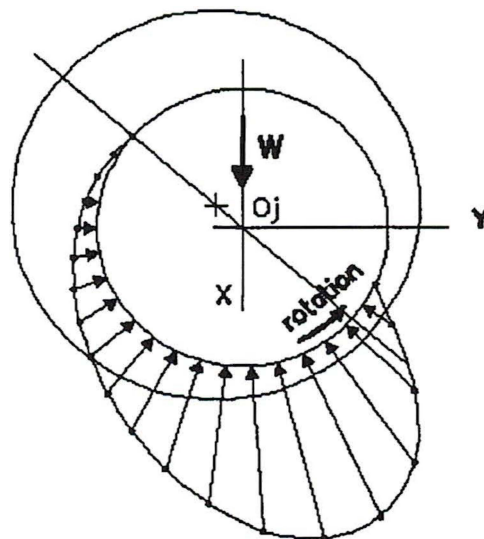# ANALYSIS OF CAROLINA POWER & LIGHT
ROXBORO PLANT UNIT 4ID MOTOR FAN BEARING

Contract No. XXA4000264

by

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ANALYSIS OF UNIT 4ID MOTOR FAN BEARING

I. DISCUSSION AND CONCLUSIONS

1.1 Background and Introduction

The AC motor bearing under study has had a history of high temperature and overheating, particularly in the summertime. The object of this investigation was to evaluate the performance characteristics of the motor bearing with respect to the temperature characteristics and bearing lubrication. The motor in question is an Allis-Chalmers Series 45647, frame H5164 motor. The motor is rated for 4,000 HP and the motor was uprated from 715 RPM to 894 RPM .

The motor rotating element weighs approximately 8,500 lbs. The total motor weight is 23,300 lbs. The motor length is 97 in. The bearing diameter is 7.50 in. and the total bearing length is approximately 7.75 in. The effective length of the bearing for carrying load is 6.0 in. The bearing is a split half bronze sleeve bearing with an effective bearing arc of 120°. There is an oil supply groove of approximately 30° on each side of the bearing. The bearing is currently lubricated by two oil rings.

A number of aspects of the motor bearing system were evaluated. First, the critical speeds of the motor were computed. The motor manufacturer claims a critical speed of 4,875 RPM. The computation of the critical speed showed that it could be as low as 2,000 RPM. This is sufficiently above the operating speed to not represent a problem.

The next step of the study was to analyze the load capacity, bearing stiffness and damping, and temperature rise in the sleeve bearing. Calculations were made for the bearing under various conditions of bearing clearance, load, and speed for the temperature rise and required oil flow requirements. The sleeve bearing is classified as heavily loaded and a positive oil supply system must be provided to the bearing to prevent overheating with conventional lubricants.

A bearing cartridge was supplied to RODYN Vibration for evaluation. A finite element model was generated for the sleeve bearing and a deflection and stress analysis was performed on the sleeve liner. The sleeve is only partially supported by the bearing housing. The center portion is not supported, in order to allow for clearance for the rotation of the two oil rings. The analysis shows that the bearing deforms in the center and may rub on the ends. Examination of the bearing shell indicates rubbing on the bearing ends. The deformation of the bearing shell can also lead to heating.

A fourth aspect of this study was to investigate the use of a synthetic lubricant for the motor bearing. A synthetic motor bearing lubricant has been used at the CP&L Asheville plant with great success. The synthetic lubricant Royal Purple has been used in the Asheville plant for motor bearings. The motor temperatures were reduced by the use of the Royal Purple oil. Information on this lubricant has been obtained from the manufacturer. The Royal Purple synthetic lubricant appears to be superior to other synthetic lubricants, such as Mobil 1, due to the use of a proprietary extreme pressure additive.

1.2 Conclusions

The 4,000 HP motor operating with ring lubrication has insufficient oil supply and is thermally unstable. Under summertime conditions with a warm oil sump, the oil viscosity reduces, causing less lubricant to be fed to the bearings by the oil rings. This results in a higher temperature rise. The temperature will continue to increase with time until unacceptable temperatures are attained. The temperature can be maintained to a minimal temperature rise by supplying at least one gallon of oil per minute to each bearing by a forced oil supply system.

Another promising alternative is that it may be possible to operate the motor bearing without a forced lubrication oil supply by changing the oil lubricant to the Royal Purple synthetic oil with the Synerlec™ extreme pressure additive. The Royal Purple oil appears to micro-polish the bearing surface and ionically bond with the bearing material. This results in a reduced friction in comparison to either a standard petroleum oil or a synthetic oil such as Mobil 1. In addition, the oil slinger rings may be redesigned to add grooving or slots which can increase the amount of oil supplied to the bearing. Although the bearing will operate under boundary lubrication conditions, it is believed that the Royal Purple oil will lead to acceptable temperature levels of operation.

Therefore, it is highly recommended that the ID motor fan bearings be tested with the Royal Purple oil and redesigned slinger rings. Temperature rise vs. time should be recorded with the standard Texaco R&O 68 oil and with the synthetic oil. The motor amperage should be recorded for an indication of bearing friction losses.

II. ANALYSIS OF MOTOR BEARING

2.1 Description of Bearing

The motor is supported by split half-sleeve bearings. The shaft diameter is 7.5 in. and the effective load carrying length of the bearing is 6.0 in. The top half of the bearing has two circumferential slots in it for the operation of the oil slinger rings. Figure 2.1 is a top view of the lower split half bearing. There is a circumferential groove on each end of the bearing. At the bottom of the grooves are oil drain holes.

Figure 2.2 shows a side view of the lower sleeve bearing with the tapered oil feed grooves. The use of slinger rings is common with electric motors up to 2,000 HP. However, there is no information that oil slinger rings may be used successfully in a 4,000 HP motor. Figure 2.3 represents a typical slinger ring application in an electric motor. The ring rides on the top surface of the shaft carrying oil which is deposited to the oil relief groove. The amount of oil delivered is only about 1/10 of the oil required for full film lubrication. The bearing currently operates under marginal boundary lubrication with high heat being generated.

2.2 Bearing Operating Characteristics

2.2.1 Bearing Load Capacity

The ideal fluid film bearing operates by generating a film of oil between the rotating shaft and the bearing. If there is an ample oil supply, then the rotating shaft does not touch the bearing. The oil supply also carries off a substantial amount of the heat generated in the bearing. For a given load, the film thickness between the shaft and the bearing increases with speed.

The fundamental quantity governing the relationship between the bearing load and film thickness is the dimensionless Sommerfeld number. For a particular bearing type with a given length to diameter ratio, the Sommerfeld number establishes a unique relationship between the bearing operating variables of speed, loading, and eccentricity ratio. The Sommerfeld number is given by:

$$S(\epsilon) = \frac{\mu \, N_s \, LD}{W} \left(\frac{R}{C_r}\right)^2$$

Where:

μ = Dynamic Viscosity, $\frac{Lb\text{-}sec}{in^2}$ (Reyns)

N_s = Shaft Speed, RPS

L = Bearing Length, in

D = Bearing Diameter, in

W = Bearing Load, Lb

R = Journal Radius, in
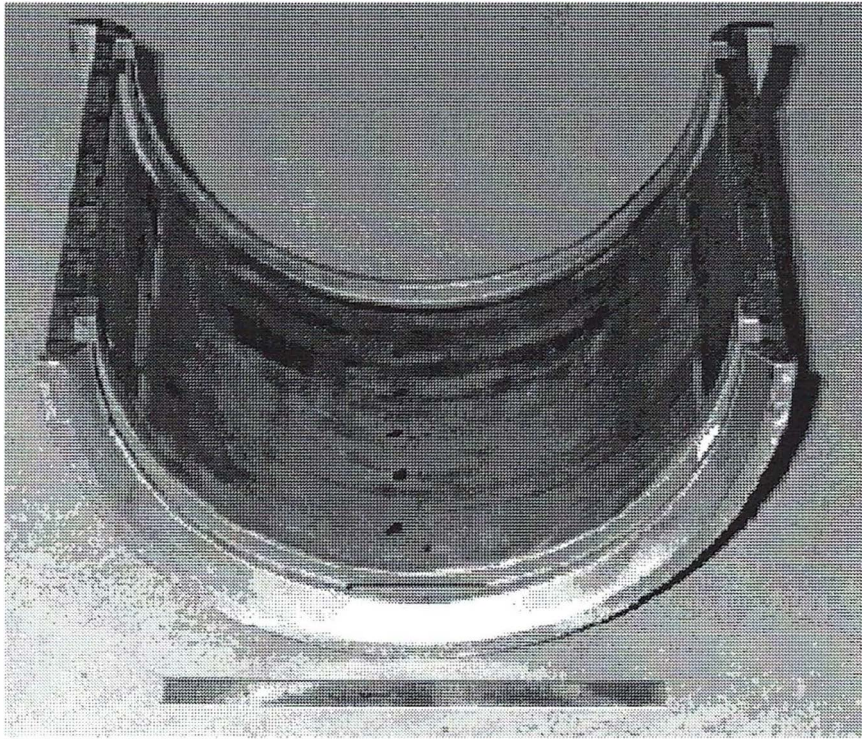
C_r = Bearing Radial Clearance, in

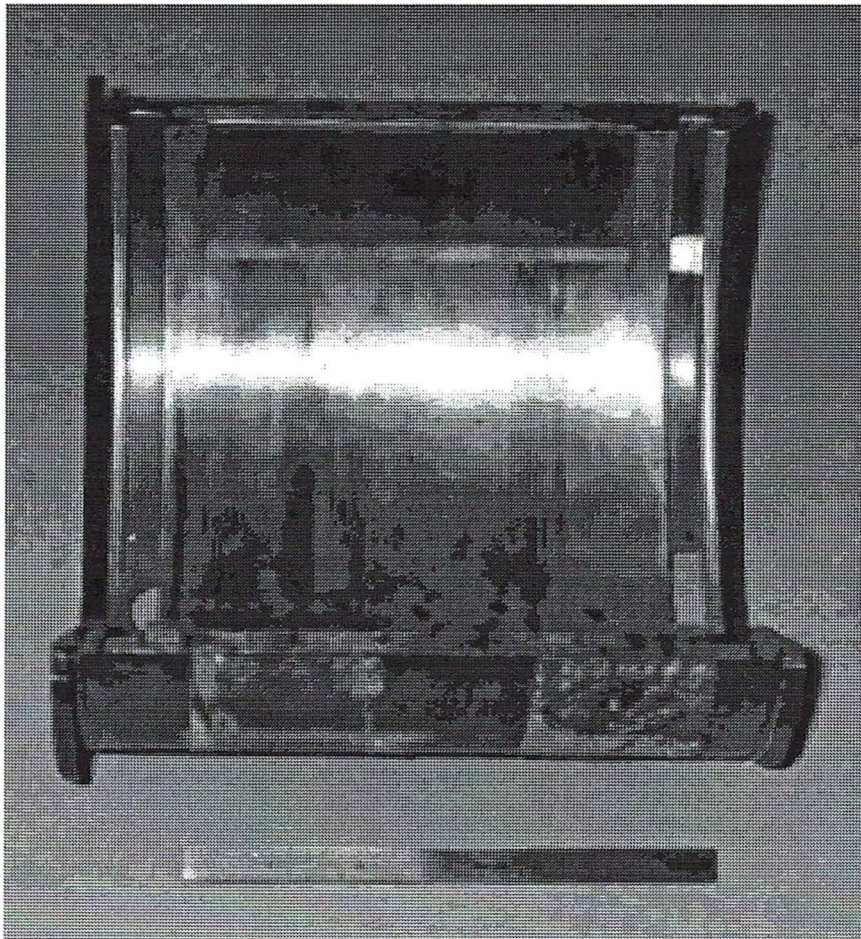**Figure 2.1 - Top View of Motor Bottom Sleeve Bearing**

Figure 2.2 - Side View of Motor Sleve Bearing Showing Oil Feed Groove

A low Sommerfeld number is an indication of a heavily loaded bearing, whereas a high Sommerfeld number represents a lightly loaded bearing. On motor startup, for example, there is metal-to-metal contact of the shaft with the bearing. If there is a proper oil supply to the bearing, then the shaft will lift off and establish an oil film between the rotating shaft and journal. Full shaft lift-off is seldom accomplished using oil rings to supply the bearing .

Figure 2.3 represents a typical industrial 1,700 HP motor bearing with oil lubrication rings provided. The advantage of this design is that the bearing is self-contained and does not require a separate oil supply. Figure 2.4 illustrates the two types of grooving used with ring-oiled bearings. The concept here is that the edge of the lower bearing sleeve will wipe oil off the ring into the bearing recess. The amount of oil supplied by this procedure is barely sufficient to wet the bearing surface. It is completely insufficient to supply a full film of oil.

