PERFORMANCE OF ADIPATE DIESTER SYNTHETIC LUBRICANTS IN THE HYDRODYNAMIC REGIME

by

Nigel H. New

Manager of Product Development

Glacier Metal Company, Ltd.

Alperton, Wembley,

Middlesex, England

and

Richard H. Schmaus

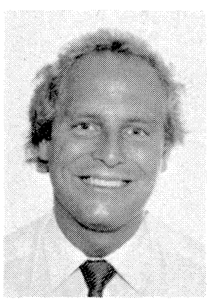
Project Engineer

Central Engineering Division

Exxon Chemical Company

Florham Park, New Jersey

Nigel H. New was educated at City University in London, England, and in 1963 he became a Research Engineer at Ricardo (1927) Limited. In 1965 he joined Glacier Metal Company as an Application Engineer, working on the design of bearings for diesel engines and on computer aided design programs. Later, he became Senior Project Engineer, and, subsequently, Manager of Special Projects Department. In that capacity his responsibilities ranged from procurement of computer facilities to troubleshooting of bearing problems on customer sites. In 1982 he became Manager of Product Development, responsible for new materials and products within the company.



Richard H. Schmaus is a Project Engineer with the Central Engineering Division of the Exxon Chemical Company. During the last year, much of his time has been devoted to providing machinery technical assistance to the Paramins Technology Division of Exxon Chemicals, in support of their synthetic lubricants development, testing and marketing efforts. His responsibilities also include consulting and field assistance on rotating machinery problems. Mr. Schmaus holds B.S. and M.S. degrees in Mechanical Engineering from the University of Virginia.

ABSTRACT

Diester based synthetic lubricants provide numerous performance advantages over mineral oils in industrial applications. The synthetics not only permit significant extension of oil drain intervals and application over a wider temperature range than mineral oils, but field experience also indicates that synthetic oils develop thicker and stronger films than their mineral oil counterparts. This results from the combination of the diester basestock and appropriate additives.

A laboratory evaluation has been conducted to quantify the performance advantages of the synthetic oils over mineral oils in hydrodynamic lubrication. The power loss, maximum pad temperature, and oil film thickness in a tilting pad thrust bearing were measured for an ISO VG 32 mineral oil and ISO VG 32 and VG 10 diester based synthetic oils. Results of the tests revealed that the comparable grade mineral and synthetic oils yielded similar bearing power losses and pad temperatures, while the synthetic product developed significantly thicker films. Relative to the VG 32 mineral oil, the VG 10 synthetic lubricant yielded lower bearing power losses and cooler operation, while developing a slightly thicker film.

These laboratory test results confirm the field experience that replacement of mineral oil with a suitable grade synthetic lubricant can yield significant economic and technical benefits, including reduction in bearing power losses, without sacrificing machine protection.

INTRODUCTION

Industrial applications of adipate diester synthetic lubricants are becoming increasingly common. Extensive laboratory test programs and field experience have demonstrated numerous performance advantages of these lubricants over mineral oils.

The high chemical stability of adipate diester lubricants, and the corresponding potential for significant extension of oil drain intervals and for successful application in high temperature services have been verified in many applications and are well documented. In addition, field experience also indicates that this class of synthetic lubricants provides considerably greater film strength than its mineral oil counterparts. This greater film strength implies that mineral oils may be replaced with less viscous, adipate diester lubricants and thereby yield energy savings without sacrificing machine protection.

FIELD EXPERIENCE WITH ADIPATE DIESTER SYNTHETIC LUBRICANTS

The use of adipate diester synthetic lubricants has reduced machinery maintenance requirements in many applications. For example, several centrifuge gearboxes in a northeastern chemical plant were providing a low service factor, due to frequent gear failures, when the gearboxes were lubricated with an ISO VG 220 EP mineral oil. The bronze-on-steel helical gears increased the 1800 rpm shaft speed of the 50 horsepower motor up to the centrifuge bowl speed of 4500 rpm. Transient loads from the process and occasional shock loads on the gears led to gear failures every three months. The mineral oil was replaced with an adipate diester synthetic EP

220 lubricant in September 1981 in an effort to improve gearset life. No gear failures have occurred in the twenty-two months since the change to synthetics. The superior wear protection provided by the synthetic lubricant is graphically illustrated by the gear photographs in Figure 1.

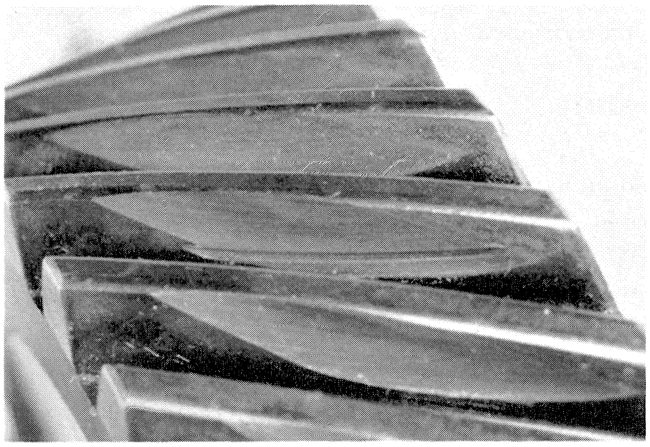


Figure 1a. Centrifuge Gears From Field Test (Three Months Operation on Mineral Oil).

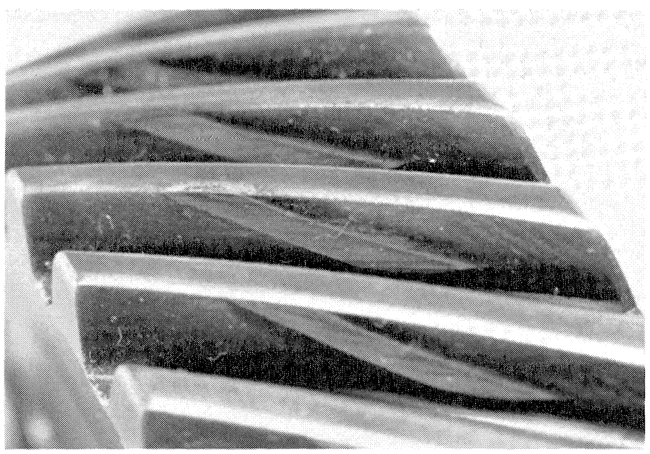


Figure 1b. Centrifuge Gears From Field Test (Six Months Operation on Synthetic Lubricant).

In another application, an ISO VG 32 adipate diester synthetic lubricant replaced ISO VG 100 mineral oil as the cylinder lubricant in a 1200 horsepower integral gas engine compressor. Although the primary objective of the test was to demonstrate energy savings, maintenance credits were also captured by using the less viscous synthetic lubricant. Even though a lower feed rate was used with the synthetic product, compressor piston ring replacements were reduced from an average of three to four per year with mineral oil to none in fifteen months' operation with the synthetic lube. In addition, there was no detectable wear in other compressor wearing parts.

Other cases are also documented in which adipate diester lubricants without EP additives have been successfully applied in equipment where EP additives are recommended for satisfactory operation with mineral oils. In other applications, thinner grades of synthetic oils have replaced more viscous mineral oils and performed well, while a similar reduction in the viscosity of a mineral oil would result in damage to the equipment. These field data indicate that the synthetic lubricants provide stronger films than do mineral oils.

LABORATORY TESTS

Independent tests have been conducted by the Glacier Metal Co., Ltd. to substantiate the observed differences in running characteristics between adipate diester synthetic lubricants and mineral oils. The objective of the tests was to quantify the advantages of the synthetic lubricants in hydrodynamic lubrication, in terms of operating film thickness, bearing power loss and operating temperature.

Test Basis

The lubricant performance tests were conducted by running a specially instrumented industrial tilting pad thrust bearing in a bearing test machine, and measuring the lubricant film thickness, pad temperatures and bearing power loss as functions of the bearing load and shaft speed. A line diagram of the test machine is shown in Figure 2, while the test bearing is shown in Figure 3.

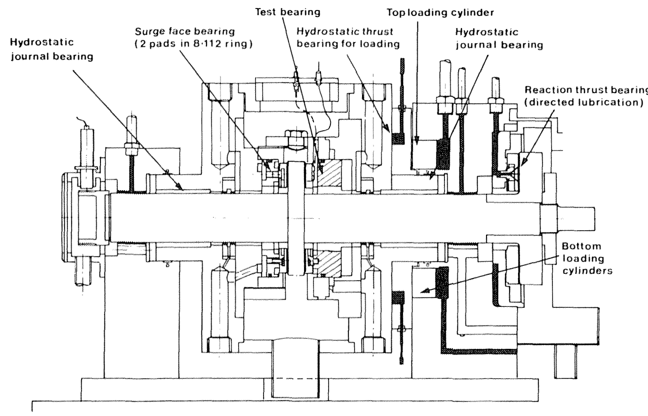


Figure 2. Sirius Test Machine.

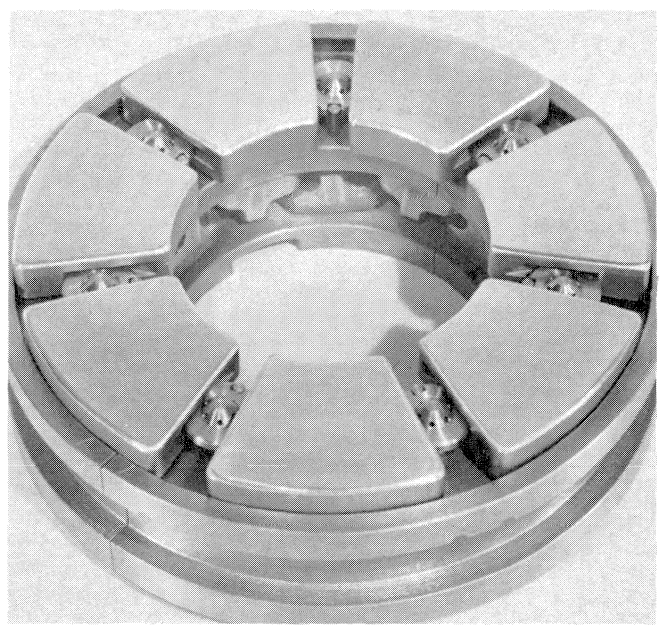


Figure 3. Test Bearing 7125 CQ as Fitted to the Test Machine.

Test Bearing

The test bearing was a single acting, tilting pad thrust bearing with the following characteristics:

- seven thrust pads with center pivots
- outside diameter—127 mm (5.0 in.)

- inside diameter—63.5 mm (2.5 in.)
- surface area—6900 mm^2 (10.7 in.^{2})

The test bearing could be operated in either a flooded or directed lubrication mode. Most tilting pad thrust bearings employed in industrial turbomachinery are lubricated by pressurized oil in a flooded bearing housing. In this flooded mode of operation, the bearing power loss may be resolved into two components: viscous shear losses and parasitic turbulent losses (churning). Directed lubrication thrust bearings are designed to minimize the churning losses by spraying the lubricant directly onto the thrust pads and providing a large oil discharge area so that the oil does not accumulate in the bearing housing. With this design, the thrust collar does not spin in an oil bath, and thus the churning losses are minimized.

Test Lubricants

Three lubricants were evaluated in the tests. First, an ISO VG 32 mineral oil was selected as the baseline lubricant, since this class of oils is typically used in industrial turbomachinery with tilting pad thrust bearings. An adipate diester synthetic lubricant of the same viscosity grade, as well as a thinner (ISO VG 10) adipate diester lubricant, were selected for comparison to the mineral oil. For convenience, these lubricants will be called MIN 32, SYN 32 and SYN 10 in the remainder of this paper.

Instrumentation

Oil Film Thickness

The surface of a thrust pad is subjected to very high pressures and to significant temperature gradients. The film thickness transducers fitted in this environment are required to maintain both their calibration and mechanical position within close tolerances, in order to permit valid comparisons among the different oils. The transducers should allow film thickness on the order of 12 microns (0.0005 in.) to be measured with an accuracy of better than 5%. Capacitive transducers using a guard ring design were selected to meet these requirements. The transducers are constructed of steel and ceramic, to give them similar structural and thermal properties to those of the pad material and to meet the electrical constraints of the application. To ensure that they do not move during operation, they are mechanically held in the pad by a high interference.

The transducers are calibrated separately for each oil by using blocks with recesses of known depth ground into their surfaces.

Capacitive transducers are accurate and stable, provided that there is no change in the dielectric constant of the fluid. It is therefore necessary for the transducers to be fitted in positions where no cavitation or breakdown of the hydrodynamic film is likely to occur. This is easily achieved in tilting pad thrust bearings, provided that the transducers are not installed too close to the trailing edge of the pad. Film breakdown may occur at this location when high loads and speeds are encountered, because of thermal crowning of the pad, which results in a local divergent zone. It is often more difficult to arrange suitable positioning in journal bearings because the cavitated zone may be less well defined and more dependent upon operating conditions.

Temperature

Babbitt surface temperatures were measured with copper constantan thermocouples soldered with babbitt into the lining and measuring the temperature within 0.5 mm (0.02 in.) of the pad surface. The thermocouples were installed prior to the final machining operation, in order to minimize any effect that the instrumentation might have had on the bearing operating characteristics.

Power Loss

A very sensitive method of measurement is required in order to permit identification of small differences in power loss. Deduction of power loss from oil flow and oil temperature is frequently used, but this technique provides insufficient accuracy for this test. The inaccuracy of this method may be attributed primarily to conductive and radiative heat losses, which are not taken into consideration. Furthermore, the results are very sensitive to the location of the temperature sensors. It has been demonstrated that the calculated power loss may be varied by as much as 30% by changing the position of the inlet and outlet thermocouples. In addition, the resolution of a thermocouple-based system is inadequate for this application when the temperature rise is small.

To circumvent these difficulties, the power loss was measured directly by measuring the bearing reaction torque and the test machine speed. The special design of the test machine permits direct measurement of the bearing reaction torque as the whole test head floats on hydrostatic bearings and the thrust load is applied with a hydrostatic reaction plate. The support journals are outside of the torque measuring system.

Test Envelope

The test machine was run at speeds from 5,000 to 15,000 rpm, while the bearing loads ranged from 10,000 to 40,000 Newtons (2250 to 9000 pounds). This corresponds to mean sliding velocities from 24.4 to 73.1 m/sec (80 to 240 ft/sec) and to a specific load range of 1.45 to 5.80 MPa (210 to 840 psi).

TEST RESULTS

Power Loss

The bearing power losses at two representative values of pad loading are presented in Figures 4a and 4b as functions of the lubricant, the lubrication mode [flooded lubrication (F/L) vs. directed lubrication (D/L)] and the mean sliding velocity of the bearing. The oils of similar viscosity grade demonstrated similar power loss characteristics, as would be expected from hydrodynamic theory. At lower speeds, the SYN 32 consumed slightly more power than the MIN 32. This result may be attributed to the higher specific heat and higher viscosity index of the diester lubricant, which lead to a greater mean viscosity for the SYN 32 in the bearing at lower speeds. In addition, the

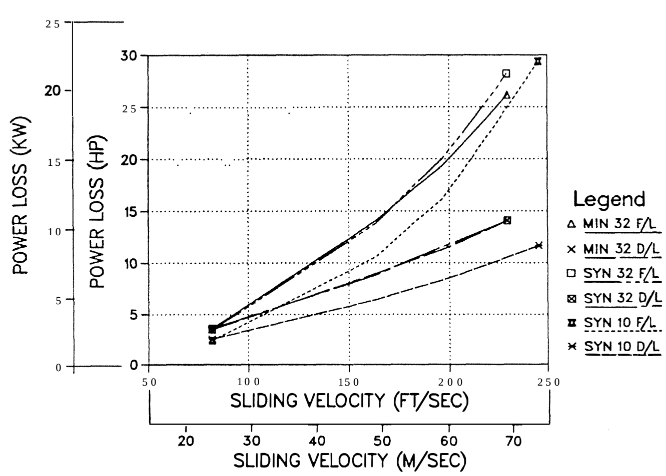


Figure 4a. Power Loss Vs. Sliding Velocity at 1.42 MPa (206 psi) Pad Load.

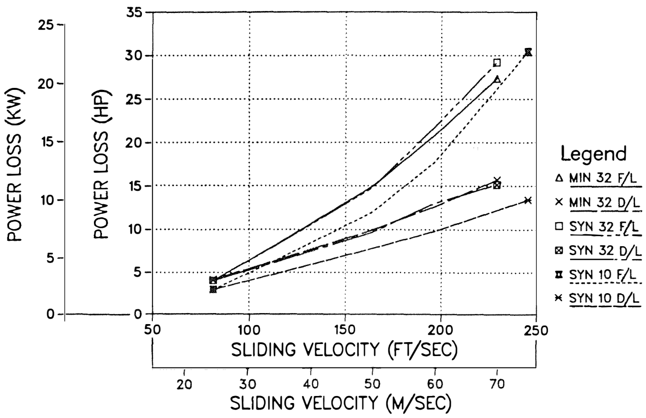


Figure 4b. Power Loss Vs. Sliding Velocity at 2.84 MPa (412 psi) Pad Load.

SYN 10 provided a 20 to 30% reduction in bearing power loss, as would be expected from viscometric considerations.

In the directed lubrication mode, the power consumption of the bearing was considerably reduced as a result of the near-elimination of the parasitic churning losses on the collar. The power saving increases with speed, and savings as high as 50% are shown at shaft speeