ lbsec/in
No Load	0	8,000	11,930	0	2.275×10^{-3}	0.470	45,000	87.0
	0	15,000	22,360	0	2.275×10^{-3}	2.645	126,900	131.7
	0	19,200	29,100	0	2.275×10^{-3}	5.100	190,000	157.0
	0	25,000	37,280	0	2.275×10^{-3}	10.700	300,000	186.8
Normal Static Load	248	8,000	11,930	.61	0.890×10^{-3}	0.502	280,000	243.9
	248	15,000	22,360	.37	1.480×10^{-3}	3.215	253,800	190.7
	248	19,200	29,100	.29	1.62×10^{-3}	5.750	270,000	192.0
	248	25,000	37,280	.21	1.800×10^{-3}	10.600	359,000	200.5
160% Normal Static Load	397	8,000	11,930	.72	0.637×10^{-3}	0.248	700,000	435.0
	397	15,000	22,360	.48	1.180×10^{-3}	2.880	341,000	211.0
	397	19,200	29,100	.39	1.49×10^{-3}	6.050	332,000	200.0
	397	25,000	37,280	.30	1.590×10^{-3}	11.150	442,000	222.0

TABLE IICCALCULATED BEARING PERFORMANCE CHARACTERISTICSGenerator Design No. 30829

Design Speed = 12,000 RPM

Lubricant = potassium at 800 F

Viscosity = 0.2758×10^{-7} lbsec/in²Mass Density = 0.6896×10^{-4} lbsec²/in⁴

Bearing Diameter, D = 6.0 in.

Bearing Length, L = 3.0 in.

Bearing Clearance Ratio, $C_p/R = 1.3 \times 10^{-3}$ in/in⁻³Pad to Shaft Clearance Ratio, $C_p/R = 1.857 \times 10^{-3}$ in/in.Preload Coefficient, $m = 0.3$ Four Pad Bearing: $\beta = 80^\circ$; $\phi_p/\beta = 0.55$

Steady-State Load Direction is midway between pads

Unit Load at start up = 67 psi

Load Condition	Steady State Load W, lbs.	Speed (bON) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness (h) _{p min.} in.	Pivot Power Loss per Brg.	Fluid Film Stiffness $K_x = K_y$ lbs/in	Fluid Film Damping $B_x = B_y$ lbsec/in
No Load	0	4,000	14,790	0	3.30×10^{-3}	0.705	64,100	248
	0	8,000	29,590	0	3.30×10^{-3}	4.485	189,000	367
	0	12,000	44,380	0	3.30×10^{-3}	13.25	360,000	460
	0	15,000	55,480	0	3.30×10^{-3}	23.10	516,000	536
Normal Static Load	1202	4,000	14,790	0.74	0.86×10^{-3}	0.975	1,424,000	1670
	1202	8,000	29,590	0.50	1.65×10^{-3}	5.90	714,000	793
	1202	12,000	44,380	0.31	2.20×10^{-3}	18.0	740,000	745
	1202	15,000	55,480	0.24	2.51×10^{-3}	31.2	914,000	838
160% Normal Static Load	1922	4,000	14,790	0.82	0.59×10^{-3}	1.05	3,420,000	2127
	1922	8,000	29,590	0.61	1.29×10^{-3}	6.00	1,370,000	1210
	1922	12,000	44,380	0.44	1.75×10^{-3}	18.50	1,150,000	990
	1922	16,000	55,480	0.37	2.08×10^{-3}	32.50	1,180,000	939

TABLE IIDCALCULATED BEARING PERFORMANCE CHARACTERISTICSGenerator Design No. 10526

Design speed = 30,000 RPM

Lubricant = potassium at 500F

Viscosity = 0.3838×10^{-8} lb.sec/in²Mass Density = 0.728×10^{-4} lb.sec²/in⁴

Bearing Diameter, D = 1.5 in.

Bearing Length, L = 0.75 in.

Bearing Clearance Ratio, $C_B/R = 1.3 \times 10^{-3}$ in/in.Pad to Shaft Clearance Ratio, $C_P/R = 1.857$ in/in.

Bearing Preload Coefficient, m = 0.3

Four Pad Bearing: $\beta = 80^\circ$; $\theta_p/\beta = 0.55$

Steady-State Load Direction is midway between pads

Unit load at start up, $W_{grav}/LD = 52$ psi.

Load Condition	Steady State Load W, lbs.	Speed (60N) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness $(h_p)_{min}$, in.	Power Loss per Brg. HP	Fluid Film Stiffness $K_x = K_y$ lbs/in.	Fluid Film Damping $B_x = B_y$ lb.sec/in
No Load	0	10,000	2079	0	0.975×10^{-3}	0.025	13,210	19.5
	0	20,000	4159	0	0.975×10^{-3}	0.150	35,300	26.6
	0	30,000	6237	0	0.975×10^{-3}	0.465	63,800	33.0
	0	40,000	8319	0	0.975×10^{-3}	1.015	102,800	39.5
Normal Static Load	58	10,000	2079	0.71	0.283×10^{-3}	0.035	241,000	116.3
	58	20,000	4159	0.52	0.468×10^{-3}	0.166	143,000	56.8
	58	30,000	6239	0.38	0.600×10^{-3}	0.585	142,000	50.0
	58	40,000	8319	0.28	0.702×10^{-3}	1.02	154,700	49.0
160% Normal Static Load	93	10,000	2079	0.79	0.205×10^{-3}	0.040	52,200	189.3
	93	20,000	4159	0.64	0.351×10^{-3}	0.171	283,000	89.2
	93	30,000	5239	0.50	0.490×10^{-3}	0.585	220,000	66.2
	93	40,000	8319	0.35	0.633×10^{-3}	1.00	196,000	56.6

TABLE IIE
CALCULATED BEARING PERFORMANCE CHARACTERISTICS

Generator Design No. 10826

Design speed = 30,000 RPM

Lubricant = potassium at 800F

Viscosity = 0.2758×10^{-7} lb.sec/in²

Mass Density = 0.6896×10^{-4} lb.sec/in²

Bearing Diameter, D = 1.8 inches

Bearing Length, D = 0.90 inches

Bearing Clearance Ratio, $C_P/R = 1.3 \times 10^{-3}$ in/in³

Pad to Shaft Clearance Ratio, $C_P/R = 1.857 \times 10^{-3}$ in/in

Bearing Preload Coefficient, $m = 0.3$

Four Pad Bearing: $\beta = 80^\circ$; $\theta_P/\beta = 0.55$

Steady State Load Direction is midway between pivots

Unit Load at startup, $W_{grav}/LD = 36$ psi

Load Condition	Steady State Load W, lbs	Speed (60N) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness $(h_p)_{min}$, in.	Power Loss per Brg.	Fluid Film Stiffness $K_x = K_y$ lbs/in.	Fluid Film Damping $B_x = B_y$ lb.sec/in
No Load	0	10,000	3,944	0	1.170×10^{-3}	0.046	15,180	22.8
	0	20,000	7,888	0	1.170×10^{-3}	0.303	44,400	34.1
	0	30,000	11,832	0	1.170×10^{-3}	0.92	83,000	42.9
	0	40,000	15,776	0	1.170×10^{-3}	1.99	130,000	50.3
Normal Static Load	60	10,000	3,944	0.69	0.363×10^{-3}	0.056	165,900	95.5
	60	20,000	7,888	0.42	0.678×10^{-3}	0.319	110,400	56.8
	60	30,000	11,832	0.26	0.870×10^{-3}	0.90	120,000	54.0
	60	40,000	15,776	0.14	1.010×10^{-3}	2.01	154,500	57.2
160% Normal Static Load	96	10,000	3,944	0.76	0.281×10^{-3}	0.065	352,000	144.6
	96	20,000	7,888	0.54	0.538×10^{-3}	0.325	188,000	73.7
	96	30,000	11,832	0.40	0.705×10^{-3}	0.96	178,000	63.8
	96	40,000	15,776	0.28	0.843×10^{-3}	1.95	196,000	63.2

TABLE IIFCALCULATED BEARING PERFORMANCE CHARACTERISTICS

Generator Design No. 20528

Generator Speed = 24,000 RPM

Lubricant = potassium at 500 F.

Viscosity = 0.3838×10^{-8} lbsec/in²Mass Density = 0.7286×10^{-4} lbsec²/in⁴

Bearing Diameter, D = 2.8 in.

Bearing Length, L = 1.4 in.

Bearing Clearance Ratio, $C_p/R = 1.3 \times 10^{-3}$ in/in³Pad to Shaft Clearance Ratio, $C_p/R = 1.857 \times 10^{-3}$ in/inBearing Preload Coefficient, $m = 0.3$ Four Pad Bearing: $\beta = 80^\circ$; $\theta_p/\beta = 0.55$

Steady-State Load Direction midway between pivots

Unit Load at startup, $W_{grav}/LD = 52$ psi

Load Condition	Steady State Load W, lbs.	Speed (60N) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness $(h_p)_{min}$, in.	Power Loss per Brg.	Fluid Film Stiffness $K_x = K_y$ lbs/in	Fluid Film Damping $B_x = B_y$ lbsec/in
No Load	0	10,000	7,247	0	1.82×10^{-3}	.398	48,100	73.9
	0	20,000	14,490	0	1.82×10^{-3}	2.505	142,300	110.6
	0	24,000	17,390	0	1.82×10^{-3}	4.000	185,000	120.0
	0	30,000	21,740	0	1.82×10^{-3}	6.975	259,000	134.2
Normal	201	10,000	7,247	.59	$.75 \times 10^{-3}$	0.435	283,000	202.5
Static	201	20,000	14,490	.35	1.19×10^{-3}	2.508	266,000	155.8
Load	201	24,000	17,390	.29	1.30×10^{-3}	4.010	280,000	152.0
	201	30,000	21,740	.21	1.44×10^{-3}	7.050	323,000	151.0
160%	322	10,000	7,247	.70	$.55 \times 10^{-3}$	0.920	641,000	330.0
Normal	322	20,000	14,490	.45	1.00×10^{-3}	2.615	383,000	189.0
Static	322	24,000	17,390	.38	1.14×10^{-3}	4.300	376,000	176.5
Load	322	30,000	21,740	.29	1.29×10^{-3}	7.650	417,000	176.0

TABLE IIGCALCULATED BEARING PERFORMANCE CHARACTERISTICSGenerator Design No. 20826

Design Speed = 16,000 RPM

Lubricant = potassium at 7800 F
Viscosity = 0.2758×10^{-7} lbsec/in²Mass Density = 0.6896×10^{-4} lbsec²/in⁴

Bearing Diameter, D = 4.0 in.

Bearing Length, L = 2.0 in.

Bearing Clearance Ratio, $C_B/R = 1.3 \times 10^{-3}$ in/in₃Pad to Shaft Clearance Ratio, $C_P/R = 1.857 \times 10^{-3}$ in/in

Preload Coefficient, m = 0.3

Four Pad Bearing: $\beta = 80^\circ$; $\theta_p/\beta = 0.55$

Steady-State Load Direction is midway between pivots

Unit Load at start up, $W_{grav}/LD = 35$ psi

Load Condition	Steady State Load W, lbs.	Speed (60N) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness $(h_p)_{min}$, in.	Power Loss per Brg.	Fluid Film Stiffness $K_x = K_y$ lbs/in	Fluid Film Damping $B_x = B_y$ lbsec/in
No Load	0	6,000	11,680	0	2.60×10^{-3}	0.40	37,900	97.1
	0	10,000	19,470	0	2.60×10^{-3}	1.63	88,600	137.9
	0	16,000	31,150	0	2.60×10^{-3}	5.90	201,000	190.0
	0	25,000	48,690	0	2.60×10^{-3}	19.95	421,400	264.0
Normal	281	6,000	11,680	.65	0.91×10^{-3}	0.44	317,300	393.9
Static	281	10,000	19,470	.45	1.43×10^{-3}	1.69	231,400	223.0
Load	281	16,000	31,150	.27	1.86×10^{-3}	6.40	275,000	215.0
	281	25,000	48,690	.12	2.28×10^{-3}	19.80	450,000	269.8
100%	449	6,000	11,680	.74	0.68×10^{-3}	0.47	64,300	511.6
Normal	449	10,000	19,470	.56	1.14×10^{-3}	1.72	400,000	313.3
Static	449	16,000	31,150	.32	1.75×10^{-3}	6.35	360,000	260.0
Load	449	25,000	48,690	.14	2.24×10^{-3}	19.90	500,000	282.6

TABLE IIHCALCULATED BEARING PERFORMANCE CHARACTERISTICS

Generator Design No. 40827

Design speed = 8000 RPM

Lubricant = potassium at 800 F
Viscosity = 0.2758×10^{-7} lbsec/in²Mass Density = 0.6896×10^{-4} lbsec²/in⁴

Bearing Diameter, D = 9.0 in.

Bearing Length, L = 9.0 in.

Bearing Clearance Ratio, $C_B/R = 1.3 \times 10^{-3}$ in/in₃Pad to Shaft Clearance Ratio, $C_P/R = 1.857 \times 10^{-3}$ in/in

Preload Coefficient, m = 0.3

Four Pad Bearing: $\beta = 80^\circ$, $\theta_p/\beta = 0.55$

Steady-State Load Direction is midway between pivots

Unit Load at start up = $W_{\text{grav}}/LD = 47$ psi

Load Condition	Steady State Load W, lbs.	Speed (60N) RPM	Mean Reynolds Number	Eff. Ecc. Ratio ϵ_{EFF}	Minimum Pivot Point Film Thickness $(h_p)_{\text{min}}$, in.	Power Loss per Brg.	Fluid Film Stiffness $K_x = K_y$ lbs/in	Fluid Film Damping $B_x = B_y$ lbsec/in
No Load	0	3,000	29,580	0	5.85×10^{-3}	4.02	158,000	825
	0	5,000	49,300	0	5.85×10^{-3}	18.7	380,000	1198
	0	8,000	78,880	0	5.85×10^{-3}	71.0	840,000	1680
	0	10,000	98,600	0	5.85×10^{-3}	132.0	1,270,000	2050
Normal	3787	3,000	29,580	0.57	2.52×10^{-3}	4.98	1,330,000	3680
Static	3787	5,000	49,300	0.36	3.75×10^{-3}	19.9	1,440,000	3740
Load	3787	8,000	78,880	0.17	4.80×10^{-3}	72.5	2,140,000	4300
	3787	10,000	98,600	0.09	5.33×10^{-3}	134.5	3,100,000	5040
160%	6060	3,000	29,580	0.67	1.93×10^{-3}	5.18	2,350,000	5840
Normal	6060	5,000	49,300	0.44	3.28×10^{-3}	20.1	1,900,000	4359
Static	6060	8,000	78,800	0.23	4.60×10^{-3}	74.0	2,410,000	4050
Load	6060	10,000	98,600	0.12	5.20×10^{-3}	136.0	3,180,000	4280

TABLE III

STIFFNESS AND COMPRESSOR STRESS OF THE BEARING PIVOTS

Generator Design Number	Maximum Bearing Load W. lbs.	Diameter of Curva- ture, Ball Seat, d_2 , in.	Diameter of Curva- ture Socket Seat, d_1 , in.	Total Bearing Pivot Stiffness K_{pivot} , lbs/in.	Maximum Hertzian (Compressive) Stress in Pivot s, psi
10526	93	0.610	0.622	1,350,000	124,000
10826	96	1.170	1.192	1,600,000	76,000
10827	107	1.410	1.439	1,750,000	78,000
20528	322	1.700	1.734	2,800,000	79,000
20827	397	2.340	2.387	3,100,000	69,000
20826	449	2.740	2.795	3,400,000	71,000
30829	1922	4.300	4.386	6,500,000	85,000
40827	6060	6.250	6.375	10,800,000	98,000

TABLE IV
SUPPORT DISK CHARACTERISTICS

Generator Design No.	Disk O.D. D_D , in.	Disk I.D. D_C , in.	Disk Thickness t , in.	Radial Stiffness of Disk K_{disk} , lbs/in.	1st Critical Speed of Disk - Brg.Housing Assembly(axial mode) $N_{c,disk}$ - C.P.M.
10526	18.2	3.00	0.27	6×10^6	40,500
10826	18.3	3.60	0.32	9×10^6	40,800
10827	18.9	4.00	0.30	10×10^6	38,500
20528	25.5	5.60	0.51	17×10^6	32,500
20827	26.6	7.00	0.54	24×10^6	26,500
20826	27.4	8.00	0.54	29×10^6	22,600
30829	45.2	14.00	1.08	64×10^6	16,800
40827	60.3	18.00	1.49	83×10^6	11,000

Fig. I Dynamic Model of Bearing and Support System

- Notes:
1. Bearing centerlines at stations 2 and 11
 2. Pole centerlines at stations 5 and 8
 3. Dynamic balance correction planes at stations 4 and 9
 4. Rotor lengths between stations as shown in Figures IVA through IVH for each rotor.

Fig. II Schematic of Rotor Stations in Critical Speed and Unbalance Response Calculations

Figure III A Undamped System Critical Speed Map

Figure III B Undamped System Critical Speed Map

Figure III C Undamped System Critical Speed Map

Figure IV A Unbalance Response at Design Speed

Figure IV B Unbalance Response at Design Speed

Figure IV C Unbalance Response at Design Speed

Figure V A Dynamic Orbit in Bearings vs Speed, for "In-Phase" and "Out-of-Phase" Locations of the Mass Unbalances

Figure V B Dynamic Orbit in Bearings vs Speed, for "In-Phase" and "Out-of-Phase" Locations of the Mass Unbalances

Figure V C Dynamic Orbit in Bearings vs Speed, for "In-Phase" and "Out-of-Phase" Locations of the Mass Unbalances

Figure VI B Dynamic Orbit in Bearing at Design Speed vs Fluid Film Stiffness, for "Out-of-Phase" Location of the Mass Unbalances

APPENDIX A - PIVOT AND HOUSING SUPPORT DISK

1. Pivot Stiffness and Maximum Compressive Stress

In order to calculate the pivot stiffness of a pivoted pad bearing, the loads carried by each of the individual pads must be determined. For a preloaded bearing, all pads, including the ones that are located in the upper half of the bearing, i.e. above the load line from a convergent wedge and generate positive film pressures.

For a four pad bearing, where the net bearing load is directed midway between the pivots of the two pads in the lower half of the bearing, we have:

$$W_{brg} = 2 (P_L - P_U) \cos 45 \quad (A-1)$$

where W_{brg} = net bearing load, lbs.

P_L = load carried by each of the two lower pads

P_U = load carried by each of the two upper pads

P_L and P_U , for any given bearing and set of operation conditions, including the net load (W_{brg}) carried by the bearing, have to be determined from calculation of the individual pad eccentricity ratios. However, for a four pad bearing, with a preload coefficient of 0.3 and an effective eccentricity ratio of about 0.4 to 0.5, the analysis of the individual bearing pads shows that the load on each of the upper pads is about 1/3 that on each of the lower pads, i.e.

$$P_L \approx 3 P_U \quad (A-2)$$

For this particular operating condition, which corresponds to that of the generator bearings at design speed, the individual pad loads may be obtained from Equations A-1 and A-2, for the given value of net bearing load, W_{brg} .

The pivot stiffnesses for each individual pads can then be calculated from the well known relations for Hertzian contact deformations. For the case of a spherical ball in a cylindrical socket, this relation is given, for example,

in Table XIV, Condition 8 on Page 289 of Ref. 5. From this reference, the radial elastic deflection at the contact is given by:

$$y = \lambda (P^2/k^2 \delta)^{1/3} \quad (A-3)$$

where: y = radial deflection in inches

λ is a tabulated function in the above noted reference

P = contact load in lbs (i.e. in our case, the individual pad loads P_L and P_U for the lower and upper bearing pads respectively).

$$k = \frac{4E}{3(1-\nu^2)}$$

where E = Young's modulus ($\approx 24 \times 10^6$ for tool steels at 800F)

and ν = Poisson's ratio = 0.3

and δ , for the case of a spherical ball in a cylindrical socket, $= \frac{4}{\frac{2}{R_2} - \frac{1}{R_1}}$

where R_2 is the radius of curvature of the spherical ball and R_1 is the radius of curvature of the cylindrical socket.

By differentiation of Equation A-3, we obtain the pivot stiffness as.

$$K_{\text{pivot}} = \frac{dP}{dy} = \frac{3(k^2 P \delta)^{1/3}}{2\lambda} \quad (A-4)$$

From Equation A-4 and making use of Equations A-1 and A-2, the pivot stiffness for each pad may be computed. In the design used here, the socket radius of curvature was chosen to be 2% larger than that of the ball seat to minimize sliding motions in the pivot, i.e.

$$R_1 = 1.02 R_2 \quad (A-5)$$

For a four pad bearing, it can also be shown that the total pivot stiffness in the bearing is the sum of the pivot stiffnesses of one of the lower pads and one of the upper pads, i.e.

$$(K_{\text{pivot}})_{\text{total}} = (K_{\text{pivot}})_{\text{lower pad}} + (K_{\text{pivot}})_{\text{upper pad}} \quad (\text{A-6})$$

where the pivot stiffnesses of the lower and upper pads are obtained as described above, for loads P_L and P_U respectively.

The calculated total bearing pivot stiffnesses were thus obtained and are given in Table III for each generator bearing. The bearing loads used in these calculations are the maximum static loads on the bearings.

It is important to note here that, as shown by Equation A-4, the pivot stiffness varies as the 1/3 power of the load. Thus, the pivot stiffness changes are moderately small for wide variations of the bearing load.