Figure 2.5 represents the bearing pressure profile at 100 RPM with a radial load of 5,000 lbs. for a bearing diametrical clearance of 0.010 in. (10 mils). For this case, the Sommerfeld number is S = 0.0169 and the dimensionless bearing eccentricity ratio (ϵ) is 0.917. At an eccentricity ratio of 1.0, the shaft is touching the bearing. The minimum film thickness under the shaft is given by:

$$h_{\min} = \frac{Cd}{2}(1-\epsilon) = 5(1-.917) = .42 \ mils$$

Figure 2.6 represents the same bearing with the speed increased from 100 to 900 RPM. The bearing is operating at a lower eccentricity ratio of 0.61, which means that there is a thicker film between the shaft and the bearing. In this case, the minimum film thickness has increased from 0.41 mils to almost 2 mils, assuming full lubrication.

Figure 2.7 represents the bearing dimensionless eccentricity ratio vs. speed for the 10-mil bearing clearance. The bearing is considered to have established full film lubrication and lift-off when the operating eccentricity drops below 0.9. This occurs at about 100 RPM . Figure 2.8 is derived from Figure 2.7 and shows the actual minimum film thickness vs. speed for the case of the 10-mil diametrical clearance. This figure shows that if the bearing diametral clearance is 10 mils, then there will be a well established oil film of 2 mils at running speed.

Figure 2.9 represents the bearing pressure profile and eccentricity at 400 RPM, with the bearing clearance increased from 10 to 20 mils diametral and the oil viscosity reduced from 2 to 1 micro-Reyn. At 400 RPM, it is seen that the minimum thickness is:

$$h_{\min} = \frac{20 \ mils}{2}(1-0.95) = 0.5 \ mils.$$

Figure 2.10 shows the bearing pressure profile at 800 RPM with the 20-mil diametral clearance. The effective lower bearing arc for load capacity is only about 60°.
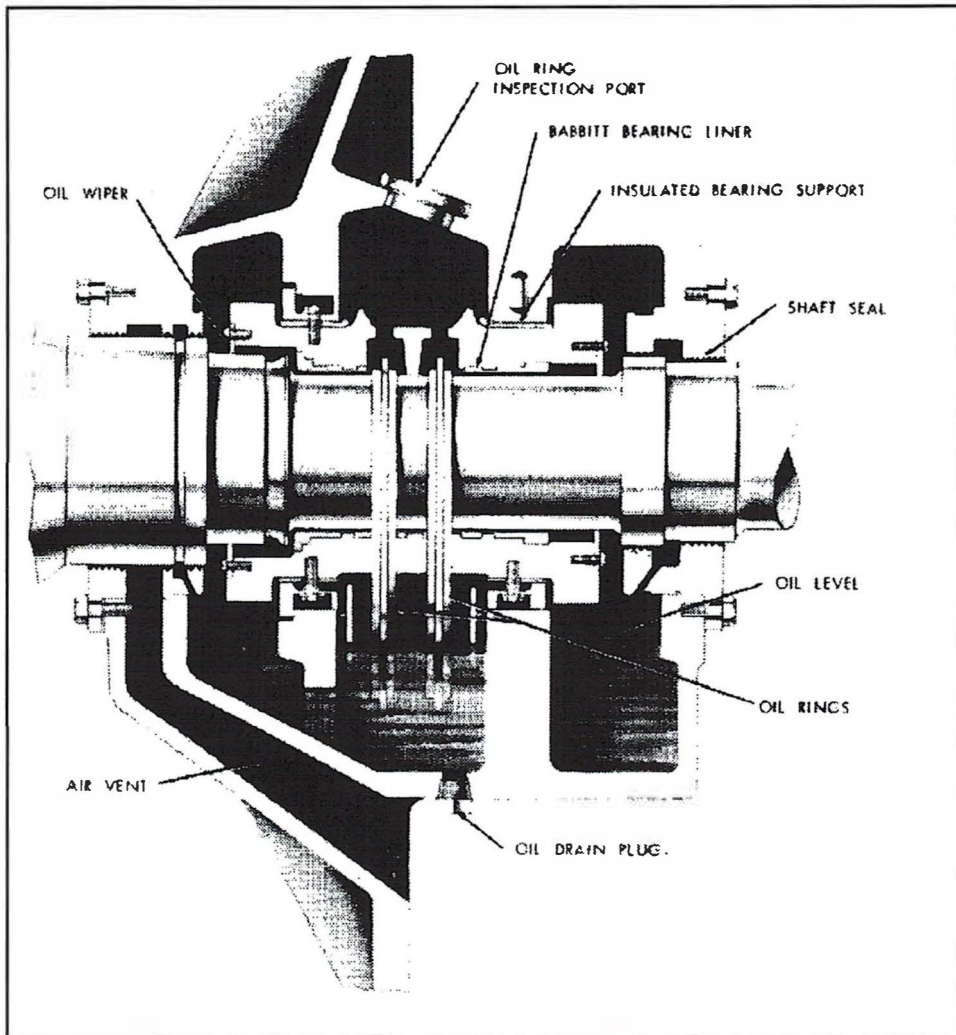
Figure 2.3 Typical Industrail Motor Bearing With Oil Rings
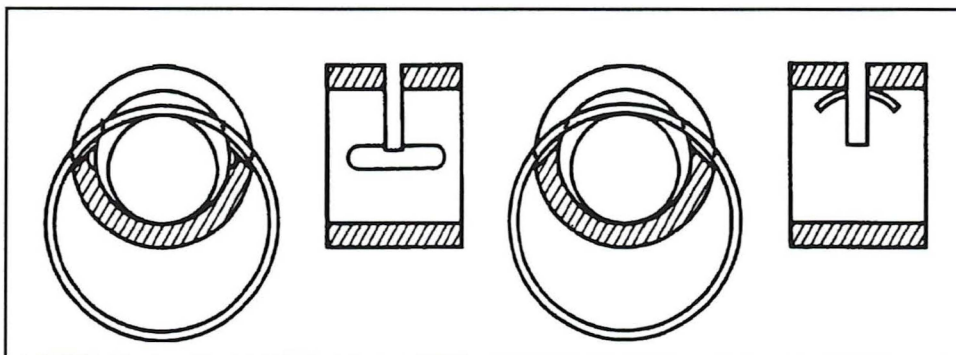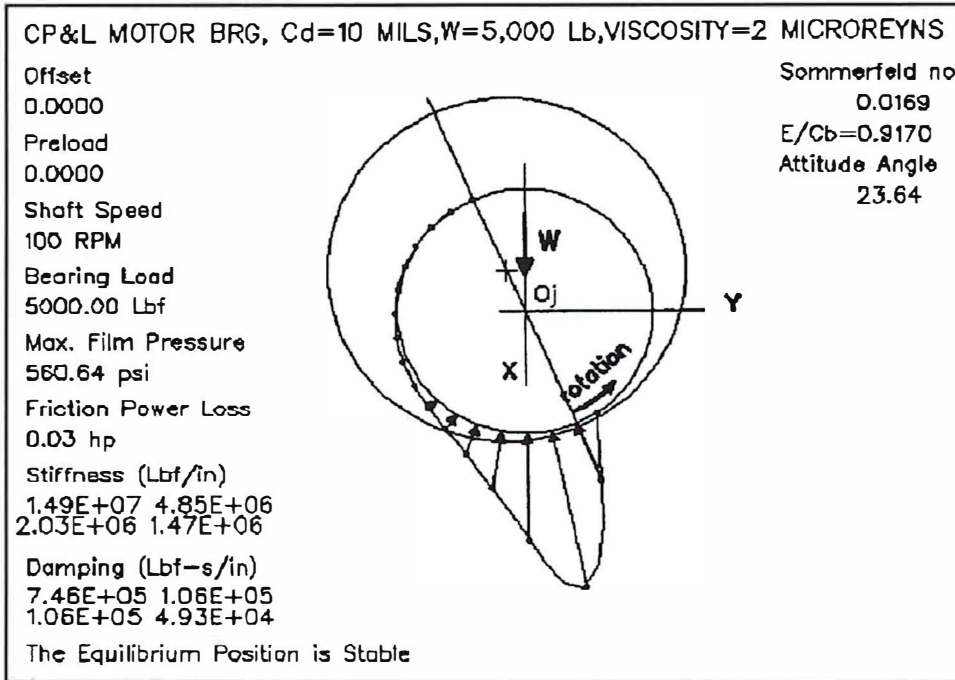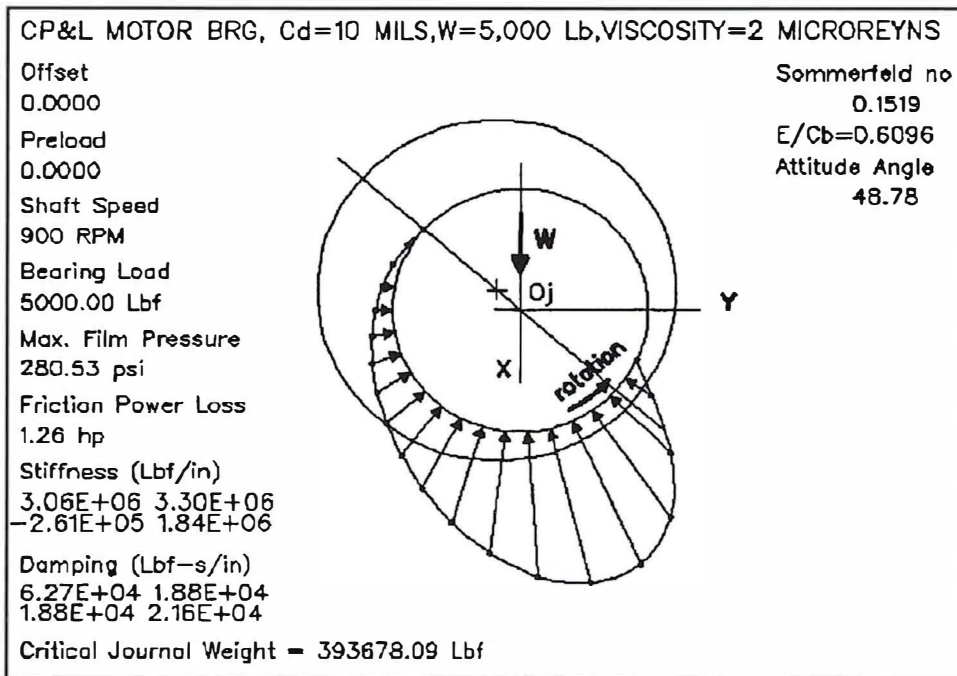**(Fuller)**

Figure 2.4 - Two Types of Grooving Used in Ring-Oiled Bearings
(Kaufman and Boyd)

Figure 2.5 - Motor Bearing Pressure Profile and Eccentricity
At 100 RPM, Cd = 10 Mils

Figure 2.6 - Motor Bearing Pressure Profile and Eccentricity
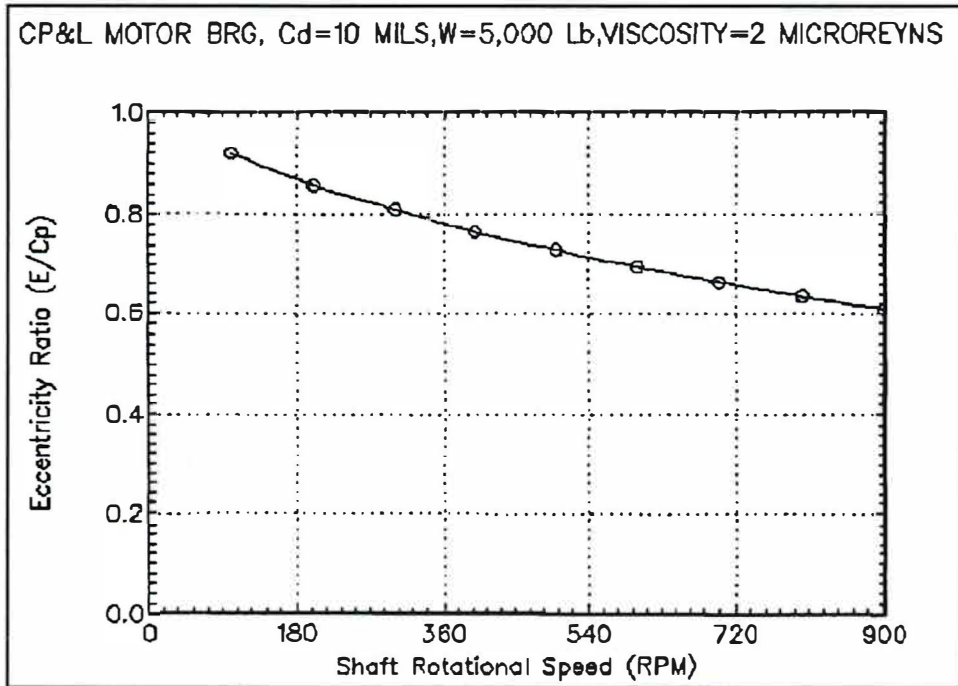**At 900 RPM, Cd = 10 Mils**

