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POWERPLANT LUBRICANT SELECTION FOR IMPROVED EFFICIENCY AND ENVIRONMENTAL IMPACT REDUCTION

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ABSTRACT

Computational models were used to optimize bearing performance by adjusting a number of lubricant properties. This computational optimization showed that the most beneficial characteristics to hydrodynamic bearing operation were high viscosity index (VI) and high specific heat capacity. Four environmentally adapted synthetic lubricants were developed to provide these characteristics including: ISO VG32 with 259 VI, ISO VG22 with 245 VI, ISO VG22 with 336 VI, and ISO VG15 with 226 VI. A full scale bearing test machine was then operated with these lubricants in addition to mineral based turbine oils, ISO VG68 with 103 VI and ISO VG32 with 105 VI, to determine the effect on bearing performance and to validate the models. The new lubricants reduced bearing power loss by up to 20% and significantly reduced bearing temperatures with somewhat reduced film thickness. The machine was then operated to provide equivalent minimum viscosity with the new lubricants by varying inlet temperature, finding that changes in power loss were less substantial with equivalent minimum viscosity.

Comparison of simulated and experimental results led to development of a simple, practical method to estimate benefits and operational parameters for lubricants based on viscosity grade, viscosity index and a simplified description of the ma-

chine's bearings. Other, less tangible, factors considered are bio-degradability and impact of power loss reduction.

INTRODUCTION

The increasing focus on renewable energy sources including wind, solar and hydro-electric power has lead to the installation of industrial machines in natural environments. To minimize the hazardous environmental impact that these machines can cause, many owners are evaluating and switching to new, environmentally adapted lubricants (EALs). However, lubricant change intervals can exceed 30 years and changing lubricant in large rotating machines is a major undertaking often requiring replacement of packings and extensive cleaning. With significant purchase price differences between mineral and synthetic lubricants, a balance between initial costs and operational benefits must be clear.

Meanwhile, developments in the synthetic lubricants which make up EALs have made it possible to not only reduce environmental impact but at the same time significantly improve machine performance. This has turned lubricants from necessary maintenance products into performance enhancing investments.

With a host of different synthetic lubricants in production, all with unique base oils and additives, it can be quite confusing for the end user to decide which lubricant to use, and to de-

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termine how a specific lubricant can affect a specific machine's performance.

Experimental Studies

A number of experimental studies have been conducted on fluid film bearings, both thrust and journal, in laboratories and in the field which have helped to develop a description of the effects that can be expected following a lubricant change. While the most commonly reported result is a decrease in maximum temperature of the bearing, the specific results and conditions of the experiments have some variation.

In laboratory tests, power loss reductions and equivalent film thickness were found to result from changing to a thinner synthetic lubricant in thrust bearings by New and Schmaus [1] and later by Glavatskih and Larsson [2]. Boehringer and Neff found significant improvements in machine performance upon changing to a di-ester based lubricant [3] in a full scale hydropower thrust bearing. Similar improvements in a full scale hydropower machine were found by Glavatskih [4] upon changing to an ester based lubricant in a combined thrust journal bearing. Ferguson et al. [5] investigated ISO VG68, VG46 and VG32 in a large thrust bearing test rig and proposed using a special numerical modeling software package to predict maximum bearing temperatures from oil bath temperatures and thereby predict bearing performance characteristics. Calculated results agreed well with experiments on a large thrust bearing. Lower power losses were found for lower viscosity grade (VG) and higher oil bath temperatures. Significant decreases in film thickness were also observed but it was argued that the lower film thickness was still adequate to maintain machine safety.

Investigations with journal bearings by Swanson et al [6] in work with VI improvers found that the thermal performance of mineral-based oils could be improved to match that of synthetic lubricants. Separately, Dmochowski [7] and McCarthy et al [8] found similar effects in their studies on the performance of journal bearings with high VI polyalphaolefin and ester based lubricants. However, power loss reductions in journal bearing studies were generally lower magnitude, percentage wise, than those found for thrust bearings and laboratory results have shown less reduction than field experiences.

The majority of earlier experimental work has kept inlet or oil bath temperature constant so as to keep initial operating conditions equivalent for each of the lubricants. Power loss, film thickness and temperature were then found to compare performance of the varying lubricants. Temperatures throughout the bearing were used to calculate lubricant viscosity which was then compared, finding that in the case of the higher VI lubricants, viscosity was equivalent in the region of highest temperature (lowest viscosity) [7, 8]. The findings from McCarthy et al showed that the film thickness was lower for the higher VI lubricants until the speed and load had increased to a point at which the minimum

FIGURE 1. HYDRO-DYNAMIC JOURNAL BEARING TEST RIG.

viscosities of the lubricants became equivalent and it was concluded that the reduced power losses provided by the higher VI lubricants were the result of lower bulk lubricant viscosity in the bearing.

Numerical Studies

The earliest models of fluid film bearings handled viscosity as a constant through the bearing rather than as a parameter locally affected by changes in temperature, however it was soon found that variation of lubricant viscosity plays an essential role in the Reynolds equation. This effect of including viscosity effects was shown as early as 1933 by Kingsbury [9] to reduce power loss significantly, up to 60% in some cases. Later, the generalized Reynolds equation with the variable viscosity term was derived by Dowson [10] which significantly improved the accuracy of the numerical models. The next step was done in the area of thermal boundary conditions. A heat flux based boundary was proposed by Dowson and March [11] and an adiabatic boundary by Pinkus and Bupara [12]. Around the same time, the effect of thermoelastic deformation has been taken into account Rohde and Kong Ping [13].

Through the 1980's and '90's, models for fluid film bearings progressed in accuracy with the growth of computing power. Recently, Tanaka and Hatakenaka [14] presented a very accurate numerical model including detailed description of the boundary conditions and solution scheme.

While numerical and experimental studies have provided much broader understanding of bearing function and methods for improving bearing performance, very few studies have provided useful guidance for the practicing engineer. The present study seeks to provide practical guidance for the end user regarding methodology to analyze changes in lubricant.

EXPERIMENTAL SETUP

Lubricants were tested in a 180 mm diameter, two-axial groove, Babbitt coated journal bearing with L/D ratio of 0.7. The

TABLE 1. MEASUREMENT UNCERTAINTY.

Measurement	Type	Uncertainty
Displacement (Internal)	Inductive	$\pm 5 \mu\text{m}$
Displacement (External)	Inductive	$\pm 5 \mu\text{m}$
Temperature	Thermistor	$\pm 0.5 \text{ }^\circ\text{C}$
Temperature	Thermocouple (K)	$\pm 1 \text{ }^\circ\text{C}$
Speed	Motor Electronics	0.4%
Power Loss	Motor Electronics	0.6%

test equipment, Fig. 1, has a shaft equipped with two inductive displacement sensors and four thermistors on the shaft to continually measure film thickness and shaft temperature. Temperature along the bearing surface is measured using 46 thermistors arranged to determine circumferential and axial film temperature profiles. Lubricant supply is controlled using an in-line flow meter while lubricant temperatures are monitored using thermocouples and temperature is controlled using a heat exchanger. Two inductive displacement sensors are located outside the housing for redundancy in film thickness and eccentricity measurements. The shaft is held in place by two identical roller bearings and is powered by a 43 kW motor which provides power and speed measurements through its controller. Loading is applied to the bearing housing using a pneumatic cushion equipped with four 50 kN load cells. Uncertainty for the various measurements is provided in Table 1.

The lubricants investigated in this work consist of industry standard mineral-based ISO VG68 and ISO VG32 turbine oil as well as four environmentally adapted synthetic lubricants (EALs) including an ISO VG32, two ISO VG22 and an ISO VG15. Characteristics of these lubricants are detailed in Table 2 as well as the abbreviations used for convenience.

Initial testing of the lubricants was accomplished in the same manner as the earlier mentioned experimental studies and is detailed in Table 3. Inlet temperature was kept constant while load and speed were varied with measurements taken upon reaching steady state at each testing point. After completion of the initial round of tests, the maximum temperatures in the bearing at each state were compared and calculations of minimum viscosity were made.

Further testing with VG68 was then accomplished by setting the maximum film temperature to these calculated values by changing the inlet temperature to compare lubricant performance under equivalent film viscosity. In these further tests, effort was made to achieve steady state in all cases, however, due to the high temperatures involved and efficient system cooling through

TABLE 2. LUBRICANTS USED IN EXPERIMENTAL STUDIES

Grade	Abbreviation	Viscosity, (mm^2/s)		VI
		40 $^\circ\text{C}$	100 $^\circ\text{C}$	
ISO VG68	VG68	67.3	8.79	103
ISO VG32	VG32	33.7	5.62	105
ISO VG32	SE32	32.1	8.46	259
ISO VG22	SE22	21.4	5.87	245
ISO VG22	SV22	20.4	6.81	340
ISO VG15	SE15	15.5	4.46	226

TABLE 3. TESTING PROGRAM FOR EXPERIMENTAL STUDIES.

Parameter	Unit	Value
Speed	RPM	1000, 1500, 2000, 2500
Mean Pressure	MPa	1.0, 1.5, 2.0
Inlet Temp.	$^\circ\text{C}$	40 ± 1.0
Oil Flow	l/minute	2.0 ± 0.05

radiation, some tests could not be performed at steady state while temperatures required for comparing VG68 and SE15 were only achievable in a few cases. This caused power loss comparisons for the lower VG lubricants to be somewhat less reliable than those of the higher VG lubricants.

NUMERICAL MODEL

Model Description

Numerical experiments were conducted using a thermo-hydro-dynamic (THD) model described fully by Kuznetsov et al [15]. This model includes thermal effects and flow (oil film pressure) analysis. Pressure distribution is found by solution of the simplified Reynolds equation. The finite difference technique is used with zero boundary conditions for discretization of Reynolds equation. Cavitation effects are considered by introduction of a switch function. Analysis of thermal effects is performed by solving the energy and heat transfer equations while an iterative successive-over-relaxation (SOR) method is then used to solve the final system of equations.

Boundary conditions are set to give zero heat flux for the shaft and conservation of the heat flux for oil-bearing and

bearing-air interfaces. It is assumed that the cold oil flow is split equally between two supply grooves. The oil mixing prediction is based on a predefined mixing coefficient, Tanaka and Hatakenaka [14]. Both equations are then discretized using the finite difference method and solved simultaneously with the iterative SOR method.

The solution process begins from a uniform viscosity guess, from which the Reynolds equation is solved. Next, the obtained solution is used to predict the temperatures within the oil film and the bearing. The influence of thermal expansion of the shaft and bearing are then accounted for and a new viscosity field is computed using the previously obtained temperatures. The process is then iterated using the newly calculated viscosity as the new guess. This process is repeated in a loop until no changes in viscosity are visible, that is the difference in lubricant temperatures between the two preceding iterations satisfies the convergence criterion.

Numerical Experiment Setup

Geometry for the numerically modeled bearing was identical to that of the actual test bearing. Discretization was performed on a mesh grid of size 256 points in the circumferential, 33 points in the axial and 21 points in the cross-film directions. Convergence criteria are, pointwise, 10^{-6} for the pressure and 10^{-7} for the temperature of the oil fluid film.

The goal of the numerical experiment was to achieve the same load and minimum film thickness for oils VG32, VG68, SE32, SE22 and SV22 and compare the variation of the power loss and oil maximum temperature versus rotational speed. To accomplish this, eccentricity was fixed at 0.80 and the model was run with a range of supply temperatures from 5°C to 80°C. Then an interpolation of load, maximum oil temperature and power loss versus supply oil temperature was performed to determine values for the given load. This yielded equivalent performance at 52 kN load (2.2 MPa mean bearing pressure), slightly higher than the 2.0 MPa maximum mean pressure used in the experiments.

RESULTS

The experimental and numerical investigations both demonstrated that lubricants of widely varied viscosity grade and viscosity index can provide film forming function in a journal bearing. The results further highlight the sensitivity of power losses in journal bearings to small changes in viscosity in the bulk of the bearing. An examination of the temperature through the bearing pointed out viscosity zones that occur when lubricants are replaced while comparing the power loss and change in eccentricity allowed for further comparison of the lubricants.

FIGURE 2. TEMPERATURE AND VISCOSITY AT 2 MPa LOAD AND 24 M/S.

Temperature and Viscosity Zones

During the initial investigation with constant inlet temperature of 40° it was found that film thickness in the bearing was equivalent for both VG68 and SE32 in cases where the lubricant viscosity was equal or greater for SE32 in the area of highest temperature, Fig. 2. In this particular case, SE32 yielded approximately 10% power reduction compared to VG68. To describe where the difference in losses come from, the bearing was divided into the three zones shown in Fig. 2. Zone 1 is the region on the loaded half of the bearing where the lubricant has relatively high viscosity which can add to losses, Zone 2 can be considered the primary load bearing area where pressure and temperature are greatest, while Zone 3 is the unloaded region of the bearing where greater viscosity adds to losses only. From this division it can be determined that having the proper viscosity in the most heavily loaded region allows for the desired minimum film to form. Likewise, decreased viscosity through the rest of the bearing allows for reduction in power losses from lubricant shearing. This hypothesis led to the development of the viscosity matching experiments as well as the numerical investigation with fixed eccentricity and load.

The results of the numerical investigation reinforced the relationship between maintaining eccentricity and viscosity in Zone 2. As shown in Fig. 3, the minimum viscosity in the bearing was very similar for all lubricants at any given speed. The slightly higher viscosity seen for the high VI lubricants reflects the earlier finding that matching eccentricity occurs when the high VI lubricant has slightly higher viscosity in Zone 2 than the low VI lubricants. The maximum viscosity found in Zone 1 and 3 was lower for the high VI lubricants in the model as shown in Fig. 4. The low viscosity in the unloaded regions shows the potential loss reduction allowed by the high VI lubricants.

FIGURE 3. MINIMUM VISCOSITY IN MODEL BEARING VS. SPEED.

FIGURE 4. MAXIMUM VISCOSITY IN MODEL BEARING VS. SPEED.

Power Loss

Unlike the initial investigations, changes in power loss were much smaller in the viscosity matching experiments. However, comparison between VG68 and the high VI lubricants showed that the high VI lubricants tend to have lower power loss with increasing speed. Comparison between all lubricants and VG68 are shown in Fig. 5 and Fig. 6 for 1 MPa and 2 MPa mean pressure respectively. Very little change in power loss was observed for lower speeds, and in most cases, the power loss was seen to increase with the thinner lubricants at the lowest speed. No trend was observed for VG32 which appeared to produce a slightly higher power loss at all speeds compared to VG68. This agrees well with Ferguson et al. [5] who found that VG32 and VG68 behaved similarly at equivalent viscosity. Similar trends were observed in the model results except that the differences in power loss were within the error of the interpolation scheme.

FIGURE 5. POWER LOSS CHANGE AT 1 MPa LOAD VS. SPEED.

FIGURE 6. POWER LOSS CHANGE AT 2 MPa LOAD VS. SPEED.

Eccentricity Change

Eccentricity behaved similarly to power loss in the experimental investigation when VG68 was compared to the high VI lubricants as shown in Fig. 7 and 8 for 1 MPa and 2 MPa mean bearing pressure respectively. At lower speeds, the synthetic lubricants were unable to carry the same film thickness as their thicker mineral oil counterparts. However, as speed increases, the eccentricity provided by the synthetic lubricants became approximately equal to that of the mineral lubricant until at the highest speeds, the synthetics provided lower eccentricity. The eccentricity provided by VG32 was similar to that of the synthetic lubricants, with a higher eccentricity at lower speeds and lower eccentricity in relation to VG68 at higher speeds. This behavior is supported by the minimum viscosity results from the model, Fig. 3 in which the high VI lubricants have slightly higher minimum viscosity at all speeds for equivalent eccentricity and load. In the experiment, minimum viscosity was computed from the maximum measured temperature which was slightly lower than the actual maximum film temperature which led to minor

FIGURE 7. ECCENTRICITY CHANGE AT 1 MPA LOAD VS. SPEED.

FIGURE 8. ECCENTRICITY CHANGE AT 2 MPA LOAD VS. SPEED.

discrepancies in matching VG68 with the other, much thinner lubricants.

This effect is believed to be a result of the location of the temperature measurement as the maximum temperature measured in the bearing is somewhat lower than the actual maximum temperature in the oil film. Because this difference in temperature increases with speed and because the temperature matching was done at the lower temperature, the synthetic lubricants actually had higher viscosity in the maximum temperature region than the VG68 lubricant. This phenomenon further helps explain why the lower viscosity synthetics provided lower eccentricity than SE32 as well as the slight difference in power loss of the SE22 and SV22 compared to SE32.

METHOD FOR LUBRICANT SELECTION

With a host of new lubricants available a simple method of evaluating potential lubricants should be followed to allow for selection of the best alternative. Any evaluation should include

a determination of the value of biodegradability, cost of disposal and lubricant life time. Following this, a determination of which lubricants can be used practically, and an evaluation of how they should be operated should be carried out. Finally the various options can be compared to determine what performance changes can be expected through replacement of the existing lubricant.

Environmental Factors

Long service life and environmental friendliness often work against each other in the case of bio-lubricants. Bio-degradable lubricants, such as rapeseed oil have been found to perform well in short tests, McCarthy et al. [8], however as noted by Schneider [16] many plant-based lubricants oxidize much more rapidly than mineral oil based lubricants. Some ester based lubricants have been found to have oxidation stability equivalent to or better than mineral oils. However, ester based lubricants have been found to affect sealing materials which can turn a lubricant change into a complicated process including replacing incompatible sealing materials with compatible materials. A different view of the issue from an economist, Mann [17], reported that changing industries to bio-based lubricants is more successful when governments institute the change rather than industries take the initiative themselves. In spite of this, many industries have changed lubricants and been successful as reported by Broekhuizen [18] so the coupling between economic and environmental advantages clearly depends on the specific application.

An often overlooked aspect of environmental impact reduction is that improved efficiency of existing equipment reduces the need for new construction. Furthermore, while most renewable energy sources are variable (wind, solar, tidal, etc), most existing power sources (fossil fuel, nuclear, and hydro power) can be regulated and thus are more valuable to the electricity grid than variable sources. These regulating power machines are generally much larger than variable power sources, thus small improvements in efficiency can result in significant real increases in power output.

Viscosity Matching

Earlier recommendations for changing lubricant in machines involved using specially developed computer models to predict performance changes from an oil change. The aim of the current study has been to develop a simpler method through a clearer understanding of the functional differences in lubricants. It has been shown that film thickness is maintained in the bearing as long as viscosity of the new lubricant matches or exceeds that of the old in the loaded zone of the bearing. Thus when changing lubricant from a low VI to a high VI, viscosity should be calculated in the loaded zone for the old lubricant, then from that viscosity, the required maximum temperature of the new lubricant can be calculated.

The bearings of most large machines already have temper-

ature sensors in the oil bath and in the pads that can be used to determine the bath and 'maximum' temperature. Because of inconsistency in how temperature sensors are mounted in the pads in industry, the difference between the pad temperature and the maximum temperature in the oil film is not usually clear. However, given that these sensors should always give a temperature colder than the actual maximum, calculating with them guarantees that the minimum viscosity in the film will be greater for the new lubricant than the old. While this assumption predicts less savings through changing lubricant it trades savings for improved machine safety in the form of greater film thickness. The bearing should then be operated using the calculated maximum temperature of the new lubricant to maintain equivalent minimum viscosity.

Estimating Performance Changes

As the load carrying function of a fluid film bearing is provided by only a portion of the lubricated surface, it can be assumed that losses in the load carrying portion of the bearing should be nearly equivalent regardless of the lubricant if equivalent function is to be maintained. However, by allowing for viscosity reduction through the unloaded part of the bearing, churning losses can be significantly decreased with the use of lower viscosity grade but higher VI lubricants. Because churning losses are directly associated to viscosity, it follows that an estimate of the percentage change in power losses can be described by Eq. 1.

$$\Delta_{losses} = A * [1 - \frac{\mu_{new}}{\mu_{old}}] \quad (1)$$

Where A is the percentage of the bearing's area which contributes to churning losses *only* (e.g. non-load bearing area) and μ_{new} and μ_{old} are the viscosity of the oil bath for the new and old lubricants. A can vary greatly depending on the type of bearing in question. For example, a tilting pad thrust bearing in a power plant has considerably greater churning area than a plain journal bearing due to gaps between pads and vertical surfaces of the shaft. The temperature increase, ΔT , between the bath and the maximum temperature is assumed to be the same for the lubricants as losses are assumed to be equal in the loaded portion of the bearing. This is not entirely correct as in the experimental work, it was found that synthetic lubricants had a smaller ΔT which was partly due to the difference in specific heat capacity, ρC_P , of the lubricants; synthetic esters have slightly higher ρC_P than mineral oils, Pettersson [19]. Given equivalent losses developed in the oil film, the lubricant with higher ρC_P experiences lower ΔT .

In the case that accurate measurements of ρC_P are known, ΔT can be reasonably estimated using Eq. 2.

$$\Delta T_{new} = \frac{\rho_{old} C_{Pold} \Delta T_{old}}{\rho_{new} C_{Pnew}} \quad (2)$$

Including ρC_P in the calculations allows for more accurate estimation of bath temperatures which, in the case of increased ρC_P , allows for a greater decrease in losses due to lower bath viscosity.

The total change in power loss can finally be calculated using estimates of the bearing losses in the system. For hydroelectric power generators the bearing losses are approximately 0.2% [5] of the machine's output but this varies with machine speed.

This value appears low at first glance, however with an output of 100 MW from the machine, the bearing losses are on the order of 200 kW which is quite significant. Unlike in other industries where new lubricants are compared in terms of the savings from less frequent oil changes and changes in maintenance practices, Johnson [20], the value and the power savings in a power plant can also be directly applied as increased power output. This increased power output can then be valued at sales prices for electric power and the operation schedule of the machine.

Practicalities

It should be noted that the approach of matching viscosity presents practical limits in terms of the relationship between viscosity grade (VG) and VI. The trend for optimum performance is to reduce VG and increase VI, however this poses a challenge. The additives used to increase VI also increase VG which leads to the selection of ever thinner base fluids. Additionally, to provide matching viscosity, the required viscosity occurs at an increasingly lower temperature leading to an even lower temperature in the oil bath. An example to exemplify this being that to replace VG68 with SE15 in a hydropower turbine bearing with maximum temperature around 75°C would require the oil bath to be less than 5°C which is clearly unreasonable. In the same application, substitution with SE32 would leave the oil bath at around 30°C, a low but much more realistic temperature in a hydropower station.

The argument has previously been made that oil film thickness can be safely reduced to some degree [5, 21] by operating with a higher maximum temperature thus allowing for further loss reduction. However, this decision should be taken separately from a change of lubricant.

Practical Example

As part of a research program, the lubricant in one 10 MW turbine at the Porjus power station on the Lule River in north-

ern Sweden was changed from ISO VG68 mineral oil to an ester based ISO VG46 lubricant with VI 150. The original maximum temperature was 95°C and the original oil bath temperature was 30°C. Matching viscosity between the old lubricant and the new lubricant results in that the maximum temperature of the new lubricant should be 91°C which leads to a bath temperature of 30°C. This calculation includes a small change in ρC_p from 1.8 J/cm³ to 1.9 J/cm³ incorporated using Eq. 2. Assuming an area of losses in the large, fully immersed combined thrust/journal bearing of 50% results in a prediction for power savings in the bearing of 16%. After the actual oil change, the maximum temperature decreased to 84°C because coolant flow was unchanged from prior to the oil change. This resulted in the oil bath being a cool 24°C during the winter months. As could be expected from such a drop in bearing temperature, the oil film thickness increased by 20%. During the summer months, the temperature in the bath was allowed to increase to 35°C while the maximum temperature increased to 88°C. An 18.5% reduction in bearing power loss was measured by the temperature change in the coolant flow compared to the original lubricant while the film thickness remained thicker than prior to the oil change. The reduction in bearing power loss allowed for an increase of production of 30 kW which while small, is not insignificant considering the relatively small size of the machine.

This example highlights the validity of the simplistic viscosity matching method. After changing lubricant, temperatures and savings predicted by the methodology were reasonably close to the actual values observed. While the operators in this example could have yielded greater reductions in power loss by allowing the new lubricant to operate warmer, they were satisfied with increased safety in terms of increased film thickness.

CONCLUSION

This work has provided comparison between two standard mineral based turbine oils and four synthetic based environmentally adapted turbine oils. The comparison of lubricant performance led to the development of a simple method for the practicing engineer to determine the economic value of changing lubricant in a machine in terms of power savings and increased output potential as well as the new operating parameters for the bearings. This allows the following conclusions to be drawn:

- Equivalent machine performance (similar losses and film thickness) is provided when a high viscosity grade lubricant is replaced with a lower viscosity grade lubricant operated at lower maximum and inlet temperatures.
- Improved machine performance (equivalent eccentricity and decreased losses) can be provided when high viscosity grade lubricants are replaced with lower viscosity grade and higher viscosity index lubricants operated at lower

temperatures. This decrease in losses occurs due to a decrease in losses in the non-load bearing portions of the bearing.

- The optimum operating conditions for the new lubricant and changes in bearing performance can be predicted using a simple comparison of the minimum viscosity in the maximum temperature zone of the bearing. Matching viscosity in the maximum temperature region allows for reduced viscosity in the rest of the bearing which in turn allows for reductions in power loss in the bearing.

- Reductions in power loss in addition to the improved environmental characteristics of the new lubricants can make up for the initially higher cost of changing lubricant.

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