The pivots must also be designed for a conservatively small value of the maximum compressive stress for good long term operation. The maximum compressive stress is also given in Table XIV, Condition 8 of Ref. 5, as:

$$s_{\text{max}} = \frac{1.5 P}{\pi c d}$$

where c and d are the radii of the contact zone ellipse and are given in the same reference as:

$$c = \alpha \left(\frac{P\delta}{k} \right)^{1/3}$$

$$\text{and } d = \beta \left(\frac{P\delta}{k} \right)^{1/3}$$

where α and β are tabulated functions in the above cited reference, and δ and k are as defined above in the discussion of pivot stiffness.

The pivot dimensions and maximum compressive stress are given in Table III. These were obtained for tool steel with a modulus of elasticity of 24×10^6 at 800F and maintaining the maximum compressive stress at about 120,000 psi or less. For good long term pivot operation, both pivot elements must be hardened to about RC 60.

2. Radial Stiffness of the Bearing Housing Support.

Where the bearing centerline through which the force "P" acts is at an eccentricity a , relative to the centerline of the support disk as shown in the accompanying sketch, the bearing support stiffness has two components, in series:

- the component due to radial deflection of the disk under a load "P" and,
- the component due to angular twist of the disk, under a moment "Pa".

An upper limit of the deflection of the disk diaphragm under a radial load P at its centerline may be obtained by treating the disk as a four spoke element, with two cantiliver spokes at 90° to the load line, one spoke in compression under the load line and one spoke in tension above the load line.

The radial deflection of the disk is then:

$$y = \frac{P}{2 \left[\frac{EA}{k} + \frac{3EI}{k^3} \right]} \quad (A-7)$$

where E = Young's modulus, psi

$$A = D_c t, \text{ in}^2$$

$$k = \frac{D_D^2 - D_c^2}{2} \text{ in.}$$

$$I = \frac{t D_c^3}{12}, \text{ in}^4$$

D_D = outside diameter of disk, in.

D_c = inside diameter of disk, in

t = disk thickness, in.

The lower limit of the radial stiffness of the disk is then,

$$K_{\text{disk}} = \frac{dP}{dy} = \frac{4ED_c t}{(D_D - D_c)} \left[1 + \left(\frac{D_c}{D_D - D_c} \right)^2 \right], \quad \text{lbs.} \quad (\text{A-8})$$

The component of bearing support stiffness due to angular twist of the disk under the moment "Pa", assuming the disk to be "built-in" at its OD and ID is obtained from Chapter 10, Pages 197 and 216 of Ref. 5.

The angular twist of a circular disk with built in ID and OD, under a moment M is given in Ref. 5. as:

$$\theta = \frac{M}{\alpha E t^3}$$

where M is the moment, in.lbs.

α is a tabulated function of the I.D. to O.D. ratio

E is Young's Modulus, psi

and t is the disk thickness, in.

The radial displacement at the point of application of the load P as shown in the above sketch is then:

$$y = a\theta = \frac{a M}{\alpha E t^3} = \frac{P a^2}{\alpha E t^3}$$

Thus, the stiffness component due to eccentric loading of the support disk is then:

$$(K)_{\text{twist}} = \frac{dP}{dy} = \frac{\alpha E t^3}{a^2} \quad (\text{A-9})$$

Since the two stiffness components are in series, the overall bearing support stiffness is given by:

$$\frac{1}{K_{\text{support}}} = \frac{1}{K_{\text{disk}}} + \frac{1}{K_{\text{twist}}} \quad (\text{A-10})$$

Actually in the design of the bearing supports it was found that the disk could in all cases be located at the center plane of the housing, so that:

$$K_{\text{support}} = K_{\text{disk}}$$

This is desirable, as it provides a high support stiffness.

The disk thickness "t" was established by insuring that its lowest resonant frequency is at least 35% above the running speed. This lowest resonant frequency is the first disk axial critical speed.

From Ref. 5, Table X, Case 20, the axial stiffness of the disk is given by:

$$\frac{1}{K_a} = \frac{dy}{dP} = \frac{3(m^2-1)}{4\pi^2 Et^3} \left[a^2 - b^2 - \frac{4a^2 b^2}{a^2 - b^2} \left(\ln \frac{a}{b} \right)^2 \right] \quad (\text{A-11})$$

where K_a = axial disk stiffness, lbs/in.

y = axial deflection, in.

P = axial force, lbs.

m = $1/\nu$, where ν = Poisson's ratio

E = Young's Modulus, psi

t = disk thickness, in.

a = outer radius of disk, in.

b = inner radius of disk, in.

The first, axial mode resonant frequency is then given by

$$60 N = \frac{30}{\pi} \sqrt{\frac{K_a}{M_{\text{EFF}}}} \quad (\text{A-12})$$

where M_{EFF} is the mass of the bearing housing.

The O.D. and I.D. of the bearing housing, as well as the disk thickness, are given in Table V. The radial stiffness of the disk calculated from Equation A-8 and the first resonant frequency of the disk calculated from Equations A-11 and A-12 are also given in Table IV.

APPENDIX B - APPROXIMATE CALCULATIONS OF THE FIRST AND SECOND UNDAMPED CRITICAL SPEEDS

The undamped system critical speed maps given in Figs. IIA through IIH were obtained using the procedure given in Ref. 2, programmed on a digital computer. The first two system critical speeds can, however, be approximately obtained as follows:

The approximate value of the first critical speed is given by:

$$60N_1 = \frac{30}{\pi} \sqrt{\frac{K_{TOT}}{M}}$$

$$\text{where } \frac{1}{K_{TOT}} = \frac{1}{K_{Disk}} + \frac{1}{K_{Pivot}} + \frac{1}{K_{Film}} + \frac{1}{K_{Stub, Shaft}}$$

$$\text{and } M = \frac{1}{2} \text{ Rotor Mass.}$$

The fluid film, pivot and disk stiffnesses are given in Tables II, III and IV.

The stub shaft stiffness is given by:

$$K_{stub} = \frac{3 E I}{(SDL + L/2)^3}$$

where SDL and L are given in Table V, E is the Young's modulus for the stub shaft and $I = \frac{\pi}{64} (D^4 - D_i^4)$. D and D_i are also given in Table V.

Correction can be approximately made for the effect of the magnetic stiffness by subtracting it from the fluid film stiffness.

The approximate value of the second critical speed is given by:

$$60N_2 = \frac{30}{\pi} \sqrt{\frac{K_c}{I_R}}$$

$$\text{where: } I_R = \frac{M_{rotor}}{12} (3r^2 + l_1^2)$$

Here, r is the approximate effective diameter of the rotor treated as a solid cylinder of length ℓ_1 . (The actual second critical speed should be lower than predicted by this method due to the added inertia of the stub shafts).

$$\text{and } K_c = K_{TOT} \frac{\ell_2^2}{2}$$

where K_{TOT} is as defined above in the equation for the first critical speed and ℓ_2 is the span between bearings.

APPENDIX C - DESIGN TABLES AND PROCEDURE FOR TILTING PAD BEARINGS

The design tables for 4 and 6 pad, tilting pad bearings operating in laminar or turbulent flow regimes are given on Pages C 13 through C 119 of this Appendix. Before proceeding to these tables, it is necessary first to briefly review the tilting pad bearing geometry, the turbulent flow analysis and the bearing parameters given in the tables. A simple calculation, illustrating the use of the tables is also given.

1. The Geometry of the Tilting Pad Bearing

The tilting-pad journal bearing consists of a set of pads which are individually pivoted, as shown schematically in Fig. C-1. The geometrical parameters that define the bearings are:

- (a) Bearing Diameter (D)
- (b) Number of pads
- (c) Angular extent of the pads (β)
- (d) Slenderness ratio ($\frac{L}{D}$)
- (e) Pad clearance ratio ($\frac{C_P}{R}$)
- (f) Pivot location ($\frac{\theta_P}{\beta}$)
- (g) Geometrical preload coefficient (m)

Each bearing pad is individually pivoted, generally by means of a spherical ball seat. In this manner, the pads can track orbital motions of the shaft, in response to the changes in the fluid film forces generated in the film between each pad and the shaft, by the orbital motions of the shaft. It is this characteristic that accounts for the high stability of tilting pad bearings. Furthermore, the pivoted pads can incline to achieve parallelism with the shaft. This inherent self aligning capability is particularly important in turbomachinery applications.

The location of the bearing pivot point, measured in the circumferential direction

is, given by the angle θ_p , measured from the leading edge of the pad as illustrated in Fig. C-1. (In the axial direction the pivot is, of course, located in the mid plane of the pad). When the shaft may have to rotate in either clockwise or counter clockwise directions, it is generally desirable to locate the pivot at the mid point of the pad, i.e. at $\theta_p/\beta = 0.5$. However, in applications where the shaft rotation is unidirectional, as in the case of the alternator rotors of interest here, some improvement in bearing performance characteristics (load capacity, fluid film stiffness, fluid film damping and range of equilibrium positions of the pads) can be obtained by locating the pivot further downstream than the mid length of the pad. The design data given in Tables C1 and C2 has been obtained for a pivot position $\theta_p/\beta = 0.55$.

2. Geometrical Preloading of Tilting Pad Bearings

Geometrical preloading is used in order to achieve:

- (a) high bearing stiffness, even with zero net load on the shaft
- (b) positive fluid film pressures acting on all the bearing pads, even those located in the part of the bearing away from the direction of load. This serves to prevent pad flutter.

Geometrical preloading is achieved by assembling the pads such that their centers of curvature, with zero inclinations, form a circle of radius "a" about the center of the bearing. Figures C-2a and C-2b show, respectively, the geometrical arrangement of the pads without and with geometrical preload. Let:

$$\begin{aligned} R_p &= \text{Radius of curvature of the pads as machined} \\ R_s &= \text{Radius of shaft as machined} \\ a &= \text{Radius of preload circle} \\ C_p = R_p - R_s &= \text{Radial clearance of the pads (machined clearance)} \end{aligned}$$

In the preload case, "the bearing radius" is now defined as:

$$R_B = R_p - a$$

The "bearing radial clearance" (i.e. running clearance over the pivots at zero load) is defined as:

$$C_B = R_B - R_S = R_P - a - R_S$$

From the definition of C_P and the equation above, it is seen that

$$a = C_P - C_B$$

The preload coefficient is defined as:

$$m = \frac{a}{C_P} = 1 - \frac{C_B}{C_P}$$

For bearings lubricated with a low viscosity fluid, such as liquid potassium, it is desirable to operate the bearings with some preload, in order to achieve high stiffness and damping even in the concentric position and to insure that all pads are positively loaded.

3. Direction of the Steady State Load in a Tilting Pad Bearing

The pads of tilting pad bearings are generally assembled in one of two ways, relative to the direction of the steady state load vector, as illustrated in Figs. C-3a and C-3b.

In the case illustrated in Fig. C-3a, the pads are assembled so that the steady state load vector W passes midway between the pivots on two adjacent pads. In the case illustrated in Fig. C-3b, the pads are assembled so that the steady state load vector passes through one of the pivots.

In general, the load arrangement illustrated in Fig. C-3a (load direction midway between pivots) provides the larger value of pivot point film thickness for any given load, than does the case illustrated in Fig. C-3b (load through pivot). Furthermore, at least in the case of four pad bearings, the fluid film stiffness in the case where the steady state load vector passes midway between pivots, is equal in the x and y directions, where the x axis is taken to be along the steady state load line and the y axis is normal to it.

The tables given in this Appendix cover both load directions.

4. Effective Eccentricity Ratio of a Tilting Pad Bearing

The eccentricity ratio for pivoted pad bearings can be expressed in several ways.

Thus, defining $(e_p)_i$ as the distance between the shaft center and the center of curvature of the i 'th pad, the "pad eccentricity ratio" is defined as:

$$(\epsilon_p)_i = \frac{(e_p)_i}{C_p}$$

Defining e_B as the distance between the shaft center and the bearing center (i.e. the center of the circle of radius "a" illustrated in Fig. C-2(b)), we can define a "bearing eccentricity ratio" as follows:

$$\epsilon_B = \frac{e_B}{C_B}$$

It should be noted that the most critical film thickness in a tilting pad bearing is that directly under the pivot point. This film thickness, which is referred to as the "pivot point film thickness", is the only one in each pad, which is not affected by inclinations of the pad. Furthermore, as the load on the shaft is increased until, in the limit the shaft comes in contact with one or more pads, the point of contact will be directly under the pivot point. Thus, in the concentric (zero net bearing load) condition, the pivot point film thickness " h_p " is equal to the radial bearing clearance " C_B ". At failure, when the load is high enough to induce contact between the shaft and a pad, the pivot point film thickness is zero. The "effective eccentricity ratio" of the bearing, can therefore be defined as:

$$\epsilon_{EFF} = 1 - \frac{(h_p)_{min}}{C_B} = 1 - \frac{(h_p)_{min}}{C_p(1-m)}$$

where $(h_p)_{min}$ is the pivot point film thickness of the most heavily loaded pad or pads.

In the concentric position $\epsilon_{EFF} = 0$ and at failure, $\epsilon_{EFF} = 1$.

5. Laminar and Turbulent Flows in Bearing Films

Oil lubricated bearings generally operate in the laminar flow regime because of

the high viscosity of most hydrocarbon oil lubricants. Liquid metals, on the other hand, have relatively low viscosities so that most high speed, liquid metal lubricated bearing films are turbulent. Accordingly, Tables C-1 and C-2 below provide the design characteristics of tilting pad bearings for both laminar and turbulent flow regimes .

In the past several years, there has been considerable analytical and experimental investigation of lubrication in the super-laminar flow regimes (i.e. above the point of transition for laminar flow). Thus, the vortex flow regime is treated in Ref. 8. The general and the linearized treatments of the turbulent flow regime are given in Refs. 9 and 10 respectively. Currently, turbulent flow lubrication theory is being extended to include fluid inertia effects; Ref. 7 gives a summary review of these various regimes of super-laminar flow lubrication.

The design data in Tables C-1 and C-2 is based on the linearized turbulent flow lubrication theory described in Ref.9. This theory has been extensively verified in tests conducted with various self acting bearing types. These experimental investigations and the generally good agreement obtained with the theory are described in Ref.11 for full circular and partial arc bearings, in Ref.12 for tilting pad bearings and in Ref.13 for floating ring bearings.

In the analysis given in Ref. 9 , the incompressible, turbulent flow Reynolds Equation is obtained in the form:

$$\frac{1}{R^2} \frac{\partial}{\partial \theta} \left(\frac{h^3 G_\theta}{\mu} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left(\frac{h^3 G_z}{\mu} \frac{\partial p}{\partial z} \right) = \frac{\omega}{2} \frac{\partial h}{\partial \theta} + \frac{\partial h}{\partial t}$$

where G_θ and G_z are functions of the local Reynolds number $\frac{R\omega h}{\nu}$. In the limit, as the Reynolds number approaches zero, G_θ and G_z approach the value 1/12, and the above equation reduces to the conventional, laminar flow Reynolds Equation.

The characteristics of turbulent flow bearings (i.e. load capacity, fluid film friction, hydrodynamic flow, fluid film stiffness and damping) are obtained from integrations of the above equation, conducted numerically by finite differences and using the values of G_θ and G_z at each point in the film, calculated as shown in Ref. 9.

It should be noted that in order to calculate the local value of the Reynolds number (and hence G_θ and G_z) at each point in the film, it is necessary only to define a mean value of the Reynolds number ($Re_{mean} = \frac{R\omega C_p}{\nu}$) as well as the eccentricity ratio and attitude angle (ϵ_p and ϕ_p respectively) for each bearing pad since:

$$\begin{aligned} \text{Local Reynold Number} &= \frac{R\omega h}{\nu} = \frac{R\omega C_p}{\nu} [1 - \epsilon_p \cos (\theta - \phi_p)] \\ &= Re_{mean} [1 - \epsilon_p \cos (\theta - \phi_p)] \end{aligned}$$

Thus, the turbulent bearing data in Tables C-1 and C-2 are given in terms of the mean Reynolds number, $R\omega C_p/\nu$. As noted above, however, the data in the tables has been obtained, taking account of the local values of the Reynolds number at each point in the film.

The point of transition from laminar to super-laminar flow in a bearing film will depend on the clearance ratio in the bearing and the bearing eccentricity, as well as on the mean Reynolds number, as discussed in Refs. 7 and 8. For design purposes, however, and assuming conventionally small values of the pad to shaft clearance ratio, $C_p/R \leq 0.002$ inches/inch, the laminar flow design data should be used for values of mean Reynolds number up to 1000. If the mean Reynolds number is greater than 1000, the turbulent flow data should be used. This is given in Tables C-1 and C-2 for various values of the mean Reynolds number, up to 30,000.

6. Design Data for Tilting Pad Bearings in Laminar and Turbulent Flow Regimes

The data in Tables C-1 and C-2 covers a wide range of design parameters, as follows:

Bearing type: tilting pad

Number of pads per bearing: 4 pads and 6 pads

Pad included angle: $\beta = 80^\circ$ pads for the four pad bearing

$\beta = 50^\circ$ pads for the six pad bearing

Pivot position: $\theta_p/\beta = 0.55$

Slenderness ratio: $L/D = 0.5$ and 1.0

Geometrical preload factor: $m = 0.3$ and 0.5

Laminar Flow

Turbulent flow at: $Re_{\text{mean}} = 1000, 10,000, 20,000 \text{ and } 30,000$

Direction of the steady-state bearing load: (a) midway between the pivots of a pair of adjacent pads and, (b) through the pivot of one of the pads.

Table C-1 gives the design data for four pad bearings and Table C 2 gives the design data for six pad bearings. Each page of these two tables gives two tabulations of the normalized (i.e. non-dimensional values of) the bearing performance characteristics as functions of the effective bearing eccentricity ratio. The lower tabulation in each page is for the case where the steady-state load is directed midway between pivots and the upper tabulation is for the case where the steady-state load is directed through a pivot. The conditions (number of pads, pad included angle, pivot position, slenderness ratio, geometrical preload factor and laminar or turbulent flow, and the value of mean Reynolds number in the latter case) are all defined at the top of each page.

Following is the definition of all the normalized characteristics given in these tables:

Reynolds Number

This is the mean Reynolds number: $Re_{\text{mean}} = \frac{\pi D N C_P^*}{\nu}$ (1)

Preload

This is the geometrical preload factor: $m = 1 - \frac{C_B}{C_P}$ (2)

Eff. Ecc. Ratio

This is the effective eccentricity ratio of the bearing:

$$\epsilon_{\text{EFF.}} = 1 - \frac{(h_p)_{\text{min}}}{C_B} \quad (3)$$

Norm. Load

This is the normalized value of the steady-state bearing load:

$$\bar{W} = \frac{W}{\mu N L D} \left(\frac{C_P}{R} \right)^2 \quad (4)$$

* See Page C-12a for Tilting Pad Bearing Nomenclature

Norm. Friction

This is the normalized value of the friction torque of the bearing:

$$\bar{T} = \frac{4C_p T}{\mu NLD^3} \quad (5)$$

The power loss in the bearing films, expressed in horsepower, as

$$(HP)_{BRG} = \frac{2\pi NT}{6600} \quad (5a)$$

Norm. Hyd. Flow

This is the normalized value of the flow introduced into the fluid films between the shaft and the pads, by the shaft rotation:

$$\bar{Q}_{HYD} = \frac{Q_{HYD}}{NDLC_p} \quad (6)$$

(Note: The hydrodynamic bearing flow is, as defined above, the flow that actually circulates through the thin fluid film between the shaft and the pads. It is not necessarily equal to the flow that must be circulated through the bearing cavity between the seals at each end of the bearing to cool the bearing region and take out the heat generated by fluid film friction. This latter flow may be larger, or smaller than the hydrodynamic flow depending on several design factors such as (a) whether or not the bearing cavity is flooded, (b) the heat generated in the bearing and its seals, (c) the heat introduced into or removed from the bearing region by conduction and, (d) the allowable temperature rise in the flow through the bearing cavity. Assuming a flooded bearing cavity, the flow that must be circulated through the bearing cavity for cooling is obtained from the heat balance equation:

$$q \gamma c_p \Delta t = 0.707 (HP)_{total} \quad (6a)$$

where: q is the lubricant flow circulated through the bearing cavity, in^3/sec

γ is the density of the lubricant, lbs/in^3

c_p is the specific heat of the lubricant, $\text{BTU}/(\text{lb} \times ^\circ\text{F})$

Δt is the allowable temperature rise of the lubricant between outlet from and inlet to the bearing cavity, $^\circ\text{F}$.

$(HP)_{total}$ is the sum of: (a) the heat generated in the bearing film, (b) the heat generated in the seals and, (c) the heat introduced into the bearing region by conduction, all expressed in horsepower.

The hydrodynamic bearing flow, Q_{HYD} , obtained from expression (6) above is, in the case of a flooded bearing cavity, largely a recirculating flow generated by the shaft rotation. It serves principally to allow an estimate to be made of the maximum temperature in the film. This maximum film temperature can be approximately estimated by the following relation:

$$t_{max} - t_{in} = \frac{0.707(HP)_{BRG}}{\gamma c_p Q_{HYD}} \quad (6b)$$

where t_{max} is the maximum temperature in the fluid film, $^{\circ}F$
 t_{in} is the mean temperature in the bearing cavity, $^{\circ}F$
 and $(HP)_{BRG}$ is the heat generated in the bearing film, expressed in HP)

Norm. Stiff (X)

This is the normalized value of the fluid film stiffness along the radial axis, parallel to the steady state load line (see Fig. C3):

$$\bar{K}_x = \frac{K_x}{2\mu NL} \left(\frac{C_P}{R}\right)^3 \quad (7)$$

Norm. Stiff (Y)

This is the normalized value of the fluid film stiffness along the radial axis normal to the steady state load line (see Fig. C3):

$$\bar{K}_y = \frac{K_y}{2\mu NL} \left(\frac{C_P}{R}\right)^3 \quad (8)$$

Norm. Damp. (X)

This is the normalized value of the fluid film damping along the radial axis parallel to the steady state load line (see Fig. C3):

$$\bar{B}_x = \frac{2\pi NB_x}{K_x} \quad (9)$$

Norm. Damp. (Y)

This is the normalized value of the fluid film damping along the radial axis normal to the steady state load line (see Fig. C3):

$$\bar{B}_y = \frac{2\pi NB_y}{K_y} \quad (10)$$

7. Sample Calculation

Tables C1 and C2 are given on Pages C 13 through C 54. Following is an example illustrating the use of the tables. The data in this example is for the Westinghouse Alternator Rotor, Design No. 10827.

Given: Steady state load per bearing 67.1 lbs. (This is the sum of the rotor weight and the steady state electrical loading, in each bearing).

Bearing diameter (D) = 2"

Bearing length (L) = 1"

Bearing clearance ratio ($C_{B/R}$) = 0.0013 in/in

Lubricant = Potassium at 800F, Viscosity (μ) = 2.76×10^{-8} lbsec/in²

Mass Density (ρ) = 0.69×10^{-4} lbsec²/in⁴

Design speed (N) = 333 R.P.S.

Use a four pad bearing, with the steady state load line passing midway between the pivots of an adjacent pair of pads.

Calculate: Effective eccentricity ratio (ϵ_{EFF})

Minimum pivot point film thickness (h_{pmin})

Bearing power loss (HP)_{BRG}

Fluid film stiffness and damping coefficients (K_x , K_y , B_x and B_y).

Procedure:

Radial Bearing Clearance: $C_B = \left(\frac{C_B}{R}\right) \cdot R = (0.0013)(1) = 0.0013$ inches

Pad to Shaft Clearance: $C_P = \frac{(C_B)}{(1 - m)} = \frac{(0.0013)}{(1 - 0.3)} = 0.00186$ inches

Kinematic Viscosity: $\nu = \frac{\mu}{\rho} = \frac{2.76 \times 10^{-8}}{0.69 \times 10^{-4}} = 4.0 \times 10^{-4}$ in²/sec

Mean Reynolds Number: $Re_{mean} = \frac{\pi D N C_P}{\nu}$
 $= \frac{(\pi)(2)(333)(1.86 \times 10^{-3})}{(4.0 \times 10^{-4})}$
 $= 9740$

$$\begin{aligned}\text{Normalized Bearing Load: } \bar{W} &= \frac{W}{\mu NLD} \left(\frac{C_P}{R}\right)^2 \\ &= \frac{(67.1)(0.00186)^2}{(2.76 \times 10^{-8})(333)(1)(2)} = 12.6\end{aligned}$$

Entering Table C1 (for four pad bearings), with $m = 0.3$, load between pivots and interpolating between $Re_{\text{mean}} = 1000$ and $10,000$ to obtain the data for $Re_{\text{mean}} = 9740$, we obtain for $\bar{W} = 12.6$:

$$\begin{aligned}\epsilon_{\text{EFF}} &= 0.37 \\ \bar{T} &= 161 \\ \bar{K}_x &= \bar{K}_y = 39 \\ \bar{B}_x &= \bar{B}_y = 1.22\end{aligned}$$

Hence, from the relations 3, 5, 5(a), 7, 8, 9 and 10 given earlier in this Appendix we obtain:

$$\text{Effective bearing eccentricity ratio: } \epsilon_{\text{EFF}} = 0.37$$

$$\begin{aligned}\text{Minimum pivot point film thickness: } (h_p)_{\text{min}} &= C_B(1 - \epsilon_{\text{EFF}}) \\ &= (0.0013)(1 - 0.37) \\ &= 0.82 \times 10^{-3} \text{ inches}\end{aligned}$$

$$\begin{aligned}\text{Bearing Friction Torque: } T &= \frac{\mu NLD^3 \bar{T}}{4C_P} \\ &= \frac{(2.76 \times 10^{-8})(333)(1)(2)^3(161)}{(4)(1.86 \times 10^{-3})} \\ &= 1.59 \text{ inlbs}\end{aligned}$$

$$\begin{aligned}\text{Bearing Power Loss: } (\text{HP})_{\text{BRG}} &= \frac{2\pi NT}{6600} \\ &= \frac{(2\pi)(333)(1.59)}{(6600)} = 0.5 \text{ HP}\end{aligned}$$

Fluid Film Stiffnesses along the radial axis parallel to the steady

$$\begin{aligned}\text{state load line: } K_x &= \frac{2\mu NL\bar{K}_x}{(C_P/R)^3} \\ &= \frac{(2)(2.76 \times 10^{-8})(333)(1)(39)}{(1.86 \times 10^{-3})^3} \\ &= 112,000 \text{ lbs/inch}\end{aligned}$$

Since we are concerned here with a four pad bearing, where the steady state load is midway between pivots, $\bar{K}_y = \bar{K}_x$ and, hence,

Fluid film stiffness along the radial axis normal to the steady state load line: $K_y = K_x = 112,000$ lbs/inch.

Fluid film damping along the radial axis parallel to the steady state load line:

$$\begin{aligned} B_x &= \frac{K_x \bar{B}_x}{2\pi N} \\ &= \frac{(112,000)(1.22)}{(2\pi)(333)} \\ &= 65 \text{ lb. sec/in.} \end{aligned}$$

Again, since we are concerned with a four pad bearing where the steady state load line passes midway between pivots, $\bar{B}_y = \bar{B}_x$ and, hence,

Fluid film damping along the radial axis normal to the steady state load line: $B_y = B_x = 65$ lb.sec/in.

8. NOMENCLATURE FOR TILTING PAD BEARINGS

- a = radius of preload circle
 B_x = Bearing damping coefficient along the load axis, lbs.sec/in
 B_y = Bearing damping coefficient along the axis normal to the load line, lbs.sec/in
 C_B = Bearing radial clearance, inches
 C_p = Machined pad to shaft clearance, inches
 c_p = Specific heat, BTU/lb $^{\circ}$ F
 D = Bearing diameter, inches
 e_B = Bearing eccentricity (distance between shaft center and bearing center,) in.
 e_p = Pad eccentricity (distance between shaft center and center of curvature of pad), inches
 G_{θ}, G_z = Turbulence coefficients, functions of the local Reynolds Number.
 h = Film thickness, inches
 h_p = film thickness at a pivot point, inches
 $(h_p)_{\min}$ = Minimum pivot point thickness (i.e. film thickness at pivot of most heavily loaded pad), inches
 HP = Power loss, horsepower
 $(HP)_{\text{brg}}$ = Bearing power loss, horsepower
 K_x = Bearing stiffness coefficient, along the load axis, lbs/in.
 K_y = Bearing stiffness coefficient, along the axis normal to the load line, lbs/in.
 L = Bearing length, in.
 m = Geometrical preload coefficient
 N = Rotor speed, revolutions per second
 p = Fluid film pressure, psi
 Q = Flow, in 3 /sec.
 Q_{hyd} = Hydrodynamic bearing flow, in 3 /sec.
 q = Lubricant supply rate to bearing, in 3 /sec.

Nomenclature for Tilting Pad Bearings (cont'd)

- R = Bearing radius, in.
 R_p = Radius of curvature of a bearing pad, in.
 R_s = Shaft radius, in.
 Re = Local Reynolds Number (see Page C-5)
 Re_{mean} = Mean Reynolds Number (See Eq. 1, Page C-7)
 T = Bearing friction torque, in.lbs.
 t = Temperature, $^{\circ}F$
 t_{in} = Lubricant temperature at pad inlet, $^{\circ}F$
 t_{max} = Maximum film temperature, $^{\circ}F$
 W = Bearing load, lbs
 z = Axial coordinate, in.
 β = Pad included angle, degrees
 γ = Weight density, lbs/in^3
 (Δt) = Temperature rise, $^{\circ}F$
 ϵ_B = Bearing eccentricity ratio (See Page C-4)
 ϵ_{EFF} = Effective bearing eccentricity ratio (See Eq. 3, Page C-7)
 ϵ_p = Pad eccentricity ratio (See Page C-4)
 θ = Circumferential coordinate, radians
 θ_p = Angle from leading edge of pad to pivot point, degrees
 μ = Absolute viscosity, $lb.sec./in^2$
 ν = Kinematic viscosity, $in^2/sec.$
 ρ = Mass density, $lb.sec^2/in^4$
 ϕ_p = Pad attitude angle, radians
 ω = Angular velocity, radians/second

Nomenclature for Tilting Pad Bearings (cont'd)

Superscripts

Superscript bar indicates normalized value of the parameter

The normalized bearing parameters are defined in Equations 1 through 10,
on Pages C-7 through C-9.

TABLE C-1

DESIGN DATA FOR THE FOUR PAD, TILTING PAD BEARING

Table C-1 (cont'd)

C-14

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
LAMINAR FILM

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.252E 02	0.566E 01	0.639E 01	0.639E 01	0.142E 01	0.142E 01
0.100	0.578E 00	0.253E 02	0.566E 01	0.682E 01	0.639E 01	0.138E 01	0.142E 01
0.200	0.123E 01	0.257E 02	0.565E 01	0.829E 01	0.639E 01	0.128E 01	0.142E 01
0.300	0.207E 01	0.264E 02	0.563E 01	0.114E 02	0.639E 01	0.113E 01	0.142E 01
0.400	0.327E 01	0.275E 02	0.560E 01	0.174E 02	0.639E 01	0.987E 00	0.142E 01
0.500	0.515E 01	0.293E 02	0.554E 01	0.293E 02	0.639E 01	0.840E 00	0.142E 01
0.600	0.853E 01	0.321E 02	0.543E 01	0.560E 02	0.639E 01	0.682E 00	0.142E 01
0.700	0.153E 02	0.374E 02	0.531E 01	0.128E 03	0.639E 01	0.526E 00	0.142E 01
0.800	0.320E 02	0.455E 02	0.518E 01	0.372E 03	0.639E 01	0.450E 00	0.142E 01
0.850	0.519E 02	0.571E 02	0.511E 01	0.774E 03	0.639E 01	0.404E 00	0.142E 01
0.900	0.979E 02	0.775E 02	0.504E 01	0.207E 04	0.639E 01	0.340E 00	0.142E 01
0.928	0.160E 03	0.942E 02	0.499E 01	0.447E 04	0.639E 01	0.296E 00	0.142E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.252E 02	0.566E 01	0.639E 01	0.639E 01	0.142E 01	0.142E 01
0.100	0.817E 00	0.254E 02	0.566E 01	0.682E 01	0.682E 01	0.138E 01	0.138E 01
0.200	0.174E 01	0.262E 02	0.563E 01	0.829E 01	0.829E 01	0.128E 01	0.128E 01
0.300	0.293E 01	0.276E 02	0.559E 01	0.114E 02	0.114E 02	0.113E 01	0.113E 01
0.400	0.462E 01	0.299E 02	0.553E 01	0.174E 02	0.174E 02	0.987E 00	0.987E 00
0.500	0.728E 01	0.334E 02	0.542E 01	0.293E 02	0.293E 02	0.840E 00	0.840E 00
0.600	0.120E 02	0.390E 02	0.519E 01	0.560E 02	0.560E 02	0.682E 00	0.682E 00
0.700	0.216E 02	0.496E 02	0.495E 01	0.128E 03	0.128E 03	0.526E 00	0.526E 00
0.800	0.453E 02	0.658E 02	0.470E 01	0.372E 03	0.372E 03	0.450E 00	0.450E 00
0.850	0.735E 02	0.890E 02	0.456E 01	0.774E 03	0.774E 03	0.404E 00	0.404E 00
0.900	0.138E 03	0.129E 03	0.441E 01	0.207E 04	0.207E 04	0.340E 00	0.340E 00
0.928	0.227E 03	0.163E 03	0.432E 01	0.447E 04	0.447E 04	0.296E 00	0.296E 00

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.382E 02	0.581E 01	0.753E 01	0.753E 01	0.148E 01	0.148E 01
0.100	0.684E 00	0.383E 02	0.580E 01	0.800E 01	0.753E 01	0.144E 01	0.148E 01
0.200	0.144E 01	0.387E 02	0.579E 01	0.959E 01	0.753E 01	0.134E 01	0.148E 01
0.300	0.242E 01	0.393E 02	0.577E 01	0.130E 02	0.753E 01	0.118E 01	0.148E 01
0.400	0.377E 01	0.404E 02	0.573E 01	0.193E 02	0.753E 01	0.102E 01	0.148E 01
0.500	0.586E 01	0.420E 02	0.567E 01	0.320E 02	0.753E 01	0.867E 00	0.148E 01
0.600	0.955E 01	0.434E 02	0.554E 01	0.594E 02	0.753E 01	0.721E 00	0.148E 01
0.700	0.168E 02	0.433E 02	0.542E 01	0.128E 03	0.753E 01	0.609E 00	0.148E 01
0.800	0.344E 02	0.557E 02	0.528E 01	0.383E 03	0.753E 01	0.479E 00	0.148E 01
0.850	0.551E 02	0.701E 02	0.521E 01	0.806E 03	0.753E 01	0.408E 00	0.148E 01
0.900	0.102E 03	0.917E 02	0.513E 01	0.215E 04	0.753E 01	0.338E 00	0.148E 01
0.928	0.166E 03	0.107E 03	0.508E 01	0.459E 04	0.753E 01	0.297E 00	0.148E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.382E 02	0.581E 01	0.753E 01	0.753E 01	0.148E 01	0.148E 01
0.100	0.968E 00	0.384E 02	0.580E 01	0.800E 01	0.800E 01	0.144E 01	0.144E 01
0.200	0.205E 01	0.392E 02	0.577E 01	0.959E 01	0.959E 01	0.134E 01	0.134E 01
0.300	0.342E 01	0.405E 02	0.573E 01	0.130E 02	0.130E 02	0.118E 01	0.118E 01
0.400	0.533E 01	0.427E 02	0.565E 01	0.193E 02	0.193E 02	0.102E 01	0.102E 01
0.500	0.829E 01	0.459E 02	0.553E 01	0.320E 02	0.320E 02	0.867E 00	0.867E 00
0.600	0.135E 02	0.486E 02	0.528E 01	0.594E 02	0.594E 02	0.721E 00	0.721E 00
0.700	0.237E 02	0.484E 02	0.503E 01	0.128E 03	0.128E 03	0.609E 00	0.609E 00
0.800	0.487E 02	0.733E 02	0.475E 01	0.383E 03	0.383E 03	0.479E 00	0.479E 00
0.850	0.780E 02	0.102E 03	0.460E 01	0.806E 03	0.806E 03	0.408E 00	0.408E 00
0.900	0.145E 03	0.145E 03	0.444E 01	0.215E 04	0.215E 04	0.338E 00	0.338E 00
0.928	0.235E 03	0.177E 03	0.435E 01	0.459E 04	0.459E 04	0.297E 00	0.297E 00

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.155E 03	0.607E 01	0.204E 02	0.204E 02	0.158E 01	0.158E 01
0.100	0.193E 01	0.156E 03	0.606E 01	0.214E 02	0.204E 02	0.155E 01	0.158E 01
0.200	0.404E 01	0.156E 03	0.605E 01	0.247E 02	0.204E 02	0.146E 01	0.158E 01
0.300	0.663E 01	0.157E 03	0.602E 01	0.316E 02	0.204E 02	0.131E 01	0.158E 01
0.400	0.100E 02	0.159E 03	0.598E 01	0.441E 02	0.204E 02	0.116E 01	0.158E 01
0.500	0.150E 02	0.161E 03	0.590E 01	0.676E 02	0.204E 02	0.995E 00	0.158E 01
0.600	0.234E 02	0.165E 03	0.576E 01	0.116E 03	0.204E 02	0.813E 00	0.158E 01
0.700	0.387E 02	0.171E 03	0.562E 01	0.237E 03	0.204E 02	0.627E 00	0.158E 01
0.800	0.727E 02	0.182E 03	0.546E 01	0.621E 03	0.204E 02	0.502E 00	0.158E 01
0.850	0.109E 03	0.196E 03	0.537E 01	0.119E 04	0.204E 02	0.449E 00	0.158E 01
0.900	0.187E 03	0.220E 03	0.527E 01	0.287E 04	0.204E 02	0.394E 00	0.158E 01
0.928	0.282E 03	0.240E 03	0.521E 01	0.574E 04	0.204E 02	0.359E 00	0.158E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.155E 03	0.607E 01	0.204E 02	0.204E 02	0.158E 01	0.158E 01
0.100	0.272E 01	0.156E 03	0.606E 01	0.214E 02	0.214E 02	0.155E 01	0.155E 01
0.200	0.572E 01	0.157E 03	0.603E 01	0.247E 02	0.247E 02	0.146E 01	0.146E 01
0.300	0.938E 01	0.159E 03	0.597E 01	0.316E 02	0.316E 02	0.131E 01	0.131E 01
0.400	0.142E 02	0.162E 03	0.588E 01	0.441E 02	0.441E 02	0.116E 01	0.116E 01
0.500	0.213E 02	0.167E 03	0.573E 01	0.676E 02	0.676E 02	0.995E 00	0.995E 00
0.600	0.332E 02	0.174E 03	0.545E 01	0.116E 03	0.116E 03	0.813E 00	0.813E 00
0.700	0.548E 02	0.187E 03	0.517E 01	0.237E 03	0.237E 03	0.627E 00	0.627E 00
0.800	0.102E 03	0.208E 03	0.485E 01	0.621E 03	0.621E 03	0.502E 00	0.502E 00
0.850	0.155E 03	0.236E 03	0.467E 01	0.119E 04	0.119E 04	0.449E 00	0.449E 00
0.900	0.265E 03	0.285E 03	0.448E 01	0.287E 04	0.287E 04	0.394E 00	0.394E 00
0.928	0.399E 03	0.325E 03	0.436E 01	0.574E 04	0.574E 04	0.359E 00	0.359E 00

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.259E 03	0.611E 01	0.320E 02	0.320E 02	0.160E 01	0.160E 01
0.100	0.304E 01	0.260E 03	0.610E 01	0.335E 02	0.320E 02	0.157E 01	0.160E 01
0.200	0.638E 01	0.260E 03	0.608E 01	0.385E 02	0.320E 02	0.148E 01	0.160E 01
0.300	0.104E 02	0.262E 03	0.606E 01	0.488E 02	0.320E 02	0.134E 01	0.160E 01
0.400	0.157E 02	0.264E 03	0.601E 01	0.673E 02	0.320E 02	0.118E 01	0.160E 01
0.500	0.235E 02	0.267E 03	0.594E 01	0.101E 03	0.320E 02	0.101E 01	0.160E 01
0.600	0.363E 02	0.273E 03	0.579E 01	0.171E 03	0.320E 02	0.837E 00	0.160E 01
0.700	0.593E 02	0.282E 03	0.565E 01	0.339E 03	0.320E 02	0.652E 00	0.160E 01
0.800	0.110E 03	0.296E 03	0.548E 01	0.835E 03	0.320E 02	0.536E 00	0.160E 01
0.850	0.165E 03	0.313E 03	0.539E 01	0.155E 04	0.320E 02	0.483E 00	0.160E 01
0.900	0.279E 03	0.344E 03	0.529E 01	0.361E 04	0.320E 02	0.418E 00	0.160E 01
0.928	0.417E 03	0.369E 03	0.523E 01	0.710E 04	0.320E 02	0.372E 00	0.160E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.259E 03	0.611E 01	0.320E 02	0.320E 02	0.160E 01	0.160E 01
0.100	0.431E 01	0.260E 03	0.610E 01	0.335E 02	0.335E 02	0.157E 01	0.157E 01
0.200	0.902E 01	0.261E 03	0.606E 01	0.385E 02	0.385E 02	0.148E 01	0.148E 01
0.300	0.147E 02	0.264E 03	0.601E 01	0.488E 02	0.488E 02	0.134E 01	0.134E 01
0.400	0.222E 02	0.269E 03	0.592E 01	0.673E 02	0.673E 02	0.118E 01	0.118E 01
0.500	0.332E 02	0.276E 03	0.577E 01	0.101E 03	0.101E 03	0.101E 01	0.101E 01
0.600	0.513E 02	0.286E 03	0.548E 01	0.171E 03	0.171E 03	0.837E 00	0.837E 00
0.700	0.839E 02	0.304E 03	0.519E 01	0.339E 03	0.339E 03	0.652E 00	0.652E 00
0.800	0.155E 03	0.332E 03	0.486E 01	0.835E 03	0.835E 03	0.536E 00	0.536E 00
0.850	0.233E 03	0.368E 03	0.468E 01	0.155E 04	0.155E 04	0.483E 00	0.483E 00
0.900	0.395E 03	0.429E 03	0.448E 01	0.361E 04	0.361E 04	0.418E 00	0.418E 00
0.928	0.590E 03	0.478E 03	0.435E 01	0.710E 04	0.710E 04	0.372E 00	0.372E 00

Table C-1 (cont'd)

C-18

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30

TURBULENT FILM, REYNOLDS NUMBER = 30000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.353E 03	0.612E 01	0.423E 02	0.423E 02	0.161E 01	0.161E 01
0.100	0.404E 01	0.353E 03	0.612E 01	0.443E 02	0.423E 02	0.158E 01	0.161E 01
0.200	0.847E 01	0.354E 03	0.610E 01	0.508E 02	0.423E 02	0.149E 01	0.161E 01
0.300	0.138E 02	0.356E 03	0.607E 01	0.642E 02	0.423E 02	0.135E 01	0.161E 01
0.400	0.208E 02	0.359E 03	0.603E 01	0.880E 02	0.423E 02	0.119E 01	0.161E 01
0.500	0.310E 02	0.363E 03	0.595E 01	0.132E 03	0.423E 02	0.103E 01	0.161E 01
0.600	0.483E 02	0.370E 03	0.575E 01	0.216E 03	0.423E 02	0.832E 00	0.161E 01
0.700	0.776E 02	0.381E 03	0.548E 01	0.428E 03	0.423E 02	0.641E 00	0.161E 01
0.800	0.143E 03	0.399E 03	0.520E 01	0.103E 04	0.423E 02	0.541E 00	0.161E 01
0.850	0.214E 03	0.421E 03	0.505E 01	0.187E 04	0.423E 02	0.500E 00	0.161E 01
0.900	0.363E 03	0.457E 03	0.489E 01	0.414E 04	0.423E 02	0.451E 00	0.161E 01
0.928	0.543E 03	0.486E 03	0.479E 01	0.778E 04	0.423E 02	0.413E 00	0.161E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.353E 03	0.612E 01	0.423E 02	0.423E 02	0.161E 01	0.161E 01
0.100	0.572E 01	0.353E 03	0.611E 01	0.443E 02	0.443E 02	0.158E 01	0.158E 01
0.200	0.119E 02	0.355E 03	0.608E 01	0.508E 02	0.508E 02	0.149E 01	0.149E 01
0.300	0.195E 02	0.359E 03	0.603E 01	0.642E 02	0.642E 02	0.135E 01	0.135E 01
0.400	0.294E 02	0.365E 03	0.594E 01	0.880E 02	0.880E 02	0.119E 01	0.119E 01
0.500	0.439E 02	0.373E 03	0.579E 01	0.132E 03	0.132E 03	0.103E 01	0.103E 01
0.600	0.683E 02	0.387E 03	0.538E 01	0.216E 03	0.216E 03	0.832E 00	0.832E 00
0.700	0.109E 03	0.410E 03	0.483E 01	0.428E 03	0.428E 03	0.641E 00	0.641E 00
0.800	0.202E 03	0.446E 03	0.427E 01	0.103E 04	0.103E 04	0.541E 00	0.541E 00
0.850	0.303E 03	0.489E 03	0.397E 01	0.187E 04	0.187E 04	0.500E 00	0.500E 00
0.900	0.514E 03	0.562E 03	0.365E 01	0.414E 04	0.414E 04	0.451E 00	0.451E 00
0.928	0.767E 03	0.620E 03	0.346E 01	0.778E 04	0.778E 04	0.413E 00	0.413E 00

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
LAMINAR FILM

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.260E 02	0.506E 01	0.118E 02	0.118E 02	0.152E 01	0.152E 01
0.100	0.100E 01	0.261E 02	0.506E 01	0.125E 02	0.118E 02	0.149E 01	0.152E 01
0.200	0.213E 01	0.265E 02	0.505E 01	0.148E 02	0.118E 02	0.140E 01	0.152E 01
0.300	0.355E 01	0.273E 02	0.505E 01	0.197E 02	0.118E 02	0.126E 01	0.152E 01
0.400	0.549E 01	0.285E 02	0.503E 01	0.287E 02	0.118E 02	0.112E 01	0.152E 01
0.500	0.842E 01	0.303E 02	0.502E 01	0.459E 02	0.118E 02	0.971E 00	0.152E 01
0.600	0.134E 02	0.333E 02	0.499E 01	0.830E 02	0.118E 02	0.808E 00	0.152E 01
0.700	0.231E 02	0.389E 02	0.496E 01	0.177E 03	0.118E 02	0.634E 00	0.152E 01
0.800	0.458E 02	0.477E 02	0.493E 01	0.492E 03	0.118E 02	0.497E 00	0.152E 01
0.850	0.714E 02	0.601E 02	0.491E 01	0.987E 03	0.118E 02	0.431E 00	0.152E 01
0.900	0.128E 03	0.820E 02	0.490E 01	0.250E 04	0.118E 02	0.364E 00	0.152E 01
0.928	0.202E 03	0.999E 02	0.489E 01	0.520E 04	0.118E 02	0.324E 00	0.152E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.260E 02	0.506E 01	0.118E 02	0.118E 02	0.152E 01	0.152E 01
0.100	0.142E 01	0.262E 02	0.506E 01	0.125E 02	0.125E 02	0.149E 01	0.149E 01
0.200	0.301E 01	0.270E 02	0.505E 01	0.148E 02	0.148E 02	0.140E 01	0.140E 01
0.300	0.502E 01	0.285E 02	0.503E 01	0.197E 02	0.197E 02	0.126E 01	0.126E 01
0.400	0.776E 01	0.309E 02	0.501E 01	0.287E 02	0.287E 02	0.112E 01	0.112E 01
0.500	0.119E 02	0.346E 02	0.497E 01	0.459E 02	0.459E 02	0.971E 00	0.971E 00
0.600	0.189E 02	0.405E 02	0.492E 01	0.830E 02	0.830E 02	0.808E 00	0.808E 00
0.700	0.326E 02	0.518E 02	0.486E 01	0.177E 03	0.177E 03	0.634E 00	0.634E 00
0.800	0.647E 02	0.694E 02	0.480E 01	0.492E 03	0.492E 03	0.497E 00	0.497E 00
0.850	0.101E 03	0.943E 02	0.476E 01	0.987E 03	0.987E 03	0.431E 00	0.431E 00
0.900	0.181E 03	0.138E 03	0.473E 01	0.250E 04	0.250E 04	0.364E 00	0.364E 00
0.928	0.286E 03	0.173E 03	0.472E 01	0.520E 04	0.520E 04	0.324E 00	0.324E 00

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.395E 02	0.518E 01	0.145E 02	0.145E 02	0.162E 01	0.162E 01
0.100	0.123E 01	0.396E 02	0.517E 01	0.152E 02	0.145E 02	0.158E 01	0.162E 01
0.200	0.259E 01	0.400E 02	0.517E 01	0.177E 02	0.145E 02	0.149E 01	0.162E 01
0.300	0.428E 01	0.407E 02	0.516E 01	0.230E 02	0.145E 02	0.134E 01	0.162E 01
0.400	0.652E 01	0.419E 02	0.515E 01	0.327E 02	0.145E 02	0.118E 01	0.162E 01
0.500	0.984E 01	0.437E 02	0.513E 01	0.512E 02	0.145E 02	0.102E 01	0.162E 01
0.600	0.153E 02	0.467E 02	0.510E 01	0.907E 02	0.145E 02	0.843E 00	0.162E 01
0.700	0.258E 02	0.524E 02	0.506E 01	0.191E 03	0.145E 02	0.654E 00	0.162E 01
0.800	0.499E 02	0.615E 02	0.503E 01	0.522E 03	0.145E 02	0.504E 00	0.162E 01
0.850	0.766E 02	0.741E 02	0.501E 01	0.103E 04	0.145E 02	0.435E 00	0.162E 01
0.900	0.135E 03	0.963E 02	0.499E 01	0.260E 04	0.145E 02	0.366E 00	0.162E 01
0.928	0.211E 03	0.114E 03	0.498E 01	0.535E 04	0.145E 02	0.326E 00	0.162E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.395E 02	0.518E 01	0.145E 02	0.145E 02	0.162E 01	0.162E 01
0.100	0.174E 01	0.397E 02	0.517E 01	0.152E 02	0.152E 02	0.158E 01	0.158E 01
0.200	0.367E 01	0.405E 02	0.516E 01	0.177E 02	0.177E 02	0.149E 01	0.149E 01
0.300	0.605E 01	0.420E 02	0.514E 01	0.230E 02	0.230E 02	0.134E 01	0.134E 01
0.400	0.923E 01	0.443E 02	0.511E 01	0.327E 02	0.327E 02	0.118E 01	0.118E 01
0.500	0.139E 02	0.479E 02	0.507E 01	0.512E 02	0.512E 02	0.102E 01	0.102E 01
0.600	0.217E 02	0.539E 02	0.502E 01	0.907E 02	0.907E 02	0.843E 00	0.843E 00
0.700	0.365E 02	0.653E 02	0.495E 01	0.191E 03	0.191E 03	0.654E 00	0.654E 00
0.800	0.706E 02	0.834E 02	0.488E 01	0.522E 03	0.522E 03	0.504E 00	0.504E 00
0.850	0.108E 03	0.108E 03	0.484E 01	0.103E 04	0.103E 04	0.435E 00	0.435E 00
0.900	0.191E 03	0.153E 03	0.481E 01	0.260E 04	0.260E 04	0.366E 00	0.366E 00
0.928	0.298E 03	0.189E 03	0.479E 01	0.535E 04	0.535E 04	0.326E 00	0.326E 00

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.161E 03	0.539E 01	0.440E 02	0.440E 02	0.173E 01	0.173E 01
0.100	0.397E 01	0.161E 03	0.539E 01	0.457E 02	0.440E 02	0.170E 01	0.173E 01
0.200	0.828E 01	0.162E 03	0.538E 01	0.512E 02	0.440E 02	0.163E 01	0.173E 01
0.300	0.134E 02	0.163E 03	0.537E 01	0.623E 02	0.440E 02	0.152E 01	0.173E 01
0.400	0.200E 02	0.165E 03	0.535E 01	0.817E 02	0.440E 02	0.139E 01	0.173E 01
0.500	0.293E 02	0.168E 03	0.532E 01	0.116E 03	0.440E 02	0.124E 01	0.173E 01
0.600	0.437E 02	0.172E 03	0.527E 01	0.183E 03	0.440E 02	0.106E 01	0.173E 01
0.700	0.691E 02	0.180E 03	0.522E 01	0.334E 03	0.440E 02	0.852E 00	0.173E 01
0.800	0.122E 03	0.193E 03	0.516E 01	0.774E 03	0.440E 02	0.677E 00	0.173E 01
0.850	0.176E 03	0.210E 03	0.513E 01	0.143E 04	0.440E 02	0.576E 00	0.173E 01
0.900	0.281E 03	0.238E 03	0.510E 01	0.343E 04	0.440E 02	0.456E 00	0.173E 01
0.928	0.399E 03	0.261E 03	0.508E 01	0.700E 04	0.440E 02	0.378E 00	0.173E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.161E 03	0.539E 01	0.440E 02	0.440E 02	0.173E 01	0.173E 01
0.100	0.561E 01	0.161E 03	0.539E 01	0.457E 02	0.457E 02	0.170E 01	0.170E 01
0.200	0.117E 02	0.163E 03	0.537E 01	0.512E 02	0.512E 02	0.163E 01	0.163E 01
0.300	0.190E 02	0.165E 03	0.534E 01	0.623E 02	0.623E 02	0.152E 01	0.152E 01
0.400	0.283E 02	0.169E 03	0.530E 01	0.817E 02	0.817E 02	0.139E 01	0.139E 01
0.500	0.414E 02	0.175E 03	0.524E 01	0.116E 03	0.116E 03	0.124E 01	0.124E 01
0.600	0.618E 02	0.183E 03	0.515E 01	0.183E 03	0.183E 03	0.106E 01	0.106E 01
0.700	0.978E 02	0.199E 03	0.505E 01	0.334E 03	0.334E 03	0.852E 00	0.852E 00
0.800	0.173E 03	0.225E 03	0.492E 01	0.774E 03	0.774E 03	0.677E 00	0.677E 00
0.850	0.249E 03	0.258E 03	0.486E 01	0.143E 04	0.143E 04	0.576E 00	0.576E 00
0.900	0.398E 03	0.315E 03	0.480E 01	0.343E 04	0.343E 04	0.456E 00	0.456E 00
0.928	0.565E 03	0.361E 03	0.477E 01	0.700E 04	0.700E 04	0.378E 00	0.378E 00

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.269E 03	0.543E 01	0.703E 02	0.703E 02	0.174E 01	0.174E 01
0.100	0.641E 01	0.269E 03	0.542E 01	0.729E 02	0.703E 02	0.172E 01	0.174E 01
0.200	0.133E 02	0.270E 03	0.542E 01	0.814E 02	0.703E 02	0.165E 01	0.174E 01
0.300	0.216E 02	0.272E 03	0.540E 01	0.981E 02	0.703E 02	0.155E 01	0.174E 01
0.400	0.322E 02	0.275E 03	0.538E 01	0.126E 03	0.703E 02	0.142E 01	0.174E 01
0.500	0.470E 02	0.279E 03	0.535E 01	0.177E 03	0.703E 02	0.128E 01	0.174E 01
0.600	0.699E 02	0.285E 03	0.530E 01	0.272E 03	0.703E 02	0.111E 01	0.174E 01
0.700	0.110E 03	0.296E 03	0.524E 01	0.473E 03	0.703E 02	0.917E 00	0.174E 01
0.800	0.195E 03	0.315E 03	0.517E 01	0.974E 03	0.703E 02	0.781E 00	0.174E 01
0.850	0.282E 03	0.338E 03	0.514E 01	0.165E 04	0.703E 02	0.704E 00	0.174E 01
0.900	0.449E 03	0.375E 03	0.510E 01	0.360E 04	0.703E 02	0.599E 00	0.174E 01
0.928	0.629E 03	0.405E 03	0.508E 01	0.699E 04	0.703E 02	0.521E 00	0.174E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.269E 03	0.543E 01	0.703E 02	0.703E 02	0.174E 01	0.174E 01
0.100	0.906E 01	0.269E 03	0.542E 01	0.729E 02	0.729E 02	0.172E 01	0.172E 01
0.200	0.188E 02	0.271E 03	0.540E 01	0.814E 02	0.814E 02	0.165E 01	0.165E 01
0.300	0.306E 02	0.275E 03	0.538E 01	0.981E 02	0.981E 02	0.155E 01	0.155E 01
0.400	0.455E 02	0.281E 03	0.533E 01	0.126E 03	0.126E 03	0.142E 01	0.142E 01
0.500	0.665E 02	0.289E 03	0.527E 01	0.177E 03	0.177E 03	0.128E 01	0.128E 01
0.600	0.989E 02	0.301E 03	0.518E 01	0.272E 03	0.272E 03	0.111E 01	0.111E 01
0.700	0.155E 03	0.324E 03	0.506E 01	0.473E 03	0.473E 03	0.917E 00	0.917E 00
0.800	0.276E 03	0.361E 03	0.492E 01	0.974E 03	0.974E 03	0.781E 00	0.781E 00
0.850	0.399E 03	0.407E 03	0.485E 01	0.165E 04	0.165E 04	0.704E 00	0.704E 00
0.900	0.635E 03	0.481E 03	0.477E 01	0.360E 04	0.360E 04	0.599E 00	0.599E 00
0.928	0.889E 03	0.540E 03	0.473E 01	0.699E 04	0.699E 04	0.521E 00	0.521E 00

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 30000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.366E 03	0.544E 01	0.939E 02	0.939E 02	0.175E 01	0.175E 01
0.100	0.859E 01	0.366E 03	0.544E 01	0.972E 02	0.939E 02	0.173E 01	0.175E 01
0.200	0.179E 02	0.367E 03	0.543E 01	0.108E 03	0.939E 02	0.166E 01	0.175E 01
0.300	0.289E 02	0.370E 03	0.542E 01	0.130E 03	0.939E 02	0.156E 01	0.175E 01
0.400	0.431E 02	0.373E 03	0.539E 01	0.167E 03	0.939E 02	0.144E 01	0.175E 01
0.500	0.628E 02	0.379E 03	0.536E 01	0.232E 03	0.939E 02	0.130E 01	0.175E 01
0.600	0.933E 02	0.387E 03	0.532E 01	0.352E 03	0.939E 02	0.113E 01	0.175E 01
0.700	0.147E 03	0.401E 03	0.526E 01	0.599E 03	0.939E 02	0.947E 00	0.175E 01
0.800	0.261E 03	0.425E 03	0.518E 01	0.111E 04	0.939E 02	0.905E 00	0.175E 01
0.850	0.378E 03	0.453E 03	0.514E 01	0.181E 04	0.939E 02	0.801E 00	0.175E 01
0.900	0.604E 03	0.500E 03	0.510E 01	0.403E 04	0.939E 02	0.504E 00	0.175E 01
0.928	0.849E 03	0.536E 03	0.507E 01	0.832E 04	0.939E 02	0.241E 00	0.175E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.366E 03	0.544E 01	0.939E 02	0.939E 02	0.175E 01	0.175E 01
0.100	0.121E 02	0.366E 03	0.544E 01	0.972E 02	0.972E 02	0.173E 01	0.173E 01
0.200	0.253E 02	0.369E 03	0.542E 01	0.108E 03	0.108E 03	0.166E 01	0.166E 01
0.300	0.410E 02	0.374E 03	0.539E 01	0.130E 03	0.130E 03	0.156E 01	0.156E 01
0.400	0.610E 02	0.381E 03	0.535E 01	0.167E 03	0.167E 03	0.144E 01	0.144E 01
0.500	0.889E 02	0.391E 03	0.528E 01	0.232E 03	0.232E 03	0.130E 01	0.130E 01
0.600	0.132E 03	0.408E 03	0.519E 01	0.352E 03	0.352E 03	0.113E 01	0.113E 01
0.700	0.207E 03	0.436E 03	0.507E 01	0.599E 03	0.599E 03	0.947E 00	0.947E 00
0.800	0.369E 03	0.484E 03	0.493E 01	0.111E 04	0.111E 04	0.905E 00	0.905E 00
0.850	0.535E 03	0.541E 03	0.484E 01	0.181E 04	0.181E 04	0.801E 00	0.801E 00
0.900	0.855E 03	0.634E 03	0.476E 01	0.403E 04	0.403E 04	0.504E 00	0.504E 00
0.928	0.120E 04	0.707E 03	0.471E 01	0.832E 04	0.832E 04	0.241E 00	0.241E 00

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
LAMINAR FILM

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.352E 02	0.450E 01	0.189E 02	0.189E 02	0.107E 01	0.107E 01
0.100	0.117E 01	0.354E 02	0.450E 01	0.201E 02	0.189E 02	0.104E 01	0.107E 01
0.200	0.246E 01	0.359E 02	0.450E 01	0.242E 02	0.189E 02	0.974E 00	0.107E 01
0.300	0.411E 01	0.369E 02	0.449E 01	0.324E 02	0.189E 02	0.877E 00	0.107E 01
0.400	0.640E 01	0.386E 02	0.448E 01	0.482E 02	0.189E 02	0.763E 00	0.107E 01
0.500	0.998E 01	0.409E 02	0.446E 01	0.798E 02	0.189E 02	0.633E 00	0.107E 01
0.600	0.160E 02	0.455E 02	0.444E 01	0.148E 03	0.189E 02	0.521E 00	0.107E 01
0.700	0.278E 02	0.503E 02	0.442E 01	0.312E 03	0.189E 02	0.467E 00	0.107E 01
0.800	0.557E 02	0.661E 02	0.438E 01	0.876E 03	0.189E 02	0.398E 00	0.107E 01
0.850	0.877E 02	0.807E 02	0.436E 01	0.176E 04	0.189E 02	0.352E 00	0.107E 01
0.900	0.160E 03	0.100E 03	0.433E 01	0.447E 04	0.189E 02	0.297E 00	0.107E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.352E 02	0.450E 01	0.189E 02	0.189E 02	0.107E 01	0.107E 01
0.100	0.165E 01	0.355E 02	0.450E 01	0.201E 02	0.201E 02	0.104E 01	0.104E 01
0.200	0.349E 01	0.367E 02	0.449E 01	0.242E 02	0.242E 02	0.974E 00	0.974E 00
0.300	0.581E 01	0.385E 02	0.447E 01	0.324E 02	0.324E 02	0.877E 00	0.877E 00
0.400	0.905E 01	0.420E 02	0.445E 01	0.482E 02	0.482E 02	0.763E 00	0.763E 00
0.500	0.141E 02	0.467E 02	0.443E 01	0.798E 02	0.798E 02	0.633E 00	0.633E 00
0.600	0.227E 02	0.558E 02	0.439E 01	0.148E 03	0.148E 03	0.521E 00	0.521E 00
0.700	0.393E 02	0.653E 02	0.433E 01	0.312E 03	0.312E 03	0.467E 00	0.467E 00
0.800	0.788E 02	0.971E 02	0.426E 01	0.876E 03	0.876E 03	0.398E 00	0.398E 00
0.850	0.124E 03	0.126E 03	0.421E 01	0.176E 04	0.176E 04	0.352E 00	0.352E 00
0.900	0.226E 03	0.166E 03	0.416E 01	0.447E 04	0.447E 04	0.297E 00	0.297E 00

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.476E 02	0.463E 01	0.212E 02	0.212E 02	0.111E 01	0.111E 01
0.100	0.131E 01	0.477E 02	0.463E 01	0.224E 02	0.212E 02	0.109E 01	0.111E 01
0.200	0.276E 01	0.483E 02	0.462E 01	0.268E 02	0.212E 02	0.101E 01	0.111E 01
0.300	0.459E 01	0.492E 02	0.461E 01	0.355E 02	0.212E 02	0.908E 00	0.111E 01
0.400	0.708E 01	0.510E 02	0.460E 01	0.522E 02	0.212E 02	0.789E 00	0.111E 01
0.500	0.109E 02	0.492E 02	0.458E 01	0.826E 02	0.212E 02	0.700E 00	0.111E 01
0.600	0.174E 02	0.506E 02	0.456E 01	0.148E 03	0.212E 02	0.607E 00	0.111E 01
0.700	0.298E 02	0.592E 02	0.453E 01	0.320E 03	0.212E 02	0.506E 00	0.111E 01
0.800	0.589E 02	0.790E 02	0.449E 01	0.912E 03	0.212E 02	0.401E 00	0.111E 01
0.850	0.919E 02	0.944E 02	0.446E 01	0.183E 04	0.212E 02	0.349E 00	0.111E 01
0.900	0.165E 03	0.114E 03	0.443E 01	0.459E 04	0.212E 02	0.298E 00	0.111E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.476E 02	0.463E 01	0.212E 02	0.212E 02	0.111E 01	0.111E 01
0.100	0.185E 01	0.479E 02	0.462E 01	0.224E 02	0.224E 02	0.109E 01	0.109E 01
0.200	0.391E 01	0.490E 02	0.461E 01	0.268E 02	0.268E 02	0.101E 01	0.101E 01
0.300	0.649E 01	0.507E 02	0.460E 01	0.355E 02	0.355E 02	0.908E 00	0.908E 00
0.400	0.100E 02	0.544E 02	0.457E 01	0.522E 02	0.522E 02	0.789E 00	0.789E 00
0.500	0.154E 02	0.508E 02	0.454E 01	0.826E 02	0.826E 02	0.700E 00	0.700E 00
0.600	0.246E 02	0.536E 02	0.450E 01	0.148E 03	0.148E 03	0.607E 00	0.607E 00
0.700	0.422E 02	0.708E 02	0.444E 01	0.320E 03	0.320E 03	0.506E 00	0.506E 00
0.800	0.833E 02	0.110E 03	0.435E 01	0.912E 03	0.912E 03	0.401E 00	0.401E 00
0.850	0.130E 03	0.141E 03	0.430E 01	0.183E 04	0.183E 04	0.349E 00	0.349E 00
0.900	0.234E 03	0.180E 03	0.424E 01	0.459E 04	0.459E 04	0.298E 00	0.298E 00

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.173E 03	0.491E 01	0.502E 02	0.502E 02	0.123E 01	0.123E 01
0.100	0.330E 01	0.173E 03	0.491E 01	0.525E 02	0.502E 02	0.121E 01	0.123E 01
0.200	0.689E 01	0.174E 03	0.491E 01	0.607E 02	0.502E 02	0.113E 01	0.123E 01
0.300	0.112E 02	0.175E 03	0.489E 01	0.764E 02	0.502E 02	0.103E 01	0.123E 01
0.400	0.169E 02	0.177E 03	0.488E 01	0.105E 03	0.502E 02	0.912E 00	0.123E 01
0.500	0.253E 02	0.180E 03	0.486E 01	0.160E 03	0.502E 02	0.765E 00	0.123E 01
0.600	0.387E 02	0.186E 03	0.483E 01	0.273E 03	0.502E 02	0.626E 00	0.123E 01
0.700	0.628E 02	0.192E 03	0.479E 01	0.534E 03	0.502E 02	0.530E 00	0.123E 01
0.800	0.115E 03	0.211E 03	0.473E 01	0.133E 04	0.502E 02	0.447E 00	0.123E 01
0.850	0.169E 03	0.229E 03	0.470E 01	0.248E 04	0.502E 02	0.405E 00	0.123E 01
0.900	0.280E 03	0.253E 03	0.465E 01	0.575E 04	0.502E 02	0.361E 00	0.123E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.173E 03	0.491E 01	0.502E 02	0.502E 02	0.123E 01	0.123E 01
0.100	0.467E 01	0.173E 03	0.491E 01	0.525E 02	0.525E 02	0.121E 01	0.121E 01
0.200	0.974E 01	0.175E 03	0.490E 01	0.607E 02	0.607E 02	0.113E 01	0.113E 01
0.300	0.158E 02	0.177E 03	0.487E 01	0.764E 02	0.764E 02	0.103E 01	0.103E 01
0.400	0.239E 02	0.182E 03	0.484E 01	0.105E 03	0.105E 03	0.912E 00	0.912E 00
0.500	0.358E 02	0.188E 03	0.480E 01	0.160E 03	0.160E 03	0.765E 00	0.765E 00
0.600	0.548E 02	0.199E 03	0.474E 01	0.273E 03	0.273E 03	0.626E 00	0.626E 00
0.700	0.889E 02	0.211E 03	0.466E 01	0.534E 03	0.534E 03	0.530E 00	0.530E 00
0.800	0.162E 03	0.250E 03	0.455E 01	0.133E 04	0.133E 04	0.447E 00	0.447E 00
0.850	0.239E 03	0.285E 03	0.448E 01	0.248E 04	0.248E 04	0.405E 00	0.405E 00
0.900	0.396E 03	0.333E 03	0.439E 01	0.575E 04	0.575E 04	0.361E 00	0.361E 00

4 PAD BEARING L/D = 0.50

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.286E 03	0.496E 01	0.771E 02	0.771E 02	0.125E 01	0.125E 01
0.100	0.512E 01	0.286E 03	0.496E 01	0.805E 02	0.771E 02	0.123E 01	0.125E 01
0.200	0.106E 02	0.287E 03	0.495E 01	0.923E 02	0.771E 02	0.116E 01	0.125E 01
0.300	0.173E 02	0.289E 03	0.494E 01	0.115E 03	0.771E 02	0.106E 01	0.125E 01
0.400	0.260E 02	0.292E 03	0.493E 01	0.156E 03	0.771E 02	0.938E 00	0.125E 01
0.500	0.387E 02	0.296E 03	0.490E 01	0.234E 03	0.771E 02	0.791E 00	0.125E 01
0.600	0.589E 02	0.304E 03	0.487E 01	0.388E 03	0.771E 02	0.653E 00	0.125E 01
0.700	0.950E 02	0.312E 03	0.483E 01	0.726E 03	0.771E 02	0.565E 00	0.125E 01
0.800	0.172E 03	0.337E 03	0.478E 01	0.172E 04	0.771E 02	0.481E 00	0.125E 01
0.850	0.253E 03	0.358E 03	0.474E 01	0.314E 04	0.771E 02	0.432E 00	0.125E 01
0.900	0.414E 03	0.388E 03	0.469E 01	0.710E 04	0.771E 02	0.374E 00	0.125E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.286E 03	0.496E 01	0.771E 02	0.771E 02	0.125E 01	0.125E 01
0.100	0.724E 01	0.286E 03	0.496E 01	0.805E 02	0.805E 02	0.123E 01	0.123E 01
0.200	0.150E 02	0.288E 03	0.494E 01	0.923E 02	0.923E 02	0.116E 01	0.116E 01
0.300	0.245E 02	0.292E 03	0.492E 01	0.115E 03	0.115E 03	0.106E 01	0.106E 01
0.400	0.368E 02	0.298E 03	0.489E 01	0.156E 03	0.156E 03	0.938E 00	0.938E 00
0.500	0.548E 02	0.306E 03	0.485E 01	0.234E 03	0.234E 03	0.791E 00	0.791E 00
0.600	0.833E 02	0.322E 03	0.479E 01	0.388E 03	0.388E 03	0.653E 00	0.653E 00
0.700	0.134E 03	0.338E 03	0.471E 01	0.726E 03	0.726E 03	0.565E 00	0.565E 00
0.800	0.244E 03	0.387E 03	0.459E 01	0.172E 04	0.172E 04	0.481E 00	0.481E 00
0.850	0.358E 03	0.431E 03	0.451E 01	0.314E 04	0.314E 04	0.432E 00	0.432E 00
0.900	0.586E 03	0.490E 03	0.442E 01	0.710E 04	0.710E 04	0.374E 00	0.374E 00

 4 PAD BEARING L/D = 0.50

 PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 30000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.387E 03	0.498E 01	0.101E 03	0.101E 03	0.126E 01	0.126E 01
0.100	0.674E 01	0.387E 03	0.498E 01	0.105E 03	0.101E 03	0.124E 01	0.126E 01
0.200	0.140E 02	0.389E 03	0.497E 01	0.120E 03	0.101E 03	0.117E 01	0.126E 01
0.300	0.228E 02	0.391E 03	0.496E 01	0.149E 03	0.101E 03	0.107E 01	0.126E 01
0.400	0.342E 02	0.395E 03	0.495E 01	0.262E 03	0.101E 03	0.949E 00	0.126E 01
0.500	0.507E 02	0.400E 03	0.492E 01	0.301E 03	0.101E 03	0.795E 00	0.126E 01
0.600	0.769E 02	0.409E 03	0.489E 01	0.494E 03	0.101E 03	0.654E 00	0.126E 01
0.700	0.123E 03	0.420E 03	0.485E 01	0.911E 03	0.101E 03	0.570E 00	0.126E 01
0.800	0.224E 03	0.450E 03	0.480E 01	0.207E 04	0.101E 03	0.501E 00	0.126E 01
0.850	0.328E 03	0.475E 03	0.476E 01	0.364E 04	0.101E 03	0.463E 00	0.126E 01
0.900	0.538E 03	0.511E 03	0.471E 01	0.779E 04	0.101E 03	0.414E 00	0.126E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.387E 03	0.498E 01	0.101E 03	0.101E 03	0.126E 01	0.126E 01
0.100	0.954E 01	0.388E 03	0.498E 01	0.105E 03	0.105E 03	0.124E 01	0.124E 01
0.200	0.198E 02	0.390E 03	0.496E 01	0.120E 03	0.120E 03	0.117E 01	0.117E 01
0.300	0.322E 02	0.395E 03	0.494E 01	0.149E 03	0.149E 03	0.107E 01	0.107E 01
0.400	0.484E 02	0.402E 03	0.491E 01	0.202E 03	0.202E 03	0.949E 00	0.949E 00
0.500	0.718E 02	0.413E 03	0.487E 01	0.301E 03	0.301E 03	0.795E 00	0.795E 00
0.600	0.108E 03	0.432E 03	0.481E 01	0.494E 03	0.494E 03	0.654E 00	0.654E 00
0.700	0.175E 03	0.453E 03	0.473E 01	0.911E 03	0.911E 03	0.570E 00	0.570E 00
0.800	0.317E 03	0.512E 03	0.461E 01	0.207E 04	0.207E 04	0.501E 00	0.501E 00
0.850	0.465E 03	0.564E 03	0.453E 01	0.364E 04	0.364E 04	0.463E 00	0.463E 00
0.900	0.761E 03	0.634E 03	0.444E 01	0.779E 04	0.779E 04	0.414E 00	0.414E 00

 4 PAD BEARING L/D = 1.00

 PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 LAMINAR FILM

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.365E 02	0.383E 01	0.319E 02	0.319E 02	0.120E 01	0.120E 01
0.100	0.191E 01	0.367E 02	0.383E 01	0.335E 02	0.319E 02	0.117E 01	0.120E 01
0.200	0.402E 01	0.373E 02	0.382E 01	0.396E 02	0.319E 02	0.110E 01	0.120E 01
0.300	0.662E 01	0.383E 02	0.382E 01	0.512E 02	0.319E 02	0.101E 01	0.120E 01
0.400	0.101E 02	0.401E 02	0.381E 01	0.731E 02	0.319E 02	0.893E 00	0.120E 01
0.500	0.153E 02	0.425E 02	0.381E 01	0.115E 03	0.319E 02	0.758E 00	0.120E 01
0.600	0.240E 02	0.474E 02	0.380E 01	0.204E 03	0.319E 02	0.627E 00	0.120E 01
0.700	0.398E 02	0.526E 02	0.379E 01	0.417E 03	0.319E 02	0.524E 00	0.120E 01
0.800	0.757E 02	0.697E 02	0.378E 01	0.110E 04	0.319E 02	0.425E 00	0.120E 01
0.850	0.115E 03	0.852E 02	0.378E 01	0.215E 04	0.319E 02	0.375E 00	0.120E 01
0.900	0.201E 03	0.106E 03	0.377E 01	0.520E 04	0.319E 02	0.324E 00	0.120E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.365E 02	0.383E 01	0.319E 02	0.319E 02	0.120E 01	0.120E 01
0.100	0.271E 01	0.369E 02	0.382E 01	0.335E 02	0.335E 02	0.117E 01	0.117E 01
0.200	0.569E 01	0.381E 02	0.382E 01	0.396E 02	0.396E 02	0.110E 01	0.110E 01
0.300	0.936E 01	0.400E 02	0.381E 01	0.512E 02	0.512E 02	0.101E 01	0.101E 01
0.400	0.143E 02	0.437E 02	0.380E 01	0.731E 02	0.731E 02	0.893E 00	0.893E 00
0.500	0.217E 02	0.485E 02	0.379E 01	0.115E 03	0.115E 03	0.758E 00	0.758E 00
0.600	0.339E 02	0.583E 02	0.378E 01	0.204E 03	0.204E 03	0.627E 00	0.627E 00
0.700	0.564E 02	0.686E 02	0.376E 01	0.417E 03	0.417E 03	0.524E 00	0.524E 00
0.800	0.107E 03	0.102E 03	0.374E 01	0.110E 04	0.110E 04	0.425E 00	0.425E 00
0.850	0.163E 03	0.133E 03	0.373E 01	0.215E 04	0.215E 04	0.375E 00	0.375E 00
0.900	0.285E 03	0.177E 03	0.372E 01	0.520E 04	0.520E 04	0.324E 00	0.324E 00

Table C-1 (cont'd)

C-30

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.496E 02	0.391E 01	0.365E 02	0.365E 02	0.127E 01	0.127E 01
0.100	0.220E 01	0.498E 02	0.391E 01	0.383E 02	0.365E 02	0.125E 01	0.127E 01
0.200	0.461E 01	0.504E 02	0.391E 01	0.447E 02	0.365E 02	0.117E 01	0.127E 01
0.300	0.755E 01	0.514E 02	0.391E 01	0.571E 02	0.365E 02	0.107E 01	0.127E 01
0.400	0.114E 02	0.532E 02	0.390E 01	0.804E 02	0.365E 02	0.940E 00	0.127E 01
0.500	0.172E 02	0.557E 02	0.390E 01	0.125E 03	0.365E 02	0.791E 00	0.127E 01
0.600	0.265E 02	0.607E 02	0.389E 01	0.219E 03	0.365E 02	0.647E 00	0.127E 01
0.700	0.433E 02	0.660E 02	0.388E 01	0.443E 03	0.365E 02	0.534E 00	0.127E 01
0.800	0.808E 02	0.834E 02	0.387E 01	0.116E 04	0.365E 02	0.429E 00	0.127E 01
0.850	0.121E 03	0.991E 02	0.386E 01	0.223E 04	0.365E 02	0.378E 00	0.127E 01
0.900	0.209E 03	0.120E 03	0.386E 01	0.535E 04	0.365E 02	0.327E 00	0.127E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.496E 02	0.391E 01	0.365E 02	0.365E 02	0.127E 01	0.127E 01
0.100	0.312E 01	0.499E 02	0.391E 01	0.383E 02	0.383E 02	0.125E 01	0.125E 01
0.200	0.652E 01	0.511E 02	0.391E 01	0.447E 02	0.447E 02	0.117E 01	0.117E 01
0.300	0.106E 02	0.531E 02	0.390E 01	0.571E 02	0.571E 02	0.107E 01	0.107E 01
0.400	0.161E 02	0.568E 02	0.389E 01	0.804E 02	0.804E 02	0.940E 00	0.940E 00
0.500	0.243E 02	0.618E 02	0.388E 01	0.125E 03	0.125E 03	0.791E 00	0.791E 00
0.600	0.375E 02	0.718E 02	0.386E 01	0.219E 03	0.219E 03	0.647E 00	0.647E 00
0.700	0.613E 02	0.825E 02	0.384E 01	0.443E 03	0.443E 03	0.534E 00	0.534E 00
0.800	0.114E 03	0.117E 03	0.382E 01	0.116E 04	0.116E 04	0.429E 00	0.429E 00
0.850	0.172E 03	0.148E 03	0.381E 01	0.223E 04	0.223E 04	0.378E 00	0.378E 00
0.900	0.296E 03	0.192E 03	0.380E 01	0.535E 04	0.535E 04	0.327E 00	0.327E 00

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.182E 03	0.411E 01	0.948E 02	0.948E 02	0.146E 01	0.146E 01
0.100	0.645E 01	0.182E 03	0.411E 01	0.982E 02	0.948E 02	0.144E 01	0.146E 01
0.200	0.133E 02	0.183E 03	0.411E 01	0.109E 03	0.948E 02	0.138E 01	0.146E 01
0.300	0.216E 02	0.184E 03	0.410E 01	0.132E 03	0.948E 02	0.129E 01	0.146E 01
0.400	0.320E 02	0.187E 03	0.409E 01	0.171E 03	0.948E 02	0.117E 01	0.146E 01
0.500	0.466E 02	0.190E 03	0.407E 01	0.241E 03	0.948E 02	0.101E 01	0.146E 01
0.600	0.687E 02	0.197E 03	0.405E 01	0.379E 03	0.948E 02	0.852E 00	0.146E 01
0.700	0.106E 03	0.205E 03	0.403E 01	0.676E 03	0.948E 02	0.721E 00	0.146E 01
0.800	0.182E 03	0.228E 03	0.400E 01	0.159E 04	0.948E 02	0.569E 00	0.146E 01
0.850	0.256E 03	0.248E 03	0.398E 01	0.296E 04	0.948E 02	0.479E 00	0.146E 01
0.900	0.394E 03	0.276E 03	0.397E 01	0.701E 04	0.948E 02	0.380E 00	0.146E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.182E 03	0.411E 01	0.948E 02	0.948E 02	0.146E 01	0.146E 01
0.100	0.913E 01	0.182E 03	0.411E 01	0.982E 02	0.982E 02	0.144E 01	0.144E 01
0.200	0.189E 02	0.184E 03	0.410E 01	0.109E 03	0.109E 03	0.138E 01	0.138E 01
0.300	0.305E 02	0.187E 03	0.408E 01	0.132E 03	0.132E 03	0.129E 01	0.129E 01
0.400	0.452E 02	0.192E 03	0.406E 01	0.171E 03	0.171E 03	0.117E 01	0.117E 01
0.500	0.659E 02	0.199E 03	0.403E 01	0.241E 03	0.241E 03	0.101E 01	0.101E 01
0.600	0.971E 02	0.212E 03	0.399E 01	0.379E 03	0.379E 03	0.852E 00	0.852E 00
0.700	0.150E 03	0.228E 03	0.394E 01	0.676E 03	0.676E 03	0.721E 00	0.721E 00
0.800	0.257E 03	0.274E 03	0.388E 01	0.159E 04	0.159E 04	0.569E 00	0.569E 00
0.850	0.362E 03	0.315E 03	0.386E 01	0.296E 04	0.296E 04	0.479E 00	0.479E 00
0.900	0.557E 03	0.370E 03	0.383E 01	0.701E 04	0.701E 04	0.380E 00	0.380E 00

Table C-1 (cont'd)

C-32

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.300E 03	0.415E 01	0.148E 03	0.148E 03	0.148E 01	0.148E 01
0.100	0.103E 02	0.301E 03	0.415E 01	0.153E 03	0.148E 03	0.147E 01	0.148E 01
0.200	0.213E 02	0.302E 03	0.414E 01	0.170E 03	0.148E 03	0.141E 01	0.148E 01
0.300	0.344E 02	0.304E 03	0.413E 01	0.202E 03	0.148E 03	0.133E 01	0.148E 01
0.400	0.510E 02	0.308E 03	0.412E 01	0.257E 03	0.148E 03	0.122E 01	0.148E 01
0.500	0.741E 02	0.313E 03	0.410E 01	0.354E 03	0.148E 03	0.107E 01	0.148E 01
0.600	0.109E 03	0.322E 03	0.408E 01	0.532E 03	0.148E 03	0.924E 00	0.148E 01
0.700	0.169E 03	0.334E 03	0.405E 01	0.872E 03	0.148E 03	0.822E 00	0.148E 01
0.800	0.291E 03	0.365E 03	0.401E 01	0.181E 04	0.148E 03	0.702E 00	0.148E 01
0.850	0.408E 03	0.392E 03	0.399E 01	0.315E 04	0.148E 03	0.623E 00	0.148E 01
0.900	0.620E 03	0.428E 03	0.397E 01	0.700E 04	0.148E 03	0.524E 00	0.148E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.300E 03	0.415E 01	0.148E 03	0.148E 03	0.148E 01	0.148E 01
0.100	0.146E 02	0.301E 03	0.415E 01	0.153E 03	0.153E 03	0.147E 01	0.147E 01
0.200	0.302E 02	0.304E 03	0.414E 01	0.170E 03	0.170E 03	0.141E 01	0.141E 01
0.300	0.487E 02	0.308E 03	0.412E 01	0.202E 03	0.202E 03	0.133E 01	0.133E 01
0.400	0.722E 02	0.316E 03	0.409E 01	0.257E 03	0.257E 03	0.122E 01	0.122E 01
0.500	0.104E 03	0.325E 03	0.406E 01	0.354E 03	0.354E 03	0.107E 01	0.107E 01
0.600	0.154E 03	0.345E 03	0.402E 01	0.532E 03	0.532E 03	0.924E 00	0.924E 00
0.700	0.239E 03	0.368E 03	0.395E 01	0.872E 03	0.872E 03	0.822E 00	0.822E 00
0.800	0.411E 03	0.430E 03	0.388E 01	0.181E 04	0.181E 04	0.702E 00	0.702E 00
0.850	0.577E 03	0.483E 03	0.384E 01	0.315E 04	0.315E 04	0.623E 00	0.623E 00
0.900	0.877E 03	0.555E 03	0.380E 01	0.700E 04	0.700E 04	0.524E 00	0.524E 00

4 PAD BEARING L/D = 1.00

PAD ANGLE = 80.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 30000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.407E 03	0.417E 01	0.196E 03	0.196E 03	0.150E 01	0.150E 01
0.100	0.137E 02	0.407E 03	0.416E 01	0.202E 03	0.196E 03	0.148E 01	0.150E 01
0.200	0.285E 02	0.409E 03	0.416E 01	0.224E 03	0.196E 03	0.143E 01	0.150E 01
0.300	0.459E 02	0.412E 03	0.415E 01	0.265E 03	0.196E 03	0.134E 01	0.150E 01
0.400	0.680E 02	0.417E 03	0.414E 01	0.335E 03	0.196E 03	0.124E 01	0.150E 01
0.500	0.988E 02	0.423E 03	0.412E 01	0.456E 03	0.196E 03	0.110E 01	0.150E 01
0.600	0.145E 03	0.435E 03	0.410E 01	0.671E 03	0.196E 03	0.957E 00	0.150E 01
0.700	0.226E 03	0.450E 03	0.406E 01	0.101E 04	0.196E 03	0.937E 00	0.150E 01
0.800	0.390E 03	0.489E 03	0.402E 01	0.199E 04	0.196E 03	0.785E 00	0.150E 01
0.850	0.549E 03	0.522E 03	0.400E 01	0.350E 04	0.196E 03	0.568E 00	0.150E 01
0.900	0.838E 03	0.567E 03	0.397E 01	0.833E 04	0.196E 03	0.245E 00	0.150E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.407E 03	0.417E 01	0.196E 03	0.196E 03	0.150E 01	0.150E 01
0.100	0.194E 02	0.408E 03	0.416E 01	0.202E 03	0.202E 03	0.148E 01	0.148E 01
0.200	0.403E 02	0.411E 03	0.415E 01	0.224E 03	0.224E 03	0.143E 01	0.143E 01
0.300	0.650E 02	0.417E 03	0.413E 01	0.265E 03	0.265E 03	0.134E 01	0.134E 01
0.400	0.962E 02	0.426E 03	0.411E 01	0.335E 03	0.335E 03	0.124E 01	0.124E 01
0.500	0.139E 03	0.439E 03	0.407E 01	0.456E 03	0.456E 03	0.110E 01	0.110E 01
0.600	0.205E 03	0.463E 03	0.403E 01	0.671E 03	0.671E 03	0.957E 00	0.957E 00
0.700	0.320E 03	0.493E 03	0.396E 01	0.101E 04	0.101E 04	0.937E 00	0.937E 00
0.800	0.552E 03	0.571E 03	0.388E 01	0.199E 04	0.199E 04	0.785E 00	0.785E 00
0.850	0.777E 03	0.637E 03	0.383E 01	0.350E 04	0.350E 04	0.568E 00	0.568E 00
0.900	0.118E 04	0.726E 03	0.378E 01	0.833E 04	0.833E 04	0.245E 00	0.245E 00

TABLE C-2

DESIGN DATA FOR THE SIX PAD, TILTING PAD BEARING

 6 PAD BEARING L/D = 0.50

 PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 LAMINAR FILM

 LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.240E 02	0.757E 01	0.604E 01	0.604E 01	0.118E 01	0.118E 01
0.100	0.450E 00	0.241E 02	0.757E 01	0.632E 01	0.611E 01	0.116E 01	0.117E 01
0.200	0.947E 00	0.245E 02	0.756E 01	0.729E 01	0.642E 01	0.111E 01	0.116E 01
0.300	0.156E 01	0.252E 02	0.756E 01	0.939E 01	0.697E 01	0.103E 01	0.113E 01
0.400	0.238E 01	0.262E 02	0.755E 01	0.132E 02	0.773E 01	0.947E 00	0.110E 01
0.500	0.363E 01	0.278E 02	0.754E 01	0.210E 02	0.890E 01	0.843E 00	0.105E 01
0.600	0.577E 01	0.300E 02	0.752E 01	0.383E 02	0.105E 02	0.734E 00	0.100E 01
0.700	0.100E 02	0.339E 02	0.750E 01	0.854E 02	0.128E 02	0.612E 00	0.960E 00
0.800	0.214E 02	0.393E 02	0.748E 01	0.260E 03	0.160E 02	0.462E 00	0.910E 00
0.850	0.358E 02	0.485E 02	0.746E 01	0.576E 03	0.181E 02	0.380E 00	0.884E 00
0.900	0.713E 02	0.656E 02	0.745E 01	0.167E 04	0.205E 02	0.307E 00	0.859E 00
0.928	0.123E 03	0.798E 02	0.743E 01	0.381E 04	0.221E 02	0.272E 00	0.844E 00

 LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.240E 02	0.757E 01	0.604E 01	0.604E 01	0.118E 01	0.118E 01
0.100	0.523E 00	0.242E 02	0.757E 01	0.642E 01	0.617E 01	0.116E 01	0.117E 01
0.200	0.111E 01	0.247E 02	0.756E 01	0.773E 01	0.660E 01	0.110E 01	0.115E 01
0.300	0.190E 01	0.256E 02	0.755E 01	0.105E 02	0.756E 01	0.100E 01	0.110E 01
0.400	0.302E 01	0.271E 02	0.754E 01	0.160E 02	0.937E 01	0.910E 00	0.102E 01
0.500	0.482E 01	0.293E 02	0.752E 01	0.271E 02	0.130E 02	0.808E 00	0.923E 00
0.600	0.809E 01	0.328E 02	0.750E 01	0.522E 02	0.214E 02	0.706E 00	0.795E 00
0.700	0.150E 02	0.394E 02	0.747E 01	0.121E 03	0.445E 02	0.593E 00	0.647E 00
0.800	0.341E 02	0.484E 02	0.744E 01	0.382E 03	0.131E 03	0.453E 00	0.475E 00
0.850	0.586E 02	0.658E 02	0.742E 01	0.856E 03	0.289E 03	0.375E 00	0.386E 00
0.900	0.119E 03	0.990E 02	0.739E 01	0.250E 04	0.840E 03	0.305E 00	0.309E 00
0.928	0.209E 03	0.126E 03	0.738E 01	0.570E 04	0.190E 04	0.271E 00	0.273E 00

 6 PAD BEARING L/D = 0.50

 PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.363E 02	0.773E 01	0.715E 01	0.715E 01	0.126E 01	0.126E 01
0.100	0.529E 00	0.364E 02	0.772E 01	0.745E 01	0.723E 01	0.125E 01	0.126E 01
0.200	0.110E 01	0.368E 02	0.772E 01	0.849E 01	0.756E 01	0.120E 01	0.124E 01
0.300	0.181E 01	0.374E 02	0.771E 01	0.107E 02	0.815E 01	0.111E 01	0.121E 01
0.400	0.273E 01	0.384E 02	0.770E 01	0.148E 02	0.895E 01	0.101E 01	0.118E 01
0.500	0.409E 01	0.398E 02	0.769E 01	0.229E 02	0.102E 02	0.893E 00	0.113E 01
0.600	0.638E 01	0.420E 02	0.767E 01	0.410E 02	0.119E 02	0.768E 00	0.107E 01
0.700	0.109E 02	0.459E 02	0.765E 01	0.900E 02	0.143E 02	0.630E 00	0.102E 01
0.800	0.227E 02	0.512E 02	0.763E 01	0.271E 03	0.177E 02	0.469E 00	0.966E 00
0.850	0.375E 02	0.605E 02	0.762E 01	0.597E 03	0.199E 02	0.384E 00	0.937E 00
0.900	0.739E 02	0.777E 02	0.760E 01	0.172E 04	0.224E 02	0.309E 00	0.909E 00
0.928	0.126E 03	0.920E 02	0.759E 01	0.389E 04	0.241E 02	0.273E 00	0.892E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.363E 02	0.773E 01	0.715E 01	0.715E 01	0.126E 01	0.126E 01
0.100	0.615E 00	0.364E 02	0.772E 01	0.756E 01	0.729E 01	0.124E 01	0.126E 01
0.200	0.130E 01	0.369E 02	0.772E 01	0.895E 01	0.775E 01	0.118E 01	0.123E 01
0.300	0.218E 01	0.378E 02	0.771E 01	0.119E 02	0.876E 01	0.107E 01	0.118E 01
0.400	0.343E 01	0.392E 02	0.770E 01	0.177E 02	0.106E 02	0.966E 00	0.110E 01
0.500	0.538E 01	0.413E 02	0.768E 01	0.293E 02	0.145E 02	0.851E 00	0.988E 00
0.600	0.887E 01	0.448E 02	0.766E 01	0.555E 02	0.232E 02	0.734E 00	0.844E 00
0.700	0.161E 02	0.513E 02	0.763E 01	0.127E 03	0.474E 02	0.608E 00	0.674E 00
0.800	0.360E 02	0.604E 02	0.759E 01	0.398E 03	0.137E 03	0.458E 00	0.486E 00
0.850	0.612E 02	0.780E 02	0.757E 01	0.886E 03	0.300E 03	0.378E 00	0.392E 00
0.900	0.123E 03	0.111E 03	0.755E 01	0.257E 04	0.862E 03	0.306E 00	0.311E 00
0.928	0.215E 03	0.139E 03	0.754E 01	0.582E 04	0.194E 04	0.272E 00	0.274E 00

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.147E 03	0.803E 01	0.205E 02	0.205E 02	0.136E 01	0.136E 01
0.100	0.153E 01	0.148E 03	0.803E 01	0.211E 02	0.207E 02	0.135E 01	0.136E 01
0.200	0.318E 01	0.148E 03	0.802E 01	0.234E 02	0.214E 02	0.131E 01	0.135E 01
0.300	0.511E 01	0.149E 03	0.801E 01	0.279E 02	0.226E 02	0.124E 01	0.132E 01
0.400	0.753E 01	0.150E 03	0.800E 01	0.359E 02	0.243E 02	0.115E 01	0.129E 01
0.500	0.108E 02	0.152E 03	0.798E 01	0.510E 02	0.269E 02	0.104E 01	0.125E 01
0.600	0.159E 02	0.155E 03	0.796E 01	0.821E 02	0.305E 02	0.911E 00	0.121E 01
0.700	0.251E 02	0.160E 03	0.793E 01	0.158E 03	0.353E 02	0.759E 00	0.116E 01
0.800	0.462E 02	0.166E 03	0.789E 01	0.413E 03	0.417E 02	0.568E 00	0.110E 01
0.850	0.701E 02	0.176E 03	0.787E 01	0.834E 03	0.458E 02	0.458E 00	0.108E 01
0.900	0.123E 03	0.195E 03	0.784E 01	0.221E 04	0.506E 02	0.353E 00	0.105E 01
0.928	0.195E 03	0.210E 03	0.782E 01	0.477E 04	0.538E 02	0.300E 00	0.103E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.147E 03	0.803E 01	0.205E 02	0.205E 02	0.136E 01	0.136E 01
0.100	0.178E 01	0.148E 03	0.803E 01	0.214E 02	0.208E 02	0.135E 01	0.136E 01
0.200	0.373E 01	0.148E 03	0.802E 01	0.243E 02	0.218E 02	0.129E 01	0.134E 01
0.300	0.613E 01	0.150E 03	0.801E 01	0.305E 02	0.238E 02	0.121E 01	0.130E 01
0.400	0.930E 01	0.152E 03	0.799E 01	0.417E 02	0.276E 02	0.110E 01	0.123E 01
0.500	0.139E 02	0.155E 03	0.796E 01	0.631E 02	0.347E 02	0.994E 00	0.114E 01
0.600	0.215E 02	0.159E 03	0.793E 01	0.107E 03	0.496E 02	0.869E 00	0.100E 01
0.700	0.360E 02	0.167E 03	0.789E 01	0.219E 03	0.870E 02	0.727E 00	0.828E 00
0.800	0.708E 02	0.176E 03	0.783E 01	0.598E 03	0.213E 03	0.549E 00	0.601E 00
0.850	0.111E 03	0.195E 03	0.780E 01	0.122E 04	0.423E 03	0.446E 00	0.476E 00
0.900	0.202E 03	0.231E 03	0.776E 01	0.329E 04	0.111E 04	0.348E 00	0.360E 00
0.928	0.326E 03	0.261E 03	0.774E 01	0.713E 04	0.239E 04	0.297E 00	0.303E 00

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.246E 03	0.808E 01	0.325E 02	0.325E 02	0.138E 01	0.138E 01
0.100	0.244E 01	0.246E 03	0.807E 01	0.335E 02	0.328E 02	0.137E 01	0.137E 01
0.200	0.505E 01	0.247E 03	0.807E 01	0.369E 02	0.338E 02	0.133E 01	0.136E 01
0.300	0.810E 01	0.248E 03	0.806E 01	0.437E 02	0.357E 02	0.126E 01	0.134E 01
0.400	0.118E 02	0.250E 03	0.804E 01	0.557E 02	0.384E 02	0.117E 01	0.131E 01
0.500	0.170E 02	0.253E 03	0.802E 01	0.779E 02	0.422E 02	0.106E 01	0.127E 01
0.600	0.249E 02	0.257E 03	0.800E 01	0.122E 03	0.476E 02	0.937E 00	0.123E 01
0.700	0.387E 02	0.264E 03	0.797E 01	0.229E 03	0.547E 02	0.789E 00	0.118E 01
0.800	0.699E 02	0.272E 03	0.793E 01	0.571E 03	0.643E 02	0.600E 00	0.113E 01
0.850	0.104E 03	0.285E 03	0.791E 01	0.110E 04	0.704E 02	0.493E 00	0.110E 01
0.900	0.181E 03	0.307E 03	0.788E 01	0.273E 04	0.775E 02	0.389E 00	0.107E 01
0.928	0.281E 03	0.325E 03	0.786E 01	0.564E 04	0.821E 02	0.333E 00	0.105E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.246E 03	0.808E 01	0.325E 02	0.325E 02	0.138E 01	0.138E 01
0.100	0.283E 01	0.246E 03	0.807E 01	0.338E 02	0.330E 02	0.136E 01	0.137E 01
0.200	0.593E 01	0.247E 03	0.807E 01	0.384E 02	0.345E 02	0.131E 01	0.135E 01
0.300	0.971E 01	0.249E 03	0.805E 01	0.476E 02	0.376E 02	0.123E 01	0.131E 01
0.400	0.146E 02	0.252E 03	0.803E 01	0.643E 02	0.431E 02	0.113E 01	0.125E 01
0.500	0.218E 02	0.256E 03	0.801E 01	0.958E 02	0.536E 02	0.101E 01	0.116E 01
0.600	0.334E 02	0.262E 03	0.797E 01	0.160E 03	0.751E 02	0.893E 00	0.103E 01
0.700	0.551E 02	0.273E 03	0.793E 01	0.317E 03	0.127E 03	0.755E 00	0.862E 00
0.800	0.106E 03	0.287E 03	0.787E 01	0.825E 03	0.296E 03	0.579E 00	0.638E 00
0.850	0.165E 03	0.310E 03	0.784E 01	0.161E 04	0.561E 03	0.480E 00	0.515E 00
0.900	0.295E 03	0.352E 03	0.779E 01	0.406E 04	0.137E 04	0.382E 00	0.398E 00
0.928	0.468E 03	0.388E 03	0.776E 01	0.842E 04	0.283E 04	0.330E 00	0.338E 00

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 30000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.335E 03	0.810E 01	0.432E 02	0.432E 02	0.139E 01	0.139E 01
0.100	0.325E 01	0.335E 03	0.809E 01	0.445E 02	0.436E 02	0.137E 01	0.138E 01
0.200	0.673E 01	0.336E 03	0.809E 01	0.489E 02	0.450E 02	0.133E 01	0.137E 01
0.300	0.107E 02	0.338E 03	0.808E 01	0.578E 02	0.474E 02	0.127E 01	0.135E 01
0.400	0.157E 02	0.340E 03	0.806E 01	0.733E 02	0.508E 02	0.118E 01	0.132E 01
0.500	0.225E 02	0.344E 03	0.805E 01	0.102E 03	0.559E 02	0.107E 01	0.128E 01
0.600	0.329E 02	0.349E 03	0.802E 01	0.159E 03	0.629E 02	0.949E 00	0.124E 01
0.700	0.509E 02	0.357E 03	0.799E 01	0.294E 03	0.721E 02	0.803E 00	0.119E 01
0.800	0.912E 02	0.368E 03	0.795E 01	0.717E 03	0.846E 02	0.615E 00	0.114E 01
0.850	0.135E 03	0.383E 03	0.793E 01	0.135E 04	0.924E 02	0.511E 00	0.111E 01
0.900	0.234E 03	0.409E 03	0.790E 01	0.323E 04	0.101E 03	0.404E 00	0.108E 01
0.928	0.363E 03	0.430E 03	0.788E 01	0.647E 04	0.107E 03	0.345E 00	0.107E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.335E 03	0.810E 01	0.432E 02	0.432E 02	0.139E 01	0.139E 01
0.100	0.376E 01	0.335E 03	0.809E 01	0.450E 02	0.438E 02	0.137E 01	0.138E 01
0.200	0.788E 01	0.336E 03	0.809E 01	0.508E 02	0.458E 02	0.132E 01	0.136E 01
0.300	0.128E 02	0.339E 03	0.807E 01	0.629E 02	0.498E 02	0.124E 01	0.132E 01
0.400	0.194E 02	0.342E 03	0.805E 01	0.846E 02	0.570E 02	0.114E 01	0.126E 01
0.500	0.288E 02	0.347E 03	0.803E 01	0.125E 03	0.705E 02	0.102E 01	0.117E 01
0.600	0.440E 02	0.355E 03	0.799E 01	0.207E 03	0.980E 02	0.905E 00	0.104E 01
0.700	0.723E 02	0.368E 03	0.795E 01	0.405E 03	0.163E 03	0.768E 00	0.877E 00
0.800	0.138E 03	0.386E 03	0.789E 01	0.103E 04	0.373E 03	0.594E 00	0.655E 00
0.850	0.213E 03	0.413E 03	0.785E 01	0.198E 04	0.689E 03	0.497E 00	0.534E 00
0.900	0.381E 03	0.463E 03	0.780E 01	0.479E 04	0.162E 04	0.397E 00	0.415E 00
0.928	0.604E 03	0.505E 03	0.777E 01	0.965E 04	0.324E 04	0.341E 00	0.350E 00

6 PAD BEARING L/D = 1.00

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
LAMINAR FILM

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.246E 02	0.699E 01	0.897E 01	0.897E 01	0.121E 01	0.121E 01
0.100	0.653E 00	0.247E 02	0.699E 01	0.936E 01	0.907E 01	0.120E 01	0.121E 01
0.200	0.137E 01	0.251E 02	0.699E 01	0.107E 02	0.950E 01	0.115E 01	0.119E 01
0.300	0.301E 00	0.258E 02	0.699E 01	0.517E 01	0.102E 02	0.123E 01	0.117E 01
0.400	0.624E 00	0.269E 02	0.698E 01	0.523E 01	0.113E 02	0.120E 01	0.114E 01
0.500	0.976E 00	0.284E 02	0.698E 01	0.556E 01	0.129E 02	0.116E 01	0.110E 01
0.600	0.609E 01	0.308E 02	0.698E 01	0.478E 02	0.262E 01	0.770E 00	0.135E 01
0.700	0.114E 02	0.348E 02	0.697E 01	0.106E 03	0.238E 01	0.649E 00	0.136E 01
0.800	0.253E 02	0.402E 02	0.697E 01	0.317E 03	0.218E 01	0.505E 00	0.136E 01
0.850	0.426E 02	0.498E 02	0.696E 01	0.688E 03	0.209E 01	0.418E 00	0.136E 01
0.900	0.846E 02	0.676E 02	0.696E 01	0.194E 04	0.201E 01	0.333E 00	0.136E 01
0.928	0.144E 03	0.824E 02	0.696E 01	0.434E 04	0.197E 01	0.287E 00	0.136E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.246E 02	0.699E 01	0.897E 01	0.897E 01	0.121E 01	0.121E 01
0.100	0.759E 00	0.247E 02	0.699E 01	0.950E 01	0.915E 01	0.119E 01	0.120E 01
0.200	0.161E 01	0.253E 02	0.699E 01	0.113E 02	0.975E 01	0.114E 01	0.118E 01
0.300	0.649E 00	0.262E 02	0.699E 01	0.262E 01	0.685E 01	0.135E 01	0.123E 01
0.400	0.534E 00	0.277E 02	0.698E 01	0.218E 01	0.671E 01	0.136E 01	0.123E 01
0.500	0.439E 00	0.301E 02	0.698E 01	0.187E 01	0.661E 01	0.136E 01	0.122E 01
0.600	0.112E 02	0.337E 02	0.697E 01	0.704E 02	0.294E 02	0.759E 00	0.851E 00
0.700	0.203E 02	0.404E 02	0.696E 01	0.158E 03	0.588E 02	0.644E 00	0.702E 00
0.800	0.444E 02	0.495E 02	0.696E 01	0.475E 03	0.164E 03	0.503E 00	0.529E 00
0.850	0.744E 02	0.676E 02	0.695E 01	0.103E 04	0.349E 03	0.417E 00	0.431E 00
0.900	0.147E 03	0.102E 03	0.695E 01	0.291E 04	0.979E 03	0.332E 00	0.338E 00
0.928	0.250E 03	0.131E 03	0.695E 01	0.651E 04	0.217E 04	0.287E 00	0.290E 00

 6 PAD BEARING L/D = 1.00

 PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.372E 02	0.707E 01	0.108E 02	0.108E 02	0.134E 01	0.134E 01
0.100	0.779E 00	0.374E 02	0.707E 01	0.112E 02	0.109E 02	0.132E 01	0.133E 01
0.200	0.162E 01	0.377E 02	0.707E 01	0.126E 02	0.113E 02	0.127E 01	0.131E 01
0.300	0.264E 01	0.384E 02	0.707E 01	0.157E 02	0.121E 02	0.119E 01	0.129E 01
0.400	0.396E 01	0.394E 02	0.707E 01	0.213E 02	0.133E 02	0.108E 01	0.125E 01
0.500	0.587E 01	0.409E 02	0.707E 01	0.322E 02	0.150E 02	0.968E 00	0.121E 01
0.600	0.900E 01	0.432E 02	0.707E 01	0.560E 02	0.175E 02	0.837E 00	0.115E 01
0.700	0.150E 02	0.472E 02	0.706E 01	0.118E 03	0.207E 02	0.690E 00	0.110E 01
0.800	0.301E 02	0.526E 02	0.706E 01	0.340E 03	0.253E 02	0.524E 00	0.104E 01
0.850	0.482E 02	0.622E 02	0.706E 01	0.723E 03	0.282E 02	0.429E 00	0.101E 01
0.900	0.915E 02	0.801E 02	0.705E 01	0.201E 04	0.317E 02	0.338E 00	0.983E 00
0.928	0.152E 03	0.950E 02	0.705E 01	0.445E 04	0.339E 02	0.290E 00	0.966E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.372E 02	0.707E 01	0.108E 02	0.108E 02	0.134E 01	0.134E 01
0.100	0.904E 00	0.374E 02	0.707E 01	0.113E 02	0.109E 02	0.131E 01	0.133E 01
0.200	0.191E 01	0.379E 02	0.707E 01	0.133E 02	0.116E 02	0.125E 01	0.130E 01
0.300	0.318E 01	0.388E 02	0.707E 01	0.175E 02	0.130E 02	0.115E 01	0.125E 01
0.400	0.495E 01	0.402E 02	0.707E 01	0.253E 02	0.156E 02	0.104E 01	0.117E 01
0.500	0.767E 01	0.425E 02	0.707E 01	0.408E 02	0.208E 02	0.923E 00	0.106E 01
0.600	0.124E 02	0.460E 02	0.706E 01	0.752E 02	0.322E 02	0.800E 00	0.920E 00
0.700	0.220E 02	0.528E 02	0.706E 01	0.167E 03	0.630E 02	0.665E 00	0.742E 00
0.800	0.472E 02	0.620E 02	0.705E 01	0.497E 03	0.173E 03	0.511E 00	0.545E 00
0.850	0.781E 02	0.802E 02	0.705E 01	0.107E 04	0.364E 03	0.421E 00	0.440E 00
0.900	0.152E 03	0.114E 03	0.705E 01	0.300E 04	0.100E 04	0.334E 00	0.341E 00
0.928	0.257E 03	0.144E 03	0.705E 01	0.665E 04	0.222E 04	0.288E 00	0.292E 00

 6 PAD BEARING L/D = 1.00

 PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.152E 03	0.724E 01	0.336E 02	0.336E 02	0.147E 01	0.147E 01
0.100	0.250E 01	0.152E 03	0.724E 01	0.345E 02	0.338E 02	0.146E 01	0.147E 01
0.200	0.517E 01	0.153E 03	0.724E 01	0.377E 02	0.348E 02	0.143E 01	0.146E 01
0.300	0.827E 01	0.154E 03	0.723E 01	0.441E 02	0.366E 02	0.137E 01	0.144E 01
0.400	0.120E 02	0.155E 03	0.723E 01	0.554E 02	0.391E 02	0.129E 01	0.141E 01
0.500	0.172E 02	0.158E 03	0.722E 01	0.759E 02	0.428E 02	0.118E 01	0.138E 01
0.600	0.249E 02	0.161E 03	0.722E 01	0.116E 03	0.479E 02	0.105E 01	0.134E 01
0.700	0.381E 02	0.166E 03	0.721E 01	0.213E 03	0.545E 02	0.897E 00	0.130E 01
0.800	0.667E 02	0.172E 03	0.719E 01	0.517E 03	0.635E 02	0.684E 00	0.125E 01
0.850	0.974E 02	0.183E 03	0.719E 01	0.100E 04	0.691E 02	0.549E 00	0.122E 01
0.900	0.163E 03	0.203E 03	0.718E 01	0.255E 04	0.757E 02	0.409E 00	0.119E 01
0.928	0.248E 03	0.219E 03	0.718E 01	0.540E 04	0.799E 02	0.333E 00	0.118E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.152E 03	0.724E 01	0.336E 02	0.336E 02	0.147E 01	0.147E 01
0.100	0.290E 01	0.152E 03	0.724E 01	0.348E 02	0.340E 02	0.146E 01	0.147E 01
0.200	0.606E 01	0.153E 03	0.724E 01	0.391E 02	0.354E 02	0.141E 01	0.145E 01
0.300	0.989E 01	0.155E 03	0.723E 01	0.479E 02	0.383E 02	0.134E 01	0.142E 01
0.400	0.148E 02	0.157E 03	0.723E 01	0.635E 02	0.435E 02	0.125E 01	0.136E 01
0.500	0.219E 02	0.160E 03	0.722E 01	0.925E 02	0.532E 02	0.113E 01	0.128E 01
0.600	0.332E 02	0.165E 03	0.721E 01	0.151E 03	0.728E 02	0.101E 01	0.115E 01
0.700	0.539E 02	0.173E 03	0.719E 01	0.292E 03	0.119E 03	0.860E 00	0.975E 00
0.800	0.100E 03	0.183E 03	0.718E 01	0.744E 03	0.270E 03	0.660E 00	0.727E 00
0.850	0.152E 03	0.203E 03	0.717E 01	0.147E 04	0.512E 03	0.533E 00	0.575E 00
0.900	0.265E 03	0.241E 03	0.716E 01	0.379E 04	0.128E 04	0.402E 00	0.420E 00
0.928	0.410E 03	0.274E 03	0.716E 01	0.807E 04	0.271E 04	0.329E 00	0.339E 00

 6 PAD BEARING L/D = 1.00

 PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.254E 03	0.727E 01	0.541E 02	0.541E 02	0.149E 01	0.149E 01
0.100	0.405E 01	0.254E 03	0.727E 01	0.555E 02	0.545E 02	0.148E 01	0.149E 01
0.200	0.838E 01	0.255E 03	0.727E 01	0.604E 02	0.561E 02	0.145E 01	0.148E 01
0.300	0.133E 02	0.257E 03	0.726E 01	0.703E 02	0.588E 02	0.139E 01	0.146E 01
0.400	0.194E 02	0.259E 03	0.726E 01	0.873E 02	0.626E 02	0.131E 01	0.144E 01
0.500	0.276E 02	0.262E 03	0.725E 01	0.117E 03	0.682E 02	0.122E 01	0.140E 01
0.600	0.398E 02	0.266E 03	0.724E 01	0.177E 03	0.760E 02	0.110E 01	0.137E 01
0.700	0.605E 02	0.274E 03	0.723E 01	0.311E 03	0.861E 02	0.952E 00	0.132E 01
0.800	0.104E 03	0.283E 03	0.722E 01	0.706E 03	0.996E 02	0.759E 00	0.128E 01
0.850	0.151E 03	0.297E 03	0.721E 01	0.129E 04	0.108E 03	0.623E 00	0.125E 01
0.900	0.249E 03	0.321E 03	0.720E 01	0.310E 04	0.117E 03	0.461E 00	0.123E 01
0.928	0.372E 03	0.341E 03	0.719E 01	0.633E 04	0.124E 03	0.361E 00	0.121E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.254E 03	0.727E 01	0.541E 02	0.541E 02	0.149E 01	0.149E 01
0.100	0.470E 01	0.254E 03	0.727E 01	0.561E 02	0.547E 02	0.148E 01	0.149E 01
0.200	0.982E 01	0.256E 03	0.726E 01	0.626E 02	0.569E 02	0.144E 01	0.147E 01
0.300	0.159E 02	0.258E 03	0.726E 01	0.760E 02	0.614E 02	0.137E 01	0.144E 01
0.400	0.239E 02	0.261E 03	0.725E 01	0.996E 02	0.692E 02	0.128E 01	0.139E 01
0.500	0.352E 02	0.265E 03	0.724E 01	0.142E 03	0.836E 02	0.117E 01	0.131E 01
0.600	0.530E 02	0.272E 03	0.723E 01	0.228E 03	0.112E 03	0.105E 01	0.119E 01
0.700	0.852E 02	0.283E 03	0.722E 01	0.424E 03	0.177E 03	0.914E 00	0.103E 01
0.800	0.157E 03	0.298E 03	0.719E 01	0.100E 04	0.372E 03	0.733E 00	0.807E 00
0.850	0.235E 03	0.324E 03	0.718E 01	0.189E 04	0.667E 03	0.605E 00	0.653E 00
0.900	0.402E 03	0.371E 03	0.716E 01	0.460E 04	0.157E 04	0.451E 00	0.475E 00
0.928	0.615E 03	0.410E 03	0.715E 01	0.943E 04	0.318E 04	0.355E 00	0.368E 00

Table C-2 (cont'd)

C-44

6 PAD BEARING L/D = 1.00

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.30
 TURBULENT FILM, REYNOLDS NUMBER = 30000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.346E 03	0.728E 01	0.724E 02	0.724E 02	0.150E 01	0.150E 01
0.100	0.544E 01	0.346E 03	0.728E 01	0.743E 02	0.730E 02	0.149E 01	0.150E 01
0.200	0.112E 02	0.347E 03	0.728E 01	0.808E 02	0.750E 02	0.146E 01	0.149E 01
0.300	0.179E 02	0.349E 03	0.727E 01	0.936E 02	0.786E 02	0.140E 01	0.147E 01
0.400	0.261E 02	0.352E 03	0.727E 01	0.115E 03	0.836E 02	0.133E 01	0.145E 01
0.500	0.370E 02	0.356E 03	0.726E 01	0.155E 03	0.910E 02	0.123E 01	0.142E 01
0.600	0.532E 02	0.361E 03	0.726E 01	0.231E 03	0.101E 03	0.111E 01	0.138E 01
0.700	0.806E 02	0.370E 03	0.724E 01	0.401E 03	0.114E 03	0.976E 00	0.134E 01
0.800	0.138E 03	0.382E 03	0.723E 01	0.869E 03	0.131E 03	0.822E 00	0.129E 01
0.850	0.199E 03	0.399E 03	0.722E 01	0.156E 04	0.142E 03	0.670E 00	0.127E 01
0.900	0.328E 03	0.429E 03	0.720E 01	0.370E 04	0.155E 03	0.436E 00	0.124E 01
0.928	0.490E 03	0.453E 03	0.720E 01	0.759E 04	0.163E 03	0.268E 00	0.123E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.346E 03	0.728E 01	0.724E 02	0.724E 02	0.150E 01	0.150E 01
0.100	0.631E 01	0.346E 03	0.728E 01	0.750E 02	0.733E 02	0.149E 01	0.150E 01
0.200	0.131E 02	0.347E 03	0.728E 01	0.836E 02	0.762E 02	0.145E 01	0.148E 01
0.300	0.214E 02	0.350E 03	0.727E 01	0.101E 03	0.820E 02	0.138E 01	0.145E 01
0.400	0.320E 02	0.354E 03	0.727E 01	0.131E 03	0.922E 02	0.129E 01	0.140E 01
0.500	0.471E 02	0.360E 03	0.726E 01	0.187E 03	0.110E 03	0.119E 01	0.132E 01
0.600	0.707E 02	0.368E 03	0.725E 01	0.297E 03	0.147E 03	0.107E 01	0.121E 01
0.700	0.113E 03	0.383E 03	0.723E 01	0.545E 03	0.229E 03	0.938E 00	0.105E 01
0.800	0.207E 03	0.402E 03	0.721E 01	0.123E 04	0.460E 03	0.797E 00	0.871E 00
0.850	0.310E 03	0.433E 03	0.719E 01	0.227E 04	0.806E 03	0.651E 00	0.702E 00
0.900	0.530E 03	0.489E 03	0.717E 01	0.548E 04	0.187E 04	0.424E 00	0.452E 00
0.928	0.808E 03	0.536E 03	0.715E 01	0.113E 05	0.381E 04	0.261E 00	0.276E 00

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50

LAMINAR FILM

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.337E 02	0.561E 01	0.173E 02	0.173E 02	0.971E 00	0.971E 00
0.100	0.953E 00	0.339E 02	0.561E 01	0.181E 02	0.175E 02	0.959E 00	0.968E 00
0.200	0.199E 01	0.344E 02	0.560E 01	0.209E 02	0.184E 02	0.917E 00	0.954E 00
0.300	0.325E 01	0.353E 02	0.560E 01	0.266E 02	0.199E 02	0.857E 00	0.931E 00
0.400	0.492E 01	0.368E 02	0.560E 01	0.374E 02	0.222E 02	0.780E 00	0.901E 00
0.500	0.742E 01	0.387E 02	0.559E 01	0.581E 02	0.254E 02	0.702E 00	0.869E 00
0.600	0.115E 02	0.423E 02	0.558E 01	0.104E 03	0.300E 02	0.604E 00	0.832E 00
0.700	0.198E 02	0.454E 02	0.557E 01	0.222E 03	0.362E 02	0.495E 00	0.792E 00
0.800	0.403E 02	0.585E 02	0.556E 01	0.668E 03	0.450E 02	0.374E 00	0.750E 00
0.850	0.652E 02	0.708E 02	0.555E 01	0.141E 04	0.504E 02	0.320E 00	0.732E 00
0.900	0.125E 03	0.882E 02	0.554E 01	0.382E 04	0.570E 02	0.273E 00	0.714E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.337E 02	0.561E 01	0.173E 02	0.173E 02	0.971E 00	0.971E 00
0.100	0.110E 01	0.339E 02	0.561E 01	0.184E 02	0.176E 02	0.954E 00	0.965E 00
0.200	0.234E 01	0.347E 02	0.560E 01	0.222E 02	0.189E 02	0.901E 00	0.944E 00
0.300	0.393E 01	0.359E 02	0.560E 01	0.300E 02	0.215E 02	0.832E 00	0.906E 00
0.400	0.619E 01	0.381E 02	0.559E 01	0.450E 02	0.265E 02	0.750E 00	0.846E 00
0.500	0.978E 01	0.407E 02	0.558E 01	0.744E 02	0.363E 02	0.674E 00	0.768E 00
0.600	0.161E 02	0.465E 02	0.557E 01	0.141E 03	0.588E 02	0.580E 00	0.657E 00
0.700	0.294E 02	0.510E 02	0.555E 01	0.315E 03	0.116E 03	0.478E 00	0.527E 00
0.800	0.637E 02	0.747E 02	0.554E 01	0.979E 03	0.338E 03	0.365E 00	0.386E 00
0.850	0.106E 03	0.982E 02	0.553E 01	0.210E 04	0.712E 03	0.315E 00	0.325E 00
0.900	0.209E 03	0.131E 03	0.551E 01	0.570E 04	0.191E 04	0.271E 00	0.275E 00

Table C-2 (cont'd)

C-46

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50

TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.453E 02	0.571E 01	0.193E 02	0.193E 02	0.103E 01	0.103E 01
0.100	0.105E 01	0.454E 02	0.571E 01	0.201E 02	0.195E 02	0.102E 01	0.103E 01
0.200	0.219E 01	0.459E 02	0.571E 01	0.231E 02	0.204E 02	0.974E 00	0.101E 01
0.300	0.357E 01	0.468E 02	0.571E 01	0.290E 02	0.220E 02	0.907E 00	0.990E 00
0.400	0.538E 01	0.483E 02	0.570E 01	0.402E 02	0.244E 02	0.821E 00	0.956E 00
0.500	0.804E 01	0.501E 02	0.569E 01	0.618E 02	0.278E 02	0.731E 00	0.920E 00
0.600	0.124E 02	0.537E 02	0.568E 01	0.110E 03	0.325E 02	0.622E 00	0.879E 00
0.700	0.210E 02	0.569E 02	0.567E 01	0.232E 03	0.390E 02	0.504E 00	0.834E 00
0.800	0.422E 02	0.701E 02	0.566E 01	0.691E 03	0.481E 02	0.378E 00	0.787E 00
0.850	0.676E 02	0.824E 02	0.566E 01	0.145E 04	0.538E 02	0.322E 00	0.766E 00
0.900	0.128E 03	0.100E 03	0.565E 01	0.390E 04	0.606E 02	0.274E 00	0.744E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.453E 02	0.571E 01	0.193E 02	0.193E 02	0.103E 01	0.103E 01
0.100	0.122E 01	0.454E 02	0.571E 01	0.204E 02	0.196E 02	0.101E 01	0.102E 01
0.200	0.257E 01	0.462E 02	0.571E 01	0.244E 02	0.210E 02	0.956E 00	0.100E 01
0.300	0.431E 01	0.474E 02	0.570E 01	0.325E 02	0.237E 02	0.879E 00	0.963E 00
0.400	0.673E 01	0.495E 02	0.570E 01	0.481E 02	0.289E 02	0.787E 00	0.897E 00
0.500	0.105E 02	0.522E 02	0.569E 01	0.788E 02	0.391E 02	0.698E 00	0.809E 00
0.600	0.172E 02	0.580E 02	0.567E 01	0.149E 03	0.625E 02	0.595E 00	0.685E 00
0.700	0.310E 02	0.625E 02	0.566E 01	0.329E 03	0.122E 03	0.485E 00	0.542E 00
0.800	0.663E 02	0.864E 02	0.565E 01	0.101E 04	0.350E 03	0.368E 00	0.392E 00
0.850	0.109E 03	0.110E 03	0.564E 01	0.216E 04	0.733E 03	0.316E 00	0.329E 00
0.900	0.214E 03	0.144E 03	0.563E 01	0.582E 04	0.195E 04	0.272E 00	0.277E 00

Table C-2 (cont'd)

C-47

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.164E 03	0.599E 01	0.475E 02	0.475E 02	0.116E 01	0.116E 01
0.100	0.269E 01	0.164E 03	0.599E 01	0.490E 02	0.479E 02	0.115E 01	0.116E 01
0.200	0.556E 01	0.165E 03	0.599E 01	0.545E 02	0.496E 02	0.111E 01	0.114E 01
0.300	0.889E 01	0.166E 03	0.598E 01	0.650E 02	0.525E 02	0.105E 01	0.112E 01
0.400	0.130E 02	0.168E 03	0.597E 01	0.842E 02	0.569E 02	0.967E 00	0.109E 01
0.500	0.187E 02	0.170E 03	0.596E 01	0.119E 03	0.630E 02	0.873E 00	0.106E 01
0.600	0.275E 02	0.174E 03	0.594E 01	0.192E 03	0.712E 02	0.754E 00	0.102E 01
0.700	0.430E 02	0.178E 03	0.593E 01	0.365E 03	0.825E 02	0.613E 00	0.978E 00
0.800	0.774E 02	0.192E 03	0.590E 01	0.959E 03	0.978E 02	0.451E 00	0.930E 00
0.850	0.115E 03	0.205E 03	0.589E 01	0.190E 04	0.107E 03	0.373E 00	0.907E 00
0.900	0.199E 03	0.224E 03	0.588E 01	0.479E 04	0.118E 03	0.303E 00	0.883E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.164E 03	0.599E 01	0.475E 02	0.475E 02	0.116E 01	0.116E 01
0.100	0.312E 01	0.164E 03	0.599E 01	0.496E 02	0.482E 02	0.114E 01	0.115E 01
0.200	0.651E 01	0.165E 03	0.598E 01	0.569E 02	0.506E 02	0.109E 01	0.113E 01
0.300	0.106E 02	0.167E 03	0.598E 01	0.712E 02	0.554E 02	0.102E 01	0.110E 01
0.400	0.161E 02	0.169E 03	0.596E 01	0.978E 02	0.643E 02	0.930E 00	0.104E 01
0.500	0.240E 02	0.173E 03	0.595E 01	0.147E 03	0.808E 02	0.832E 00	0.962E 00
0.600	0.370E 02	0.179E 03	0.593E 01	0.252E 03	0.116E 03	0.716E 00	0.838E 00
0.700	0.614E 02	0.184E 03	0.590E 01	0.507E 03	0.200E 03	0.583E 00	0.675E 00
0.800	0.118E 03	0.209E 03	0.587E 01	0.139E 04	0.495E 03	0.434E 00	0.481E 00
0.850	0.182E 03	0.235E 03	0.585E 01	0.279E 04	0.964E 03	0.363E 00	0.389E 00
0.900	0.325E 03	0.271E 03	0.583E 01	0.713E 04	0.241E 04	0.298E 00	0.309E 00

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.271E 03	0.604E 01	0.736E 02	0.736E 02	0.118E 01	0.118E 01
0.100	0.421E 01	0.271E 03	0.604E 01	0.759E 02	0.743E 02	0.117E 01	0.118E 01
0.200	0.869E 01	0.272E 03	0.604E 01	0.839E 02	0.767E 02	0.113E 01	0.116E 01
0.300	0.138E 02	0.274E 03	0.603E 01	0.993E 02	0.810E 02	0.107E 01	0.114E 01
0.400	0.202E 02	0.276E 03	0.602E 01	0.127E 03	0.875E 02	0.993E 00	0.111E 01
0.500	0.290E 02	0.279E 03	0.601E 01	0.177E 03	0.964E 02	0.900E 00	0.108E 01
0.600	0.421E 02	0.285E 03	0.599E 01	0.278E 03	0.108E 03	0.785E 00	0.104E 01
0.700	0.652E 02	0.290E 03	0.597E 01	0.512E 03	0.124E 03	0.645E 00	0.100E 01
0.800	0.115E 03	0.308E 03	0.595E 01	0.126E 04	0.146E 03	0.487E 00	0.955E 00
0.850	0.169E 03	0.324E 03	0.593E 01	0.238E 04	0.160E 03	0.409E 00	0.932E 00
0.900	0.286E 03	0.346E 03	0.592E 01	0.567E 04	0.176E 03	0.336E 00	0.909E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.271E 03	0.604E 01	0.736E 02	0.736E 02	0.118E 01	0.118E 01
0.100	0.488E 01	0.271E 03	0.604E 01	0.767E 02	0.747E 02	0.116E 01	0.117E 01
0.200	0.101E 02	0.273E 03	0.603E 01	0.875E 02	0.783E 02	0.111E 01	0.115E 01
0.300	0.165E 02	0.275E 03	0.603E 01	0.108E 03	0.852E 02	0.104E 01	0.112E 01
0.400	0.249E 02	0.278E 03	0.601E 01	0.146E 03	0.980E 02	0.955E 00	0.106E 01
0.500	0.370E 02	0.283E 03	0.600E 01	0.217E 03	0.121E 03	0.859E 00	0.990E 00
0.600	0.564E 02	0.291E 03	0.597E 01	0.363E 03	0.170E 03	0.746E 00	0.872E 00
0.700	0.925E 02	0.299E 03	0.595E 01	0.706E 03	0.284E 03	0.614E 00	0.712E 00
0.800	0.174E 03	0.330E 03	0.591E 01	0.181E 04	0.655E 03	0.469E 00	0.522E 00
0.850	0.266E 03	0.360E 03	0.589E 01	0.349E 04	0.121E 04	0.397E 00	0.429E 00
0.900	0.466E 03	0.403E 03	0.586E 01	0.842E 04	0.285E 04	0.330E 00	0.345E 00

6 PAD BEARING L/D = 0.50

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 30000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.367E 03	0.606E 01	0.970E 02	0.970E 02	0.119E 01	0.119E 01
0.100	0.556E 01	0.367E 03	0.606E 01	0.100E 03	0.979E 02	0.118E 01	0.118E 01
0.200	0.114E 02	0.368E 03	0.606E 01	0.110E 03	0.101E 03	0.114E 01	0.117E 01
0.300	0.182E 02	0.370E 03	0.605E 01	0.130E 03	0.106E 03	0.108E 01	0.115E 01
0.400	0.266E 02	0.373E 03	0.604E 01	0.165E 03	0.114E 03	0.100E 01	0.112E 01
0.500	0.381E 02	0.377E 03	0.603E 01	0.229E 03	0.126E 03	0.912E 00	0.109E 01
0.600	0.552E 02	0.384E 03	0.601E 01	0.356E 03	0.141E 03	0.799E 00	0.105E 01
0.700	0.850E 02	0.391E 03	0.599E 01	0.647E 03	0.162E 03	0.661E 00	0.101E 01
0.800	0.149E 03	0.412E 03	0.597E 01	0.153E 04	0.190E 03	0.505E 00	0.967E 00
0.850	0.219E 03	0.431E 03	0.595E 01	0.283E 04	0.208E 03	0.426E 00	0.944E 00
0.900	0.369E 03	0.457E 03	0.593E 01	0.651E 04	0.228E 03	0.348E 00	0.920E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.367E 03	0.606E 01	0.970E 02	0.970E 02	0.119E 01	0.119E 01
0.100	0.644E 01	0.367E 03	0.606E 01	0.101E 03	0.983E 02	0.117E 01	0.118E 01
0.200	0.134E 02	0.369E 03	0.605E 01	0.114E 03	0.103E 03	0.112E 01	0.116E 01
0.300	0.218E 02	0.371E 03	0.605E 01	0.141E 03	0.111E 03	0.105E 01	0.113E 01
0.400	0.328E 02	0.376E 03	0.603E 01	0.190E 03	0.128E 03	0.967E 00	0.108E 01
0.500	0.486E 02	0.382E 03	0.602E 01	0.280E 03	0.158E 03	0.870E 00	0.100E 01
0.600	0.738E 02	0.392E 03	0.600E 01	0.463E 03	0.219E 03	0.759E 00	0.887E 00
0.700	0.120E 03	0.402E 03	0.597E 01	0.889E 03	0.361E 03	0.628E 00	0.729E 00
0.800	0.226E 03	0.439E 03	0.593E 01	0.221E 04	0.802E 03	0.486E 00	0.543E 00
0.850	0.343E 03	0.474E 03	0.591E 01	0.414E 04	0.144E 04	0.413E 00	0.448E 00
0.900	0.600E 03	0.524E 03	0.588E 01	0.966E 04	0.328E 04	0.342E 00	0.358E 00

Table C-2 (cont'd)

C-50

6 PAD BEARING

L/D = 1.00

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50

LAMINAR FILM

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.346E 02	0.508E 01	0.248E 02	0.248E 02	0.101E 01	0.101E 01
0.100	-0.314E 01	0.347E 02	0.508E 01	0.102E 02	0.111E 02	0.107E 01	0.106E 01
0.200	-0.270E 01	0.353E 02	0.508E 01	0.835E 01	0.974E 01	0.111E 01	0.108E 01
0.300	-0.234E 01	0.362E 02	0.508E 01	0.692E 01	0.858E 01	0.114E 01	0.111E 01
0.400	-0.206E 01	0.378E 02	0.507E 01	0.585E 01	0.766E 01	0.117E 01	0.113E 01
0.500	0.661E 01	0.397E 02	0.507E 01	0.685E 02	0.680E 01	0.740E 00	0.115E 01
0.600	0.114E 02	0.434E 02	0.507E 01	0.124E 03	0.608E 01	0.640E 00	0.117E 01
0.700	0.212E 02	0.466E 02	0.507E 01	0.265E 03	0.548E 01	0.534E 00	0.119E 01
0.800	0.485E 02	0.601E 02	0.506E 01	0.794E 03	0.331E 02	0.412E 00	0.834E 00
0.850	0.810E 02	0.729E 02	0.506E 01	0.166E 04	0.684E 02	0.350E 00	0.786E 00
0.900	0.150E 03	0.910E 02	0.506E 01	0.436E 04	0.769E 02	0.290E 00	0.767E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.346E 02	0.508E 01	0.248E 02	0.248E 02	0.101E 01	0.101E 01
0.100	-0.257E 01	0.348E 02	0.508E 01	0.974E 01	0.198E 02	0.108E 01	0.103E 01
0.200	-0.211E 01	0.356E 02	0.508E 01	0.766E 01	0.112E 02	0.113E 01	0.105E 01
0.300	-0.174E 01	0.368E 02	0.507E 01	0.608E 01	0.106E 02	0.117E 01	0.105E 01
0.400	0.355E 01	0.390E 02	0.507E 01	0.331E 02	0.197E 02	0.834E 00	0.921E 00
0.500	0.133E 02	0.418E 02	0.507E 01	0.993E 02	0.417E 02	0.726E 00	0.789E 00
0.600	0.216E 02	0.478E 02	0.507E 01	0.184E 03	0.700E 02	0.631E 00	0.681E 00
0.700	0.384E 02	0.522E 02	0.506E 01	0.396E 03	0.140E 03	0.529E 00	0.560E 00
0.800	0.804E 02	0.767E 02	0.506E 01	0.117E 04	0.400E 03	0.406E 00	0.419E 00
0.850	0.130E 03	0.101E 03	0.506E 01	0.246E 04	0.828E 03	0.344E 00	0.352E 00
0.900	0.250E 03	0.136E 03	0.506E 01	0.651E 04	0.218E 04	0.287E 00	0.290E 00

Table C-2 (cont'd)

C-51

6 PAD BEARING L/D = 1.00

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 1000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.465E 02	0.513E 01	0.279E 02	0.279E 02	0.110E 01	0.110E 01
0.100	0.149E 01	0.467E 02	0.513E 01	0.290E 02	0.282E 02	0.109E 01	0.110E 01
0.200	0.311E 01	0.472E 02	0.513E 01	0.330E 02	0.294E 02	0.104E 01	0.108E 01
0.300	0.503E 01	0.481E 02	0.513E 01	0.409E 02	0.316E 02	0.979E 00	0.106E 01
0.400	0.753E 01	0.497E 02	0.513E 01	0.556E 02	0.348E 02	0.892E 00	0.102E 01
0.500	0.111E 02	0.516E 02	0.513E 01	0.837E 02	0.393E 02	0.797E 00	0.992E 00
0.600	0.169E 02	0.553E 02	0.513E 01	0.144E 03	0.455E 02	0.683E 00	0.950E 00
0.700	0.279E 02	0.585E 02	0.512E 01	0.294E 03	0.541E 02	0.562E 00	0.905E 00
0.800	0.540E 02	0.721E 02	0.512E 01	0.834E 03	0.660E 02	0.422E 00	0.856E 00
0.850	0.844E 02	0.850E 02	0.512E 01	0.171E 04	0.735E 02	0.354E 00	0.833E 00
0.900	0.155E 03	0.103E 03	0.512E 01	0.446E 04	0.823E 02	0.292E 00	0.810E 00

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.465E 02	0.513E 01	0.279E 02	0.279E 02	0.110E 01	0.110E 01
0.100	0.173E 01	0.467E 02	0.513E 01	0.294E 02	0.284E 02	0.108E 01	0.110E 01
0.200	0.365E 01	0.475E 02	0.513E 01	0.348E 02	0.302E 02	0.102E 01	0.107E 01
0.300	0.606E 01	0.487E 02	0.513E 01	0.455E 02	0.338E 02	0.950E 00	0.103E 01
0.400	0.939E 01	0.509E 02	0.513E 01	0.660E 02	0.406E 02	0.856E 00	0.971E 00
0.500	0.145E 02	0.537E 02	0.513E 01	0.105E 03	0.539E 02	0.761E 00	0.880E 00
0.600	0.232E 02	0.597E 02	0.512E 01	0.194E 03	0.834E 02	0.652E 00	0.753E 00
0.700	0.408E 02	0.642E 02	0.512E 01	0.415E 03	0.157E 03	0.539E 00	0.607E 00
0.800	0.841E 02	0.889E 02	0.512E 01	0.121E 04	0.424E 03	0.410E 00	0.441E 00
0.850	0.135E 03	0.113E 03	0.512E 01	0.253E 04	0.863E 03	0.347E 00	0.363E 00
0.900	0.257E 03	0.149E 03	0.512E 01	0.665E 04	0.223E 04	0.288E 00	0.295E 00

6 PAD BEARING L/D = 1.00

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 10000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.170E 03	0.526E 01	0.734E 02	0.734E 02	0.129E 01	0.129E 01
0.100	0.423E 01	0.170E 03	0.526E 01	0.755E 02	0.740E 02	0.128E 01	0.129E 01
0.200	0.871E 01	0.171E 03	0.526E 01	0.829E 02	0.763E 02	0.124E 01	0.128E 01
0.300	0.138E 02	0.172E 03	0.525E 01	0.970E 02	0.803E 02	0.119E 01	0.126E 01
0.400	0.201E 02	0.174E 03	0.525E 01	0.122E 03	0.862E 02	0.111E 01	0.123E 01
0.500	0.286E 02	0.177E 03	0.525E 01	0.167E 03	0.944E 02	0.101E 01	0.120E 01
0.600	0.411E 02	0.181E 03	0.524E 01	0.258E 03	0.105E 03	0.891E 00	0.116E 01
0.700	0.624E 02	0.185E 03	0.523E 01	0.466E 03	0.120E 03	0.737E 00	0.112E 01
0.800	0.106E 03	0.200E 03	0.523E 01	0.115E 04	0.140E 03	0.540E 00	0.107E 01
0.850	0.153E 03	0.214E 03	0.522E 01	0.221E 04	0.152E 03	0.436E 00	0.105E 01
0.900	0.252E 03	0.234E 03	0.522E 01	0.544E 04	0.167E 03	0.337E 00	0.102E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.170E 03	0.526E 01	0.734E 02	0.734E 02	0.129E 01	0.129E 01
0.100	0.489E 01	0.170E 03	0.526E 01	0.763E 02	0.744E 02	0.128E 01	0.129E 01
0.200	0.101E 02	0.171E 03	0.526E 01	0.862E 02	0.777E 02	0.123E 01	0.127E 01
0.300	0.165E 02	0.173E 03	0.525E 01	0.105E 03	0.841E 02	0.116E 01	0.124E 01
0.400	0.247E 02	0.176E 03	0.525E 01	0.140E 03	0.957E 02	0.107E 01	0.118E 01
0.500	0.364E 02	0.179E 03	0.524E 01	0.204E 03	0.117E 03	0.973E 00	0.110E 01
0.600	0.547E 02	0.186E 03	0.523E 01	0.335E 03	0.160E 03	0.848E 00	0.985E 00
0.700	0.878E 02	0.191E 03	0.522E 01	0.639E 03	0.262E 03	0.701E 00	0.812E 00
0.800	0.160E 03	0.218E 03	0.521E 01	0.165E 04	0.601E 03	0.518E 00	0.581E 00
0.850	0.239E 03	0.245E 03	0.521E 01	0.324E 04	0.113E 04	0.422E 00	0.460E 00
0.900	0.408E 03	0.284E 03	0.520E 01	0.807E 04	0.274E 04	0.330E 00	0.347E 00

6 PAD BEARING L/D = 1.00

PAD ANGLE = 50.00 DEGREES, PIVOT POSITION = 0.55 PRELOAD COEFF. = 0.50
 TURBULENT FILM, REYNOLDS NUMBER = 20000.

LOAD THROUGH PIVOT

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.281E 03	0.528E 01	0.115E 03	0.115E 03	0.132E 01	0.132E 01
0.100	0.677E 01	0.281E 03	0.528E 01	0.119E 03	0.116E 03	0.131E 01	0.132E 01
0.200	0.139E 02	0.282E 03	0.528E 01	0.130E 03	0.120E 03	0.127E 01	0.131E 01
0.300	0.221E 02	0.284E 03	0.528E 01	0.150E 03	0.126E 03	0.122E 01	0.129E 01
0.400	0.321E 02	0.287E 03	0.528E 01	0.187E 03	0.134E 03	0.115E 01	0.126E 01
0.500	0.454E 02	0.290E 03	0.527E 01	0.252E 03	0.146E 03	0.106E 01	0.123E 01
0.600	0.650E 02	0.296E 03	0.526E 01	0.376E 03	0.163E 03	0.947E 00	0.120E 01
0.700	0.977E 02	0.302E 03	0.525E 01	0.647E 03	0.184E 03	0.810E 00	0.115E 01
0.800	0.165E 03	0.321E 03	0.524E 01	0.148E 04	0.213E 03	0.614E 00	0.111E 01
0.850	0.235E 03	0.339E 03	0.524E 01	0.272E 04	0.231E 03	0.494E 00	0.109E 01
0.900	0.380E 03	0.364E 03	0.523E 01	0.637E 04	0.252E 03	0.366E 00	0.106E 01

LOAD MIDWAY BETWEEN PIVOTS

EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.281E 03	0.528E 01	0.115E 03	0.115E 03	0.132E 01	0.132E 01
0.100	0.784E 01	0.281E 03	0.528E 01	0.120E 03	0.117E 03	0.131E 01	0.131E 01
0.200	0.162E 02	0.283E 03	0.528E 01	0.134E 03	0.122E 03	0.126E 01	0.130E 01
0.300	0.263E 02	0.285E 03	0.528E 01	0.163E 03	0.131E 03	0.120E 01	0.127E 01
0.400	0.393E 02	0.289E 03	0.527E 01	0.213E 03	0.148E 03	0.111E 01	0.122E 01
0.500	0.577E 02	0.294E 03	0.527E 01	0.304E 03	0.178E 03	0.102E 01	0.115E 01
0.600	0.862E 02	0.304E 03	0.526E 01	0.483E 03	0.238E 03	0.905E 00	0.104E 01
0.700	0.136E 03	0.311E 03	0.524E 01	0.879E 03	0.370E 03	0.773E 00	0.888E 00
0.800	0.246E 03	0.346E 03	0.522E 01	0.211E 04	0.782E 03	0.588E 00	0.661E 00
0.850	0.364E 03	0.379E 03	0.521E 01	0.396E 04	0.140E 04	0.477E 00	0.523E 00
0.900	0.611E 03	0.426E 03	0.520E 01	0.944E 04	0.322E 04	0.356E 00	0.380E 00

Table C-2 (cont'd)

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6 PAD BEARING				L/D = 1.00			
PAD ANGLE = 50.00 DEGREES, PIVOT POSITION =0.55 PRELOAD COEFF. =0.50							
TURBULENT FILM, REYNOLDS NUMBER = 30000.							
LOAD THROUGH PIVOT							
EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)
0.000	0.000E 00	0.381E 03	0.529E 01	0.153E 03	0.153E 03	0.133E 01	0.133E 01
0.100	0.905E 01	0.381E 03	0.529E 01	0.157E 03	0.155E 03	0.132E 01	0.133E 01
0.200	0.186E 02	0.382E 03	0.529E 01	0.172E 03	0.159E 03	0.129E 01	0.132E 01
0.300	0.295E 02	0.384E 03	0.529E 01	0.198E 03	0.167E 03	0.124E 01	0.130E 01
0.400	0.428E 02	0.388E 03	0.529E 01	0.245E 03	0.178E 03	0.116E 01	0.128E 01
0.500	0.605E 02	0.392E 03	0.528E 01	0.328E 03	0.193E 03	0.108E 01	0.125E 01
0.600	0.863E 02	0.400E 03	0.527E 01	0.484E 03	0.214E 03	0.972E 00	0.121E 01
0.700	0.129E 03	0.407E 03	0.526E 01	0.805E 03	0.242E 03	0.868E 00	0.117E 01
0.800	0.218E 03	0.431E 03	0.525E 01	0.178E 04	0.279E 03	0.654E 00	0.113E 01
0.850	0.310E 03	0.452E 03	0.524E 01	0.325E 04	0.302E 03	0.484E 00	0.111E 01
0.900	0.500E 03	0.482E 03	0.523E 01	0.765E 04	0.328E 03	0.274E 00	0.108E 01

LOAD MIDWAY BETWEEN PIVOTS													
EFF. ECC. RATIO	NORM. LOAD	NORM. FRIC.	NORM. HYD. FLOW	NORM. STIFF. (X)	NORM. STIFF. (Y)	NORM. DAMP. (X)	NORM. DAMP. (Y)						
0.000	0.000E 00	0.381E 03	0.529E 01	0.153E 03	0.153E 03	0.133E 01	0.133E 01						
0.100	0.104E 02	0.381E 03	0.529E 01	0.159E 03	0.155E 03	0.132E 01	0.133E 01						
0.200	0.217E 02	0.383E 03	0.529E 01	0.178E 03	0.162E 03	0.128E 01	0.131E 01						
0.300	0.352E 02	0.386E 03	0.529E 01	0.214E 03	0.174E 03	0.121E 01	0.128E 01						
0.400	0.524E 02	0.391E 03	0.528E 01	0.279E 03	0.195E 03	0.113E 01	0.123E 01						
0.500	0.767E 02	0.397E 03	0.528E 01	0.395E 03	0.234E 03	0.104E 01	0.116E 01						
0.600	0.114E 03	0.409E 03	0.526E 01	0.619E 03	0.308E 03	0.930E 00	0.106E 01						
0.700	0.181E 03	0.419E 03	0.525E 01	0.108E 04	0.464E 03	0.834E 00	0.945E 00						
0.800	0.325E 03	0.461E 03	0.523E 01	0.253E 04	0.948E 03	0.628E 00	0.704E 00						
0.850	0.479E 03	0.501E 03	0.522E 01	0.473E 04	0.168E 04	0.464E 00	0.517E 00						
0.900	0.802E 03	0.557E 03	0.520E 01	0.113E 05	0.387E 04	0.262E 00	0.290E 00

Fig. C-1 Geometry of Tilting Pad Bearing

4 Pad Bearing Zero Preload
(a)

4 Pad Bearing Preload = a
(Preload Coefficient = $\frac{a}{R_P - R_S}$)
(b)

Fig. C-2 Schematic of Tilting Pad Bearing

Load Direction Midway Between Pivots

Load Direction Through Pivot

Fig. C-3 Steady-State Load Directions on Tilting Pad Bearings

REFERENCES

1. Prohl, M.A., "A General Method for Calculating Critical Speeds of Flexible Rotors," Journal of Applied Mechanics, Vol. 67, 1945, pp A142-A148.
2. Lund, J.W., et al, "Rotor-Bearing Dynamics Technology," Final Reports under USAF Contract No. AF 33(615)-1895. U.S.A.F. Reports AFAPL-TR-65-45.
3. Feldman, S., "Unbalance Tolerances and Criteria," Proc. General Electric Balancing Seminar, Vol. IV, April 17, 1958.
4. Federn, K., "Unwuchttoleranzen Rotierender Körper," Werkstatt and Betrieb, Vol. 86, No. 5 (May 1953) pp 243-250.
5. Roark, R.J., "Formulas for Stress and Strain," Third Edition, McGraw Hill Book Co., 1954.
6. Materials for Potassium Lubricated Journal Bearings, G.E. Co. Reports under NASA Contract NAS3-2534.
7. Fan, C.H.T. and J.H. Vohr, "Super-Laminar Flow in Bearings and Seals," Paper 4.1 of the Proceedings on the Technical Meeting on "Super-Laminar Flow Bearings and Seals for Process Fluid Lubricated Turbomachinery," November 1 and 2, 1966. AEC Report No. NYO-3363-6, MIT Report No. 66TR66.
8. Vohr, J.H., "Experimental Study of Super-Laminar Flow Between Non-Concentric Rotating Cylinders," NASA Report CR-749.
9. Ng, C.W., & C.H.T. Pan, "A Linearized Turbulent Lubrication Theory," ASME Trans., Journal of Basic Engineering, Series D, Vol. 87, 1965, pp 675-688.
10. Elrod, H.G., and C.W. Ng, "A Theory for Turbulent Fluid Films and Its Application to Bearings," ASME Paper No. 66-Lub-12.
11. Orcutt, F.K. and E.B. Arwas, "The Steady-State and Dynamic Characteristics of a Full Circular Bearing and a Partial Arc Bearing in the Laminar and Turbulent Flow Regimes," ASME Paper No. 66-LubS-4.
12. Orcutt, F.K., "Steady-State and Dynamic Properties of Journal Bearings in Laminar and Super-Laminar Flow Regimes," I - Tilting Pad Bearings, NASA Report CR-732.
13. Orcutt, F.K. and C.W. Ng, "Steady-State and Dynamic Properties of Journal Bearings in Laminar and Super-Laminar Flow Regimes," II - Full-Floating Ring Bearings, NASA Report CR-733.