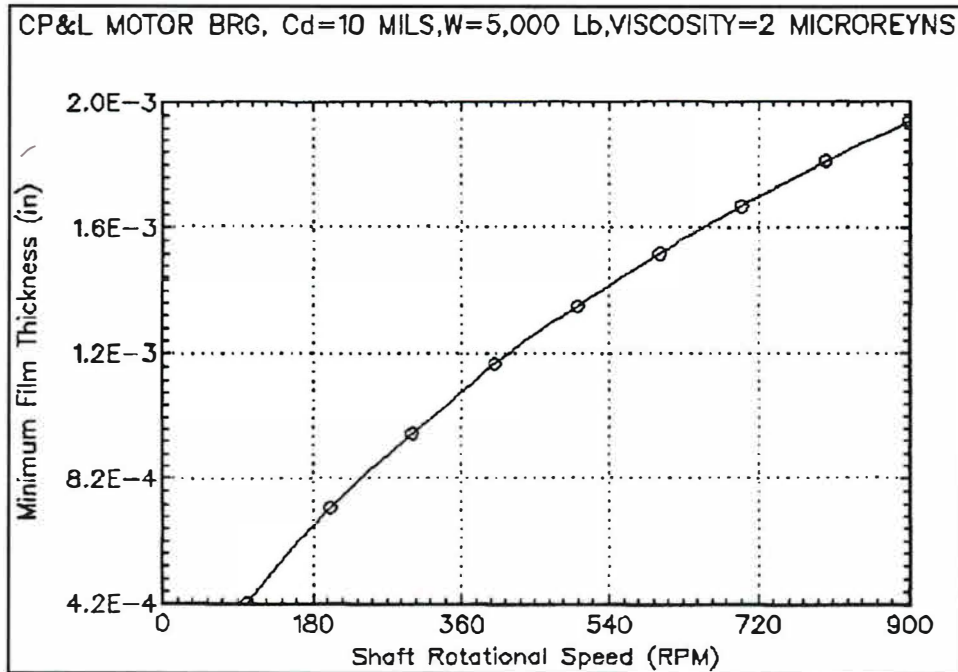**Figure 2.7 - Motor Bearing Dimensionless Eccentricity
Ratio Vs Speed For Cd=10 Mils**

**Figure 2.8 - Motor Bearing Minimum Film Thickness Vs Speed
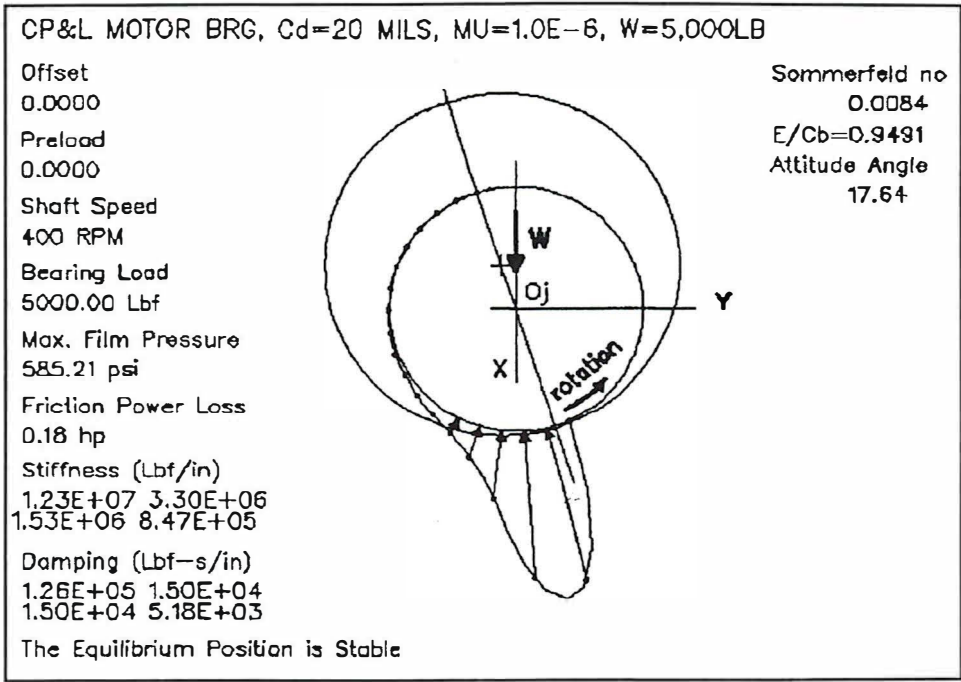For W = 5,000 Lb, Cd = 10 Mils**

CP&L MOTOR BRG, Cd=20 MILS, MU=1.0E−6, W=5,000LB

Offset
0.0000

Preload
0.0000

Shaft Speed
400 RPM

Bearing Load
5000.00 Lbf

Max. Film Pressure
585.21 psi

Friction Power Loss
0.18 hp

Stiffness (Lbf/in)
1.23E+07 3.30E+06
1.53E+06 8.47E+05

Damping (Lbf−s/in)
1.26E+05 1.50E+04
1.50E+04 5.18E+03

The Equilibrium Position is Stable

Sommerfeld no
0.0084
E/Cb=0.9491
Attitude Angle
17.64

Figure 2.9 - Motor Bearing Pressure Profile And Eccentricity At 400 RPM, Cd = 20 mils, Mu = 1.0 micro Reyns

CP&L MOTOR BRG, Cd=20 MILS, MU=1.0E−6, W=5,000LB

Offset
0.0000

Preload
0.0000

Shaft Speed
800 RPM

Bearing Load
5000.00 Lbf

Max. Film Pressure
560.64 psi

Friction Power Loss
0.54 hp

Stiffness (Lbf/in)
7.43E+06 2.43E+06
1.01E+06 7.35E+05

Damping (Lbf−s/in)
4.66E+04 6.62E+03
6.62E+03 3.08E+03

The Equilibrium Position is Stable

Sommerfeld no
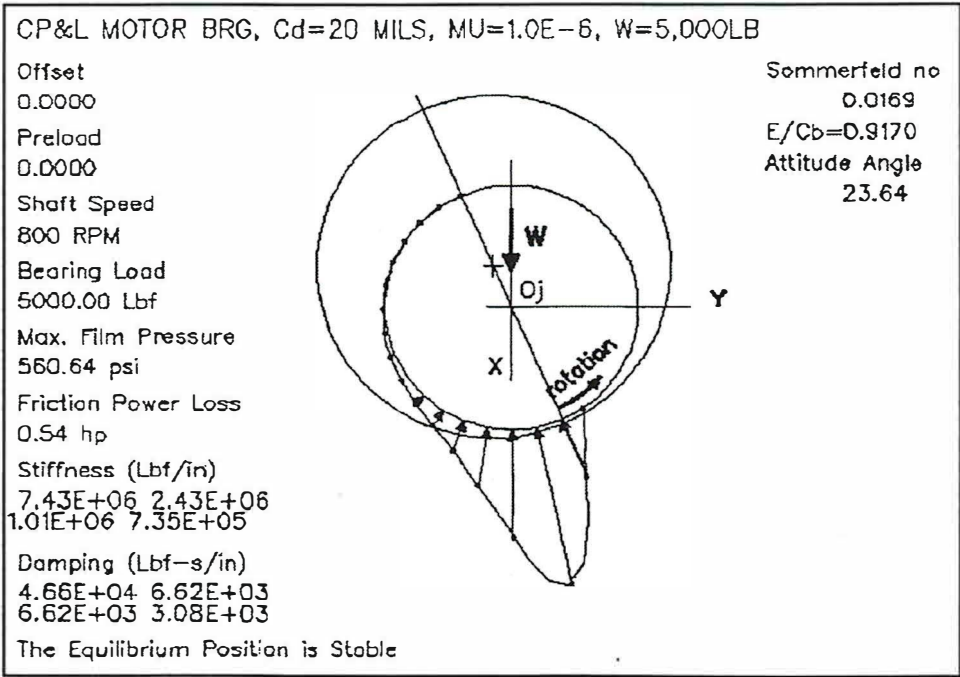0.0169
E/Cb=0.9170
Attitude Angle
23.64

Figure 2.10 - Moter Bearing Pressure Profile And Eccentricity At 800 RPM, Cd = 20 Mils, Mu = 1.0 micro Reyns

2.7

Figure 2.11 represents the minimum film thickness with the increased 20-mil diametral clearance and reduced oil viscosity. At 400 RPM, the minimum film thickness is 0.5 mils and only increases to 0.8 mils at 800 RPM. From this curve, it is apparent that with a larger bearing clearance and reduced oil viscosity, the bearing will be operating at a reduced film thickness. Thus, to maintain the film thickness of the oil film, the diametral clearance should be of the order of 10 mils and a higher viscosity is preferable. Figure 2.12 represents the HP losses of the bearing which increase with speed.
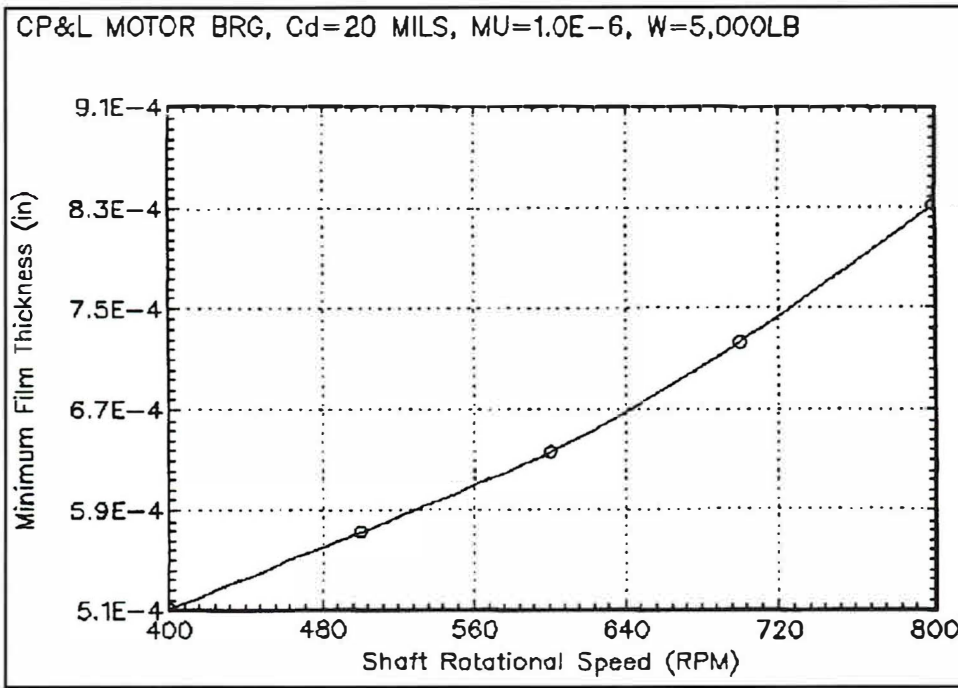
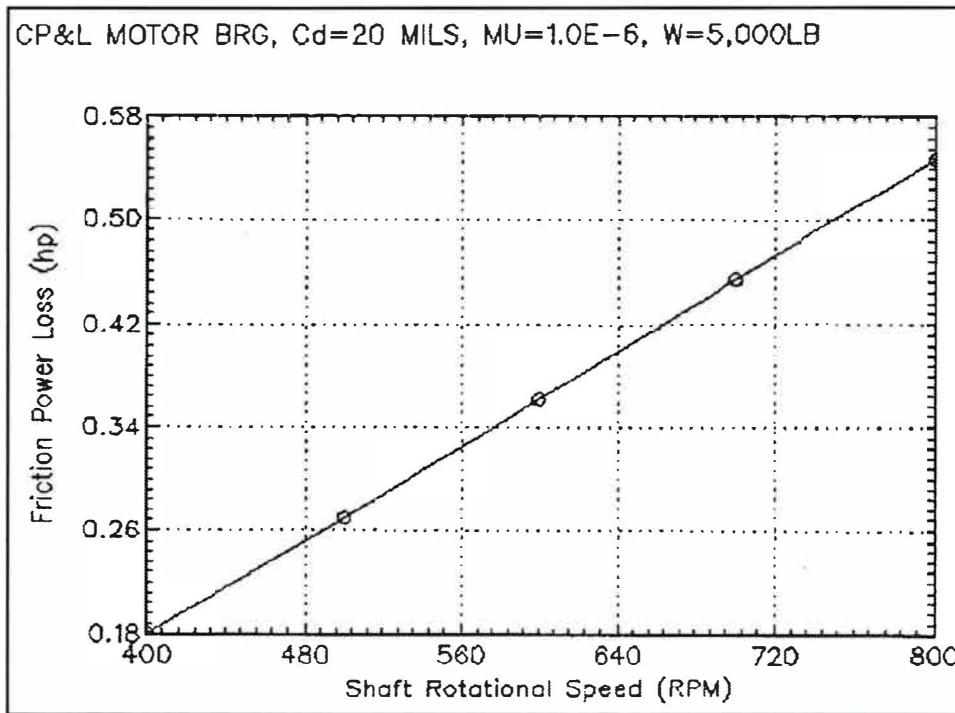**Figure 2.11 - Motor Bearing Minimum Film Thickness Vs Speed For W = 5,000 Lb And Cd = 20 Mils**

Figure 2.12 - Motor Bearing Horsepower Losses for Cd = 20 Mils

2.2.2 Oil Flow Requirements

The bearing oil supply required for full film lubrication may be approximated as follows. It is assumed that the oil is supplied at the split line so that the rotation of the shaft pumps the oil into the bearing clearance area. The shaft surface velocity is given by:

$$V = \frac{900\ RPM}{60} \times 2\pi \times R = 353\ in/\text{sec}$$

for $R = 3.75\ in.$

The total oil flow requirement is a function of the bearing pressure gradient at the inlet, rotational speed, and film thickness as follows:

$$Q = \left(\frac{Vh}{2} + \frac{1}{12\mu} \frac{dP}{dx} \right) \times L \ ; \ in^3/\text{sec}$$

If the inlet pressure gradient is neglected and the film thickness h is approximated by the bearing radial clearance, the approximate expression for oil flow is:

$$Q = \frac{VC_r L}{2} = \frac{RPM \times \pi}{60} RL\,C_r$$

or

$$Q = V_{avg} \times \left(C_r L \right)$$

For a bearing diametrical clearance of $C_d = 10$ mils, the oil flow at 900 RPM is given by:

$$Q = (176\ in/\text{sec}) \times (.005\ in \times 6\ in) = 5.28\ in^3/\text{sec}$$
$$= 316.8\ in^3/\text{min} = \underline{\textbf{1.37}}\ \textbf{\textit{gpm}}$$

If the bearing radial clearance opens up to 10 mils (20 mils diametral), the required oil flow doubles to 2.79 gpm.

Oil ring lubrication has been used with motor bearings with success with motors rated at 1,000 HP and lower. The use of oil rings to provide lubrication has been shown to be adequate for lower rated motors than the 4,000 HP that is currently being used by CP&L. Figure 2.13 (Fuller) shows the quantity of oil that may be delivered by oil rings for a 3- inch bearing. The oil flow is given in in^3/revolutions of the shaft. In all cases, the oil flow per revolution reduces with speeds above 200 RPM. This is because of the slippage of the oil ring on the surface of the shaft. Comparing curves 1 and 3 in Figure 2.13, the grooved ring in medium oil provides considerably more oil than the plain ring. Also, the use of a higher viscosity oil will deliver a greater quantity of oil than will the light oil. This implies that as the oil heats up, the viscosity reduces and, hence, less oil is delivered to the bearing. This could result in a further increase in

temperature. This would cause the bearing to become thermally unstable and the bearing temperature would continue to increase with time.

In Appendix A, the oil viscosity characteristics of various oil lubricants, including a synthetic oil (Royal Purple Maxoil), are presented. The synthetic lubricant has a temperature viscosity index of 120. The viscosity index is an indication of how quickly the oil viscosity reduces with an increase in temperature. For example, a Gulf Oil benzine ring type oil is equal to 0 (the worst); whereas a Pennsylvania long-chain paraffin oil is listed as 100. The Royal Purple Maxoil has a viscosity index of 120, which means that it will not reduce in viscosity as rapidly as the best Pennsylvania type of oil.

Figure 2.14 shows that as the temperature increases from 28°C (82°F) to 78°C (172°F), the oil flow from ring lubrication reduces by a factor of 4. Thus, excessive temperatures rapidly reduce the effectiveness of ring oil lubrication. However, if a synthetic lubricant is used, this effect is reduced because of the superior temperature-viscosity properties of the synthetic.

Figure 2.15 shows the oil flow for two rings. The maximum oil is delivered at 100 RPM. As the speed increases, the oil flow remains constant. As the shaft speed exceeds 600 RPM, the amount of oil delivered actually reduces with speed. This is caused by the slippage of the ring on the shaft with speed. Thus, the increase in the motor speed from the original value of 715 RPM to 894 RPM could cause a reduction in the amount of oil supplied to the bearing. Hence, the increased motor speed produces higher horsepower losses and less oil flow. This would cause higher bearing temperatures to occur. Note that a single ring can deliver at most 0.2 gpm, or 0.4 gpm for two rings. The requirement for full bearing lubrication is 1.3 gpm. This implies boundary lubrication and metal-to-metal contact with ring oil lubrication.

If the Royal Purple oil is used along with grooved lubrication rings, then it may be possible to operate the bearing during the summer without the need for an external oil supply. It is recommended that the bearing be tested with the synthetic oil and grooved oil rings to determine the resulting temperature rise.
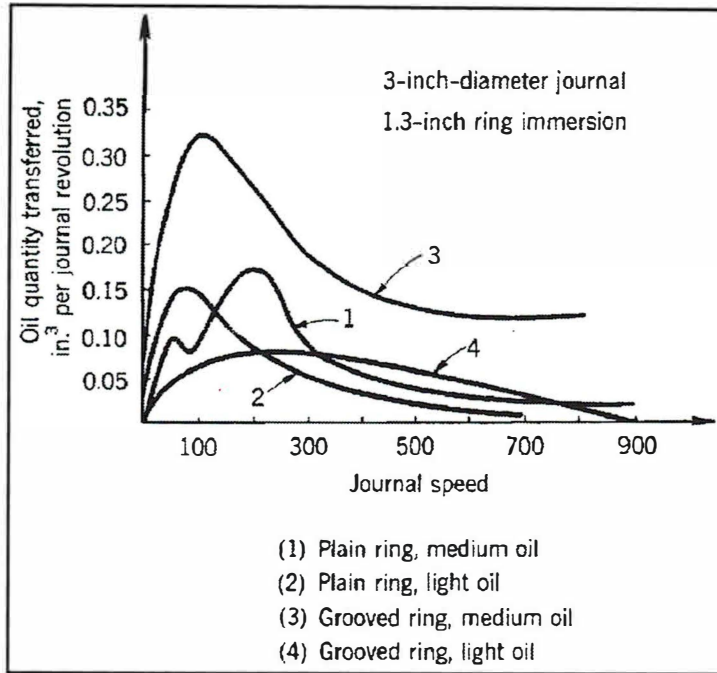
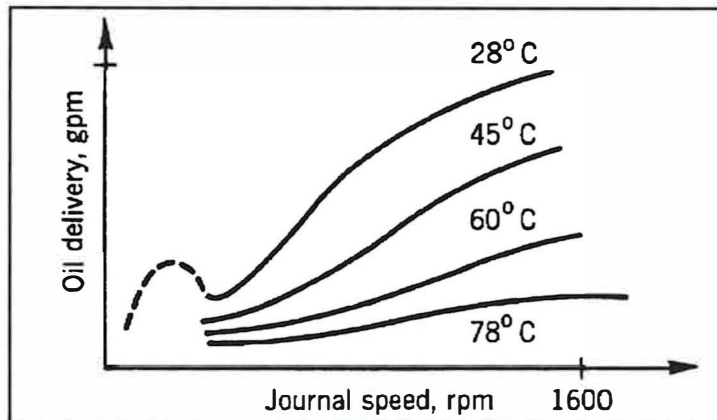**Figure 2.13 - Oil Quantity Transfer Rates For Various Oil-ring Configurations (Fuller)**

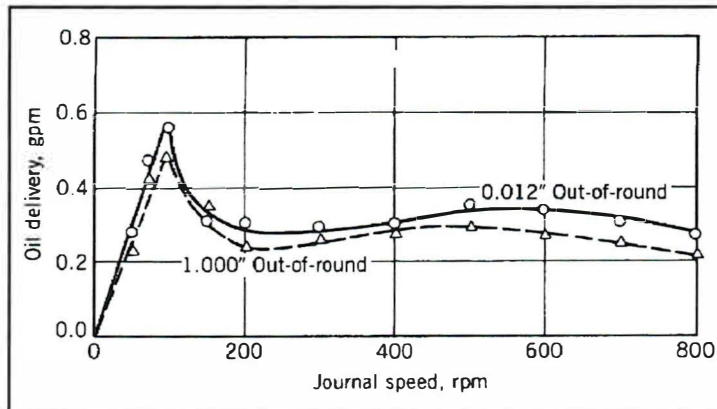**Figure 2.14 - Oil-ring Delivery As Influenced By Viscosity (Fuller).**

Figure 2.15 - Typical Oil Delivery Rate Vs. Shaft Speed (Fuller)

2.12

Table 2.1 represents the operating eccentricity, oil flow, and temperature rise experienced with full lubrication, assuming various bearing operating conditions of clearance and viscosity. The bearing characteristics were computed with the **DYROBES-Becoef** two-dimensional, finite element fluid film bearing program. In the first two cases, as shown in Table 2.1, the oil flow was computed assuming a constant oil viscosity for a full and a 120° partial arc bearing. The highest case of oil flow is with the low oil viscosity of 1.0E-6 Reyns and the large diametral clearance of 20 mils. In this case, the predicted oil flow is 3.7 GPM and the bearing minimum film thickness is .83 mils. It is therefore apparent that the bearing design clearance of 10 mils is preferable, along with a higher oil viscosity. The later cases were computed with a variable viscosity, assuming ISO 68-type oil. Various values of inlet temperature were assumed. The average oil viscosity is computed from the program. The cooler the inlet temperature, for example, the less oil flow that is required. For the 10-mil bearing clearance, the required oil flow increases from .75 GPM at an inlet temperature of 80° to 1.46 GPM at an inlet temperature of 160°. Therefore, an oil supply of 1 GPM to each bearing at an inlet temperature of 120° under normal operating conditions should be satisfactory. In this case, the temperature rise will only be approximately 16°. A worst case condition would be with an inlet temperature of 160°. This would require an oil flow of 1.5 GPM. The required oil flow to the bearing is approximately proportionate to the bearing clearance.

TABLE 2.1 - BEARING OIL FLOW AND MINIMUM FILM THICKNESS FOR VARIOUS BEARING CLEARANCES AND OIL INLET TEMPERATURES $L = 6.0$ in., $D = 7.5$ in.										
BRG TYPE	SPEED N_{RPM}	LOAD Lb	C_{dia} MILS	Q GPM	VISCOSITY REYNS	T_{in} °F	T_{max} °F	ϵ DIM	h_{min} MILS	P_{max} PSI
Full	800	5000	20	3.70	1.0 E-6	—	—	.92	.83	560
120° arc	800	5000	10	0.41	3.0 E-6	—	—	.60	2.0	318
Full	800	5000	10	0.75	6.5 E-6	80°	110°	.35	3.2	230
Full	800	5000	10	0.88	6.2 E-6	100°	122°	.42	2.9	240
Full	800	5000	10	1.04	3.6 E-6	120°	136°	.50	2.5	256
Full	800	5000	10	1.20	2.6 E-6	140°	152°	.58	2.1	273
Full	800	5000	10	1.35	1.9 E-6	160°	170°	.65	1.8	290
Full	894	5000	10	1.46	1.9 E-6	160°	170°	.63	1.9	285
Full	894	5000	15	2.67	1.9 E-6	160°	168°	.77	1.7	349
Full	894	5000	20	3.90	2.0 E-6	160°	167°	.85	1.5	420
Full	900	5000	10	1.42	2.0 E-6	—	—	.61	2.0	281
120° arc	894	5000	10	0.47	1.7 E-6	160°	174°	.61	1.7	305

2.3 Motor Bearing with Boundary Lubrication

The current motor bearing with the oil rings is operating under boundary lubrication conditions. This implies that full film lubrication does not occur between the shaft and the bearing. Under these conditions, as shown by Figures 2.13–2.15, the bearing may become thermally unstable. That is, under constant operating conditions of speed and loading, the bearing temperature increases, causing the oil viscosity to reduce, resulting in a further reduction of oil flow to the bearing. This causes additional heating in the bearing.

In this section, some of the characteristics of boundary lubrication and the use of extreme pressure additives for reducing friction are discussed. The coefficient of friction in a bronze bearing has been investigated by Rippel and other investigators. For example, Figure 2.16 shows that under the conditions of full film lubrication, the coefficient of friction reduces with reduced values of ZN/P, where Z is the viscosity in centipoise, N is the operating speed, and P is the projected bearing pressure. After a minimum value is obtained, the coefficient of friction increases rapidly. This is a region of mixed lubrication. This region is caused by a breakdown of the fluid film, causing the asperities of the surface of the journal and bronze bearing to contact with local smearing.

As the load on the bearing is increased, resulting in lower ZN/P values, a maximum coefficient of friction value is obtained. This value essentially corresponds to metal-to-metal contact. In an attempt to reduce the coefficient of friction during metal-to-metal contact, fatty acid compounds are often added to the lubricant. As shown in Figure 2.17, the fatty acid lubricant molecule adheres to the bearing and journal surface. A chemical bonding is produced, such as iron stearate with the steel in the shaft. Iron stearate is a soap compound. For example, sodium stearate corresponds to "soft soap."

The coefficient of friction is also influenced by temperature. For example, Figure 2.18 shows that for a paraffin oil, the coefficient of friction increases with temperature. This is one of the effects that contributes to the thermal instability of the current motor bearing with oil ring lubrication. As the temperature in the bearing rises, the coefficient of friction also increases, leading to additional heat being generated. The introduction of fatty acid causes a threefold reduction in the coefficient of friction from .3 to .1. The coefficient of friction with fatty acid remains fairly constant up to a critical temperature in which the fatty acid begins to break down due to the temperature rise. The metallic soap is literally "cooked" from the journal surface.

The use of extreme pressure additives can greatly reduce the coefficient of friction at the critical temperature. The three most common materials used as extreme pressure additives are phosphorous, sulfur, and chlorine. Experimental work shows that the thermal decomposition of the additive occurs during operation and a compound is formed with the metal surface, which reduces the friction or surface damage, or both. The phosphorous additives usually act by combining with the metal surface to form metallic phosphides. These films are similar to the
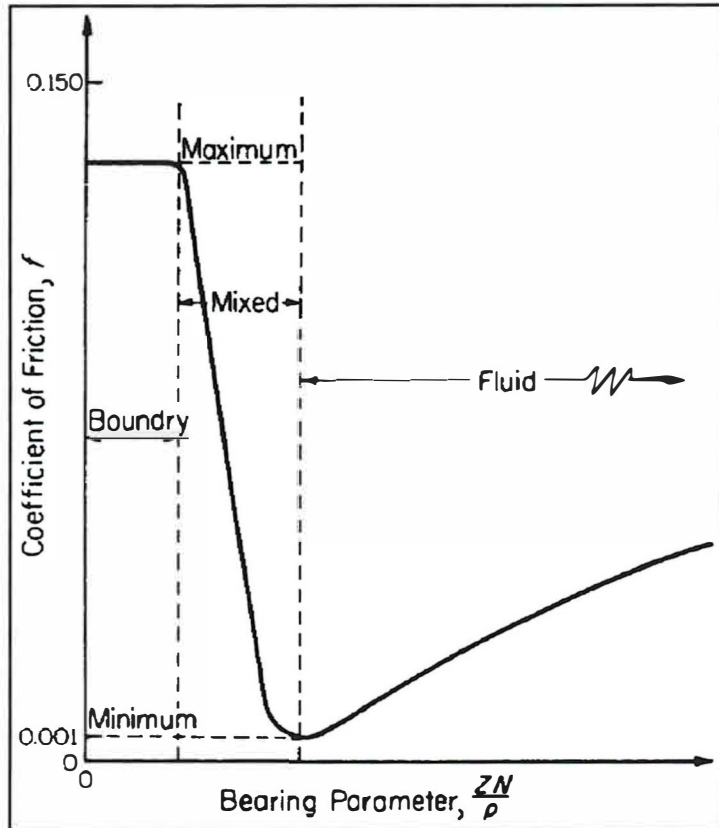
**Figure 2.16 - Variations Of Coefficient Of Friction
For Typical Bronze Sleeve Bearings (Rippel)**

**Figure 2.17 - Fatty-acid Lubricant Molecules Adhering To Bearing
and Journal Surfaces (Rippel)**

Figure 2.18 - Variation Of Coefficients Of Friction With Temperature For Several Lubricants Under Complete Boundary-lubrication Conditions (Rippel)

Figure 2.19 - Comparison Of Oil Film Strengths (Royal Purple Synthetic Lubricants, Inc.)

Figure 2.20 - Comparison of Oil Life (Royal Purple, Inc.)

way in which chloride and sulfide film develop. It has been shown that the phosphide film forms a low melting eutectic with the underlying meta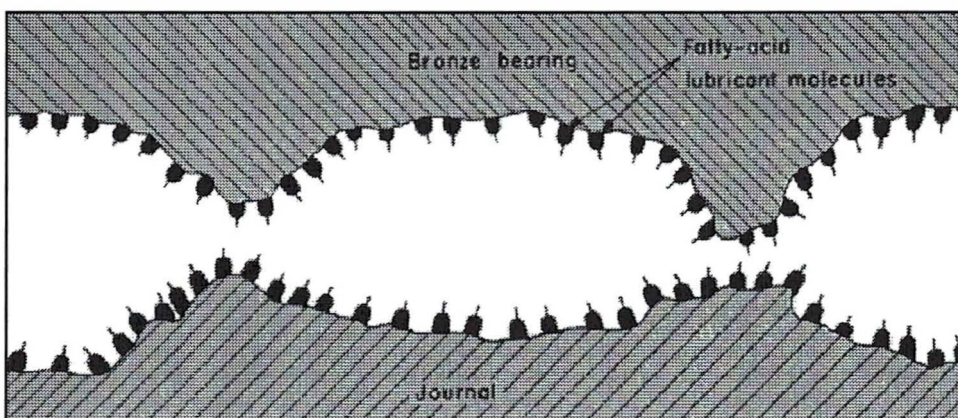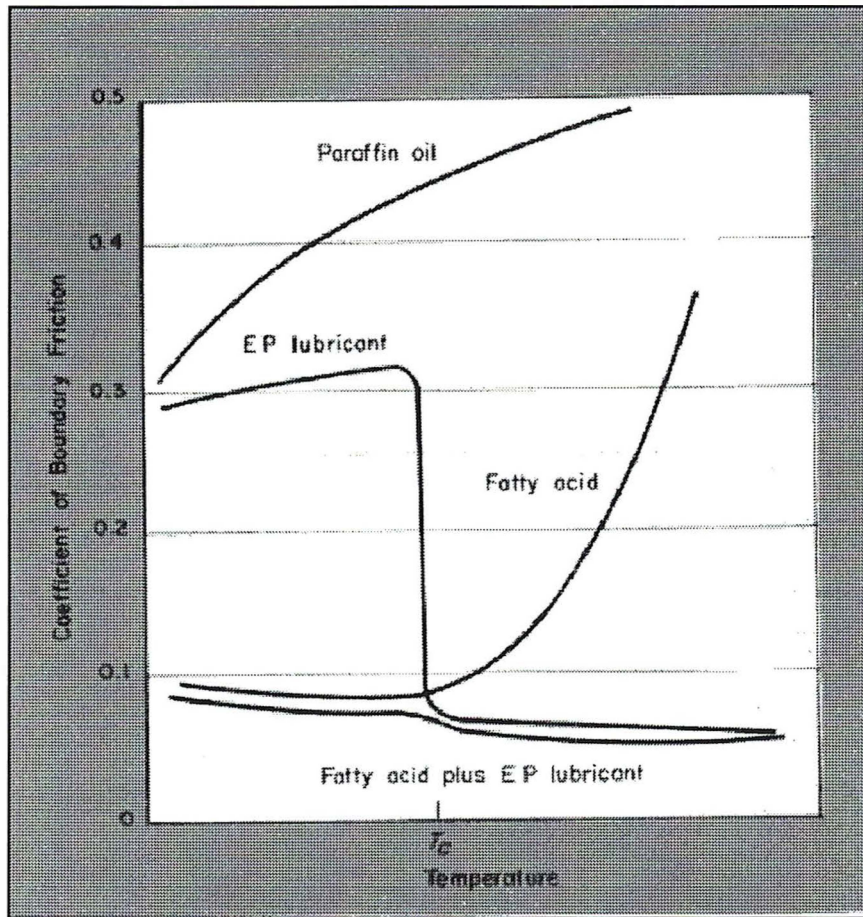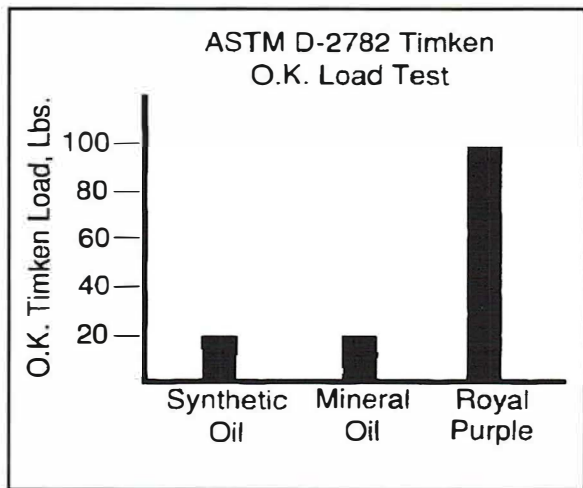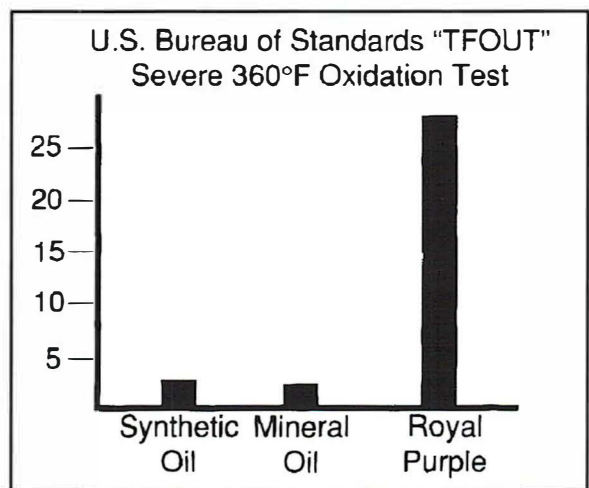l. This material is wiped or polished away by the sliding process to leave a smooth surface capable of providing hydrodynamic lubrication under more extreme conditions. Both chlorine and sulfur additives function by forming a thin film of metallic chloride or sulfide on the surface. If these films are above a critical thickness, all the shearing during sliding takes place within them and, in this way, they are able to prevent heavy surface damage and danger of seizure. They also produce a reduction in friction. The reduction in friction is higher with the chloride films. Moly additives have been used in place of chloride additives because they are non-corrosive.

In Figure 2.18, Rippel shows that the addition of a fatty acid, plus an extreme pressure lubricant, reduces the coefficient of friction over a large temperature range. Without the extreme pressure lubricant, the fatty acid compound quickly disintegrates with high temperatures.

Figure 2.19 represents the Timken load capacity test for three oils. The synthetic oil and mineral oil have similar characteristics and roughly carry 20 lbs. in the ASTM D-2782 load test. The synthetic oil, Royal Purple, however, carries five times the load capacity as either a conventional mineral oil, or synthetic oil such as Mobil 1. The test shows that the conventional synthetic oil is no better than the mineral oil for boundary lubrication. However, the Royal Purple oil has a considerably higher load capacity, due to the addition of proprietary extreme pressure additives. It is believed that the Royal Purple oil, which was developed in Germany, is a result of the synthetic lubrication work dating from World War II. These oils also have a tetrasulfide additive, rather than a chlorine extreme pressure additive, to improve boundary lubrication.

The synthetic oil is referred to as a PAO, which is a Polyalphaolefin oil. This oil is blended with a diester and is very stable over a wide temperature range. The synthetic-based oil must be blended with additives to obtain the high level of performance required. The proprietary additive in Royal Purple is referred to as Synerlec™. The additive has the ability to ionically bond or adhere to bearing materials to reduce the coefficient of friction and greatly increase the oil film strength. The synthetic lubricants with additives excel in high film strength and oxidation stability. The manufacturers of Royal Purple claim to have pioneered the development of non-corrosive, high film strength oils with outstanding water separation properties. In addition, the additive appears to micro-polish the bearing surface, which further reduces the bearing friction. Figure 2.20 is a comparison of the oil life between the synthetic, mineral, and Royal Purple oils. For the oil oxidation test at 360°F, it appears that the Royal Purple has a much higher oxidation life than either the synthetic or the conventional mineral oil. This would make the oil much more suitable for the 4,000 HP motor bearing application with oil ring lubrication.

As previously commented on, the motor bearing with ring lubrication is thermally unstable. One of the problems is the reduction of viscosity encountered with increase in temperature. The rate at which the oil viscosity reduces with an increase in temperature is known as the viscosity index of the oil. As shown in Figure 2.21, an arbitrary value of 0 was assigned to a Gulf basin oil and a value of 100 was assigned to a Pennsylvania-grade oil. The Royal Purple oil has a viscosity index of 120, which means that its viscosity reduces less with temperature than a high-grade Pennsylvania paraffin-type oil. This is a highly desirable characteristic to have for ring oil lubrication.

Figure 2.16 shows that for boundary lubrication, the coefficient of friction would be quite high. Figure 2.22 shows a comparison of a friction test showing synthetic vs. conventional oil. At the start of the test, the synthetic oil had a friction of 17.5% less than the conventional oil. However, after 15 minutes of running, it appeared that the coefficient of friction greatly reduced, such that it was 78% less than that obtained with the conventional oil. Scanning electron microscope photographs of a bearing surface before and after use with the Royal Purple seems to indicate that the surface is "micro-polished" by Synerlec. Photographs of a bearing surface using a premium competitive synthetic oil indicates the surface is galled. The change of the surface texture may account for the dramatic reduction in the dynamic friction after 15 minutes of run, as shown in Figure 2.22.

It is therefore recommended that the motor bearings be tested with the Royal Purple oil and redesigned oil rings with grooving to provide more oil flow to the bearing. The temperature change of the motor bearing should be recorded with conventional oil and with the Royal Purple oil. It is also recommended to record the motor amperage under conventional lubrication and with Royal Purple oil to determine if there appears to be a reduction in friction characteristics of the bearing.
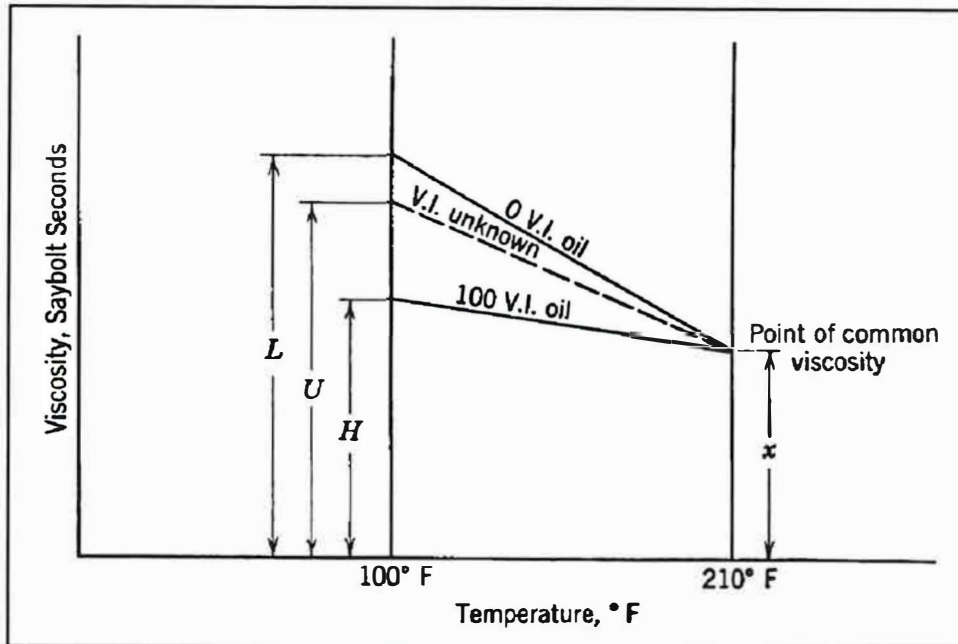
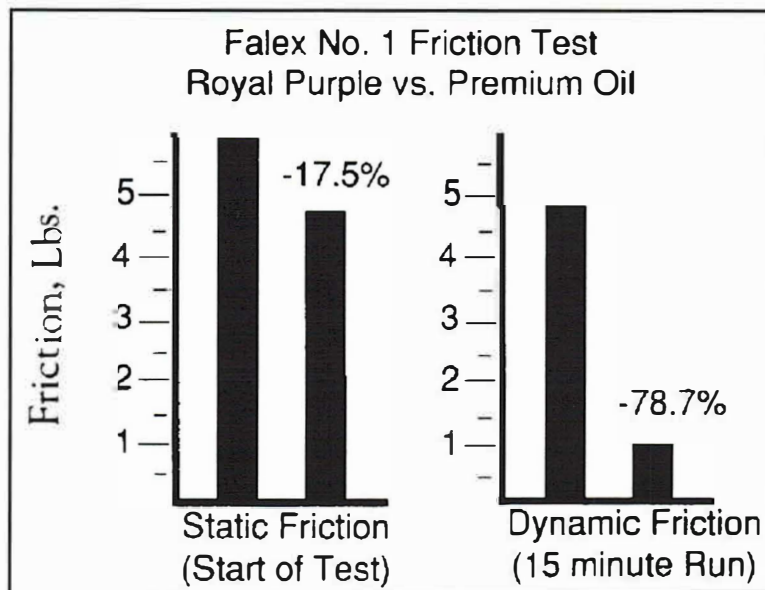**Figure 2.21 - Determination Of Viscosity Index**

**Figure 2.22 - Comparison Of Synthetic Vs.
Conventional Oil In Friction Test**

III. FINITE ELEMENT ANALYSIS OF BEARING CARTRIDGE

A finite element analysis was generated for the bronze bearing cartridge to examine the deflection and stresses in the lower bearing lining due to the hydrodynamic bearing loading. Since the bearing is not fully supported in the center, the center span may deflect under the bearing loading. A three-dimensional shell model of the cartridge was developed. A parabolic load distribution was assumed, acting along the bottom of the bearing. The load distribution is highest in the center of the bearing and tapers off towards the bearing ends.

Figure 3.1 shows the front view of the bearing before and after deformation. Figure 3.2 shows a side view of the bearing deflection. The maximum deflection in the center can be 1.8 mils. The bearing analysis program showed that under conditions of low viscosity and large bearing clearances, the maximum film thickness at running speed could be as low as .8 mils. Table 2.1 shows, for example, that under high inlet temperature of 160°, the maximum film thickness is only 1.8 mils. This corresponds to the deflection value under the pad. Therefore, even under conditions of good lubrication, contacting of the shaft with the ends of the bearing may be expected.

Figure 3.3 shows the top view of the bearing. There is a slight necking down in the center plane of the bearing. Figure 3.4A and 3.4B represent the Von Mises stress contours looking down on the bearing. In the center of the bearing, the stresses can be as high as 12,000 PSI.

Figure 3.5A and 3.5B show the translational deflection contours in the bearing. The oil pressure can cause a dimple in the center portion of the bearing cartridge. This can lead to local wearing on the ends of the bearing. Inspection of the bearing cartridge indicates wear marks at the ends. It is therefore recommended that the synthetic oil, Royal Purple, be used even with the forced oil supply supplied by the pump at one to two gallons per bearing. There will exist in the bearing conditions of boundary lubrication at the bearing ends. The use of the Royal Purple oil should result in a reduction of temperature with the oil pump, as well as with ring lubrication.
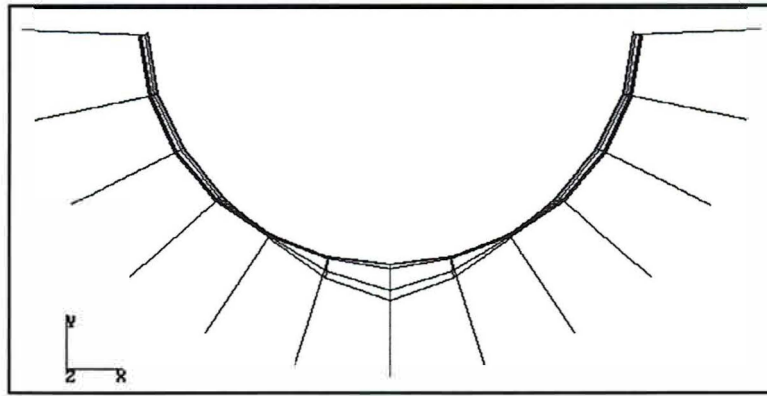
Figure 3.1 - Front View Of Bearing Deflection
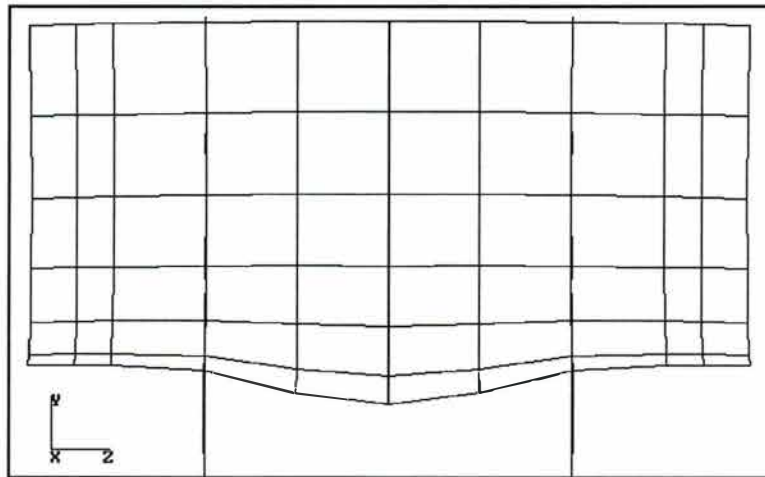
Figure 3.2 - Side View Of Bearing Deflection
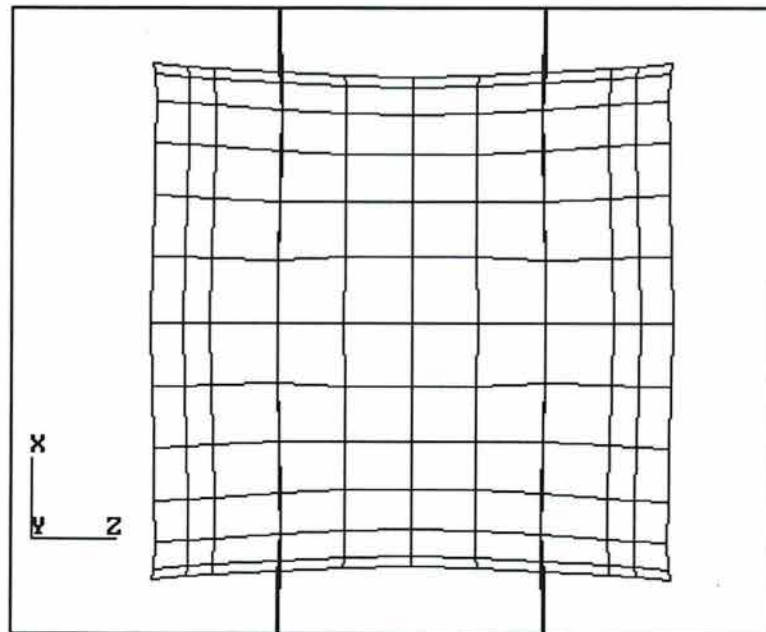
Figure 3.3 - Top View Of Bearing Deflection
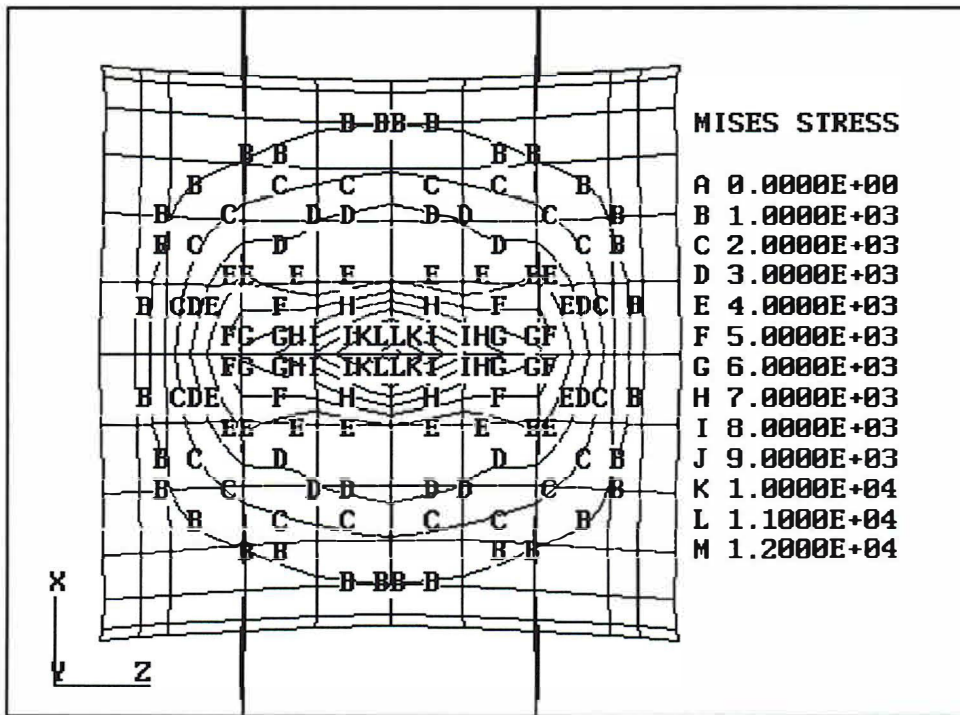
Figure 3.4a - Contour Plot of Von Mises Stress From The Bearing Top View

MISES STRESS

A 0.0000E+00
B 1.0000E+03
C 2.0000E+03
D 3.0000E+03
E 4.0000E+03
F 5.0000E+03
G 6.0000E+03
H 7.0000E+03
I 8.0000E+03
J 9.0000E+03
K 1.0000E+04
L 1.1000E+04
M 1.2000E+04
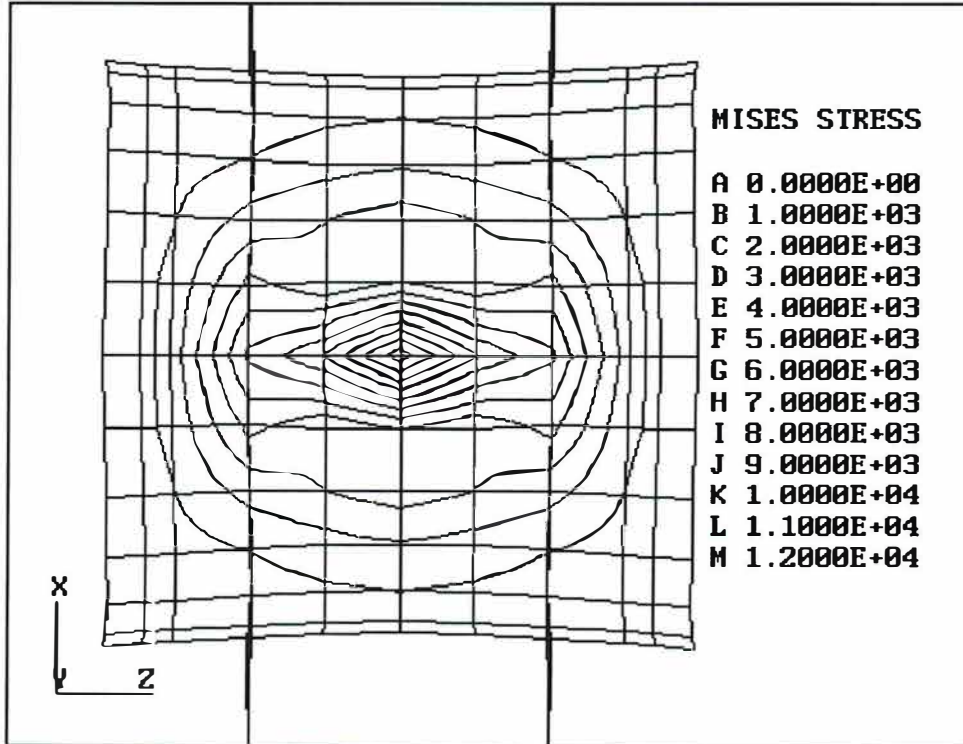
Figure 3.4b - Von Mises Stress With Contour Labels Removed

MISES STRESS

A 0.0000E+00
B 1.0000E+03
C 2.0000E+03
D 3.0000E+03
E 4.0000E+03
F 5.0000E+03
G 6.0000E+03
H 7.0000E+03
I 8.0000E+03
J 9.0000E+03
K 1.0000E+04
L 1.1000E+04
M 1.2000E+04

3.3

TRANS. DEFL. Y

A—1.8000E-03
B—1.6000E-03
C—1.4000E-03
D—1.2000E-03
E—1.0000E-03
F—8.0000E-04
G—6.0000E-04
H—4.0000E-04
I—2.0000E-04
J—1.4552E-11
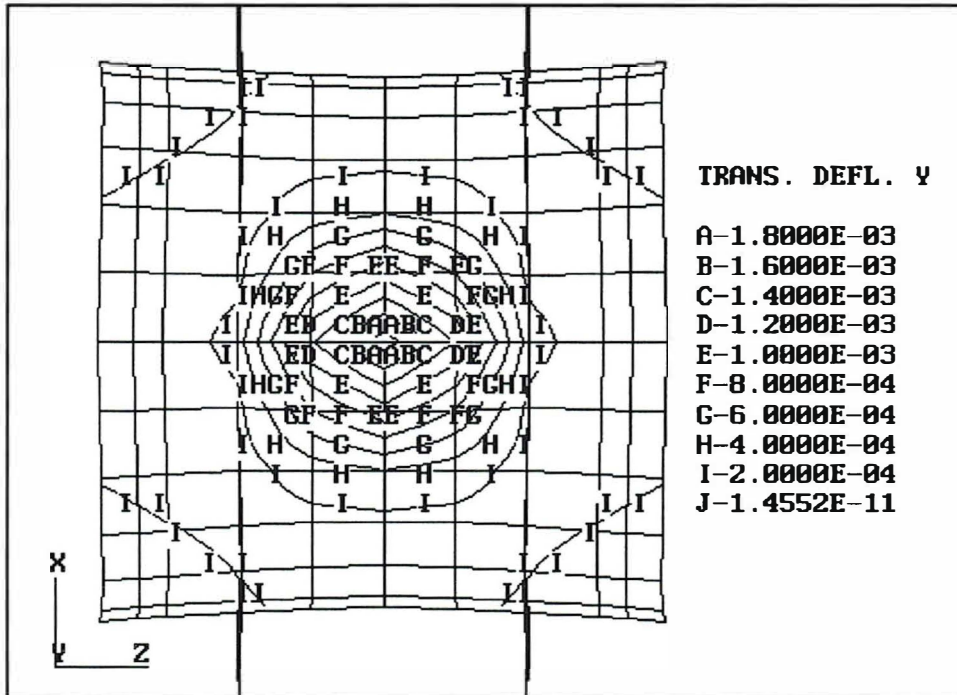
Figure 3.5a - Contour Plot Of Y-Axis Translational Deflection (Top View)

TRANS. DEFL. Y

A—1.8000E-03
B—1.6000E-03
C—1.4000E-03
D—1.2000E-03
E—1.0000E-03
F—8.0000E-04
G—6.0000E-04
H—4.0000E-04
I—2.0000E-04
J—1.4552E-11
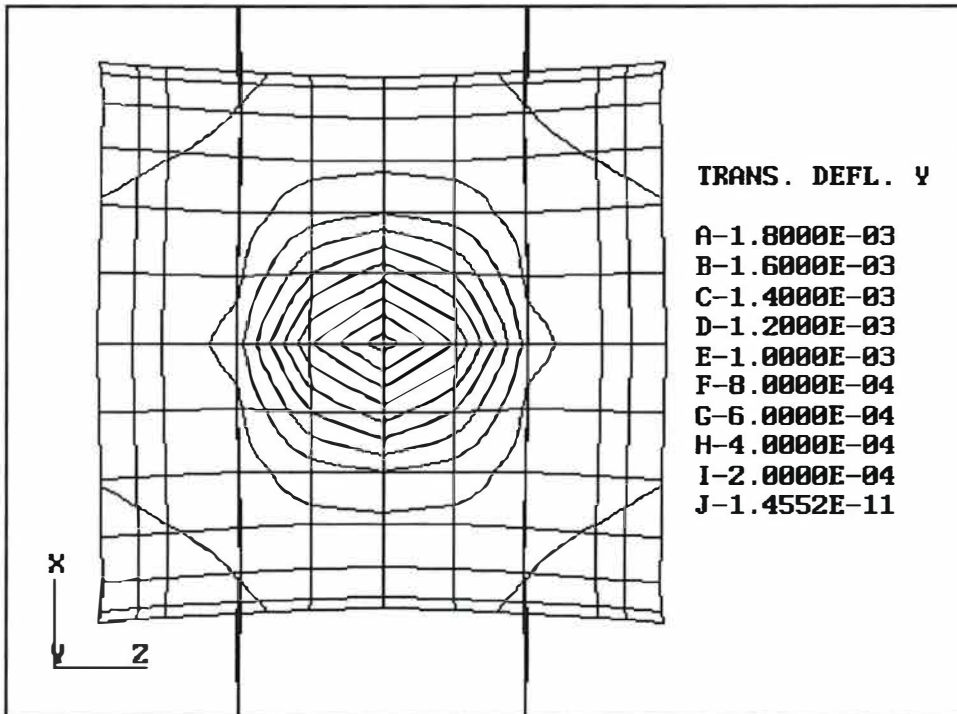
Figure 3.5b - Translational Deflection Plot With Contour Labels Removed

3.4

```
-------------------- Lubricant Properties ------------------------
Gulf Harmony 68, ISO Viscosity Grade = 68
Specific Gravity @ 60 deg. F = 0.87670
API Gravity @ 60 deg. F......= 29.9
Coefficient of Expansion.....= 0.000433
Pour Point deg. F............= 10
Flash Point deg. F...........= 450
Viscosity, cSt (centiStoke)
cSt @ 104 deg. F.............= 62.50
cSt @ 212 deg. F.............= 8.34
Specific Heat Cp (BTU/Lb/F) Coefficients
4.143871E-001,    4.806036E-004,    0.000000E+000,    0.000000E+000
```

```
-------------------- Calculated Properties ------------------------
```

degree F	Absolute Visocity Reyns	Absolute Visocity centiPoise	Kinematic Visocity centiStoke	Density (Grams/CC)	Cp (BTU/Lb/F)
20.00	2.23242E-04	1539.198	1726.82	0.8913	0.42400
30.00	1.29626E-04	893.738	1007.10	0.8874	0.42881
40.00	7.90153E-05	544.790	616.58	0.8836	0.43361
50.00	5.02964E-05	346.781	394.19	0.8797	0.43842
60.00	3.32776E-05	229.441	261.95	0.8759	0.44322
70.00	2.27922E-05	157.146	180.19	0.8721	0.44803
80.00	1.61017E-05	111.017	127.84	0.8684	0.45284
90.00	1.16958E-05	80.639	93.26	0.8647	0.45764
100.00	8.71031E-06	60.055	69.75	0.8610	0.46245
110.00	6.63439E-06	45.742	53.35	0.8573	0.46725
120.00	5.15666E-06	35.554	41.65	0.8537	0.47206
130.00	4.08207E-06	28.145	33.11	0.8501	0.47687
140.00	3.28530E-06	22.651	26.76	0.8466	0.48167
150.00	2.68395E-06	18.505	21.95	0.8431	0.48648
160.00	2.22267E-06	15.325	18.25	0.8396	0.49128
170.00	1.86353E-06	12.849	15.37	0.8361	0.49609
180.00	1.58007E-06	10.894	13.08	0.8326	0.50090
190.00	1.35352E-06	9.332	11.25	0.8292	0.50570
200.00	1.17036E-06	8.069	9.77	0.8258	0.51051
210.00	1.02069E-06	7.037	8.56	0.8225	0.51531
220.00	8.97171E-07	6.186	7.55	0.8192	0.52012
230.00	7.94310E-07	5.477	6.71	0.8159	0.52493
240.00	7.07924E-07	4.881	6.01	0.8126	0.52973
250.00	6.34805E-07	4.377	5.41	0.8093	0.53454
260.00	5.72465E-07	3.947	4.90	0.8061	0.53934
270.00	5.18954E-07	3.578	4.46	0.8029	0.54415
280.00	4.72731E-07	3.259	4.08	0.7997	0.54896
290.00	4.32568E-07	2.982	3.74	0.7966	0.55376
300.00	3.97480E-07	2.741	3.45	0.7935	0.55857

APPENDIX B

FULL FILM HYDRODYNAMIC
MOTOR-BEARING CHARACTERISTICS

CP&L MOTOR BEARING STIFFNESS AND DAMPING CHARACTERISTICS
FOR VARIOUS OIL INLET TEMPERATURES — ISO 68 OIL
N = 800, $C_{diametrical}$ = 10 mils, W = 5,000 lb

Run #	Brg Type	T_{in} °F	T_{max} °F	Bearing Stiffness				Bearing Damping				Critical Journal Weight
				Kxx Lb/in	Kxy Lb/in	Kyx Lb/in	Kyy Lb/in	Cxx Lb-Sec/in	Cxy Lb-Sec/in	Cyx Lb-Sec/in	Cyy Lb-Sec/in	
1	Plain	80°	111° (21°)	1.70 E6	3.77 E6	-1.79 E6	2.01 E6	8.60 E4	2.41 E4	2.41 E4	5.24 E4	342K
2	Plain	100°	122° (22°)	2.00 E6	3.45 E6	-1.76 E6	1.91 E6	7.69 E4	2.17 E4	2.17 E4	3.99 E4	353K
3	Plain	120°	136° (16°)	2.35 E6	3.28 E6	-7.58 E5	1.87 E6	7.17 E4	2.13 E4	2.13 E4	3.20 E4	381K
4	Plain	140°	152° (12°)	2.77 E6	3.30 E6	-4.48 E5	1.87 E6	7.10 E4	2.21 E4	2.21 E4	2.74 E4	423K
5	Plain	160°	170° (10°)	3.35 E6	3.40 E6	-1.61 E5	1.87 E6	7.28 E4	2.26 E4	2.26 E4	2.35 E4	530K
6	Partial Arc 120°	80°	109° (29°)	2.68 E6	4.06 E6	-5.30 E5	1.20 E6	9.94 E4	1.70 E4	1.70 E4	1.70 E4	177K
7	Partial Arc 120°	100°	123° (23°)	2.96 E6	3.89 E6	-4.75 E5	1.27 E6	9.62 E4	1.91 E4	1.91 E4	1.65 E4	177K
8	Partial Arc 120°	120°	139° (19°)	3.41 E6	3.76 E6	-3.63 E5	1.28 E6	9.37 E4	1.96 E4	1.96 E4	1.52 E4	181K
9	Partial Arc 120°	140°	156° (16°)	4.05 E6	3.69 E6	-2.09 E6	1.25 E6	9.20 E4	1.87 E4	1.87 E4	1.31 E4	201K
10	Partial Arc 120°	160°	174° (14°)	4.79 E6	3.70 E6	-6.48 E4	1.24 E6	9.22 E4	1.81 E4	1.81 E4	1.15 E4	243K

DyRoBeS-PC Version 2.10, Dynamic Elegance, Inc.   All Rights Reserved.

Begin Time: 00:25:56.75 am   Date: 01-03-1995

FILE : CP160F.LTO
TITLE: CP&L MOTOR BRG,Cd=10 MILS, W=5,000 LB,N=894 RPM,To=160
Number of theta grids (M): 30
Number of axial grids (N):   6
Length/Diameter Ratio (L/D):   .8000

Number of Pads (NPAD):   1
===================== Bearing Geometric Data =====================
  Pad   Leading Edge   Trailing Edge   Preload   Offset
   1   .00   360.00   .00000   .00000

<English Units>
             Bearing Length L= 6.0000 in.
                   Diameter D= 7.5000 in.
     Pad Radial Clearance Cp= .50000E-02 in.
 Bearing Radial Clearance Cb= .50000E-02 in.

                Bearing Load = 5000.0   Lbf
            Rotational Speed = 894.00   RPM

     Inlet Oil Temperature = 160.00   deg.F
  Heat carried away by oil = 80.00   percent

----------------------- Oil Properties -----------------------------
  Gulf Harmony 68, ISO Viscosity Grade = 68
  Specific Gravity @ 60 deg.F =   .87670
  Coefficient of Expansion   = .00043300
  Viscosity, cSt @ 104 deg.F =   62.500
  Viscosity, cSt @ 212 deg.F =   8.3400
  Specific Heat (BTU/Lb/F) Coefficients
          .414387   .480604E-03   .000000   .000000

_____

==

*************************** Heat Balance ***************************

     Oil Inlet Temperature =   160.00   deg.F
     Oil Dynamic Viscosity = .22227E-05   Reyns (Lbf-sec/in^2)
            Mass Density =   .30331E-01   Lbm/in^3
            Specific Heat =   .49128   BTU/Lbm/F

     Operating Temperature =   163.96   deg.F
     Oil Dynamic Viscosity = .20698E-05   Reyns (Lbf-sec/in^2)
            Mass Density =   .30281E-01   Lbm/in^3
            Specific Heat =   .49319   BTU/Lbm/F

     Max. Film Temperature =   170.03   deg.F
     Oil Dynamic Viscosity = .18624E-05   Reyns (Lbf-sec/in^2)
            Mass Density =   .30205E-01   Lbm/in^3
            Specific Heat =   .49611   BTU/Lbm/F

        Sommerfeld Number =   0.14049
  Bearing Eccentricity Ratio (E/Cb) = 0.627   (E/Cp) = 0.627
  Attitude Angle (deg.) =   47.56

  Minimum Film Thickness =   0.0018674   in.

```
       Maximum Film Pressure =      284.65    psi (Lbf/in^2)
         Friction Power Loss =      1.1802    horsepower
                Side Leakage =      1.4577    gpm (US)

Dimensional Bearing Coefficients (units: K=Lbf/in, C=Lbf-sec/in)
Kxx,Kxy,Kyx,Kyy = 3.17366E+06  3.34495E+06  -2.22969E+05  1.85861E+06
Cxx,Cxy,Cyx,Cyy = 6.40421E+04  1.96065E+04   1.96065E+04  2.15518E+04

The Critical Journal Weight =  405450    Lbf
    Instability Whirl Ratio =  0.4003
```

Synfilm®

Air Compressor & Industrial Oil

		TYPICAL SPECIFICATIONS*				Recip.	
	ISO GRADE	32	46	68	100	100	150
Physical Properties	Viscosity, SSU @ 100°F	165	239	330	474	500	750
	@ 210°F	45.5	50.7	57.1	67	56.5	85
	Viscosity Index	120	120	120	120	50	120
	Pour Point °F	-70	-65	-55	-40	-40	-40
	Flash Point °F	440	470	475	485	485	475
	Fire Point °F	495	515	520	535	535	520
	ASTM D-664, Acid No.	0.3	0.3	0.3	0.3	0.4	0.3
General Performance Tests	ASTM D-892, Foaming, ml						
	Sequence 1, 75°F	Pass	Pass	Pass	Pass	Pass	Pass
	Sequence 2, 200°F	Pass	Pass	Pass	Pass	Pass	Pass
	Sequence 3, 75°F	Pass	Pass	Pass	Pass	Pass	Pass
	ASTM D-1401 Demulsibility	Pass	Pass	Pass	Pass	Pass	Pass
	ASTM D-130, Copper Corrosion						
	3 Hours @ 210°F	1a	1a	1a	1a	1a	1a
	250 Hours @ 210°F	1a	1a	1a	1a	1a	1a
	Cincinnati Millicron Corrosion Test						
	Procedure B, 215°F	Pass	Pass	Pass	Pass	Pass	Pass
	ASTM D-665 Rust Test						
	Fresh Water	Pass	Pass	Pass	Pass	Pass	Pass
	Salt Water	Pass	Pass	Pass	Pass	Pass	Pass
Oil Life Tests	ASTM D-2893 Dry Air Oxidation						
	312 Hours @ 203°F,						
	% Viscosity Increase	0	0	0	0	0	0
	Precipitation No. (% solids)	0	0	0	0	0	0
	ASTM D4742 TFOUT Oxidation, min	1440+	1440+	1440+	1440+	1440+	1440+

* Specifications shown are typical and may vary.

NOTE:
When changing to Synfilm, Synfilm's solvency cleans wear metals and deposits left by previous oils. These wear metals and deposits may cause abnormally high values on used oil analysis until equipment is clean (See "Upgrading to Synfilm" technical bulletin).

Synfilm 100 and Synfilm Recip. 100 are interchangable, but our recommendation is:

Synfilm - for Rotary Screw Compressors, pumps, turbines, motor bearings, etc., to maximize seal compatibility.

Synfilm Recip. 100 - for Reciprocating Compressors, for maximum control of high temperture valve deposits.

CODE 700 *REGAL* OIL R&O 32
CODE 701 *REGAL* OIL R&O 46
CODE 702 *REGAL* OIL R&O 68
CODE 704 *REGAL* OIL R&O N-68
CODE 706 *REGAL* OIL R&O 100
CODE 707 *REGAL* OIL R&O N-100

Typical Characteristics

Regal Oil R&O	32	46	68	N-68	100	N-100
Appearance				Light Pale		
Gravity, API	31.8	29.9	30.3	22.9	27.8	22.0
Flash, COC, F	390	410	445	360	450	385
Pour Pt., F	-25	-15	0	-35	15	-25
Vis. cSt at 40 C	32	46	68	68	100	100
cSt at 100 C	5.4	6.8	8.6	7.2	10.9	9.0
SUS at 100 F	165	237	356	358	528	531
SUS at 210 F	44	48	55	50	64	57
Viscosity Index	104	102	97	45	92	45
Color, ASTM	1.0	1.0	1.0	L2.0	L1.5	L2.0
Ash, Wt. %	0.002	0.002	0.002	0.002	0.002	0.002
RBOT, Minutes	765	795	789	360	709	330
Turbine Oil Oxidation hrs to 2.0 Neut No.	4365	3800	3575	2000	2415	1500

Regal Oils R&O 32, 46, 68, N-68, 100 and N-100 are premium grade oils formulated to meet the stringent lubrication requirements of steam and gas turbines, hydraulic systems and air compressors. The *Regal* Oil R&O 32, 46, 68 and 100 grades are formulated with high quality solvent refined paraffin base stocks while the N-68 and N-100 are formulated with high quality solvent refined naphthenic base stocks. All grades contain a carefully balanced rust and oxidation inhibitor package and an effective anti-foamant which provides excellent anti-foam and air release characteristics. In addition, these products have excellent low temperature and water separating characteristics.

The *Regal* Oil R&O series is formulated with a special oxidation inhibitor package to provide exceptional oxidation stability. In terms of service, this means extended drain periods and greatly decreased risk of the formation of harmful oil oxidation products such as acids, varnish and sludge that can cause damaged bearings or sticking valves.

Regal Oils R&O are recommended for all types of turbines. A representative of Texaco Technical Services should be consulted prior to use of *Regal* Oil R&O series oils in GE Frame 7000 Gas Turbines. They are also recommended for air compressors and turbo-blowers, as well as hydraulic systems where an anti-wear lubricant is not required, and certain axial piston pumps containing silver plated parts. Applications also include electric hoists, pumps, small gear sets, electric motor bearings, machine tools, rolling bearings and many other industrial applications. Refer to page A-9 of the Products Manual Appendix for AGMA gear oil specifications relating to the *Regal* Oil R&O series meeting the requirements of AGMA for rust and oxidation type gear oils.

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Additional References

Royal Purple Marketing Brochure, Royal Purple Synthetic Lubricants, Inc., Humble, TX.

Specification Sheet: Synfilm Air Compressor & Industrial Oil, Royal Purple Synthetic Lubricants, Inc., Humble, TX.

Lubricating Oil, Grease and Antifreeze/Coolant Products Digest, 1992, Texaco Lubricants Company, P.O. Box 4427, Houston, TX, 77210-4427.

Begin Time: 04:11:07.87 pm   Date: 12-28-1994

FILE : CPL100.LTO
TITLE: CP&L MOTOR BEARING - 5,000 LB LOAD, Cr=5 MILS, 800 RPM
Number of theta grids (M): 30
Number of axial grids (N):   6
Length/Diameter Ratio (L/D):   .8000

Number of Pads (NPAD):   1
======================= Bearing Geometric Data =======================
   Pad   Leading Edge   Trailing Edge   Preload   Offset
    1   .00   360.00   .00000   .00000

<English Units>
             Bearing Length L= 6.0000 in.
                   Diameter D= 7.5000 in.
       Pad Radial Clearance Cp= .50000E-02 in.
   Bearing Radial Clearance Cb= .50000E-02 in.

               Bearing Load = 5000.0   Lbf
           Rotational Speed = 800.00   RPM

       Inlet Oil Temperature = 100.00   deg.F
   Heat carried away by oil =   80.00   percent

----------------------- Oil Properties ----------------------------
   Gulf Harmony 68, ISO Viscosity Grade = 68
   Specific Gravity @ 60 deg.F =   .87670
   Coefficient of Expansion   = .00043300
   Viscosity, cSt @ 104 deg.F =   62.500
   Viscosity, cSt @ 212 deg.F =   8.3400
   Specific Heat (BTU/Lb/F) Coefficients
         .414387   .480604E-03   .000000   .000000

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*************************** Heat Balance ***************************

     Oil Inlet Temperature =   100.00   deg.F
     Oil Dynamic Viscosity =   .87103E-05 Reyns (Lbf-sec/in^2)
             Mass Density =   .31106E-01 Lbm/in^3
            Specific Heat =   .46245   BTU/Lbm/F

     Operating Temperature =   112.51   deg.F
     Oil Dynamic Viscosity =   .62167E-05 Reyns (Lbf-sec/in^2)
             Mass Density =   .30941E-01 Lbm/in^3
            Specific Heat =   .46846   BTU/Lbm/F

   Max. Film Temperature =   121.80   deg.F
     Oil Dynamic Viscosity =   .49381E-05 Reyns (Lbf-sec/in^2)
             Mass Density =   .30820E-01 Lbm/in^3
            Specific Heat =   .47292   BTU/Lbm/F

     Sommerfeld Number =   0.33332
   Bearing Eccentricity Ratio (E/Cb) = 0.419   (E/Cp) = 0.419
     Attitude Angle (deg.) =   61.13

   Minimum Film Thickness =   0.0029026   in.

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Maximum Film Pressure =        239.89   psi (Lbf/in^2)
    Friction Power Loss =        2.1644   horsepower
          Side Leakage =        0.87734  gpm (US)

Dimensional Bearing Coefficients (units: K=Lbf/in, C=Lbf-sec/in)
Kxx,Kxy,Kyx,Kyy = 2.00217E+06  3.44893E+06  -1.17584E+06  1.90811E+06
Cxx,Cxy,Cyx,Cyy = 7.69457E+04  2.17139E+04  2.17139E+04  3.94383E+04

The Critical Journal Weight = 353290    Lbf
    Instability Whirl Ratio =  0.4858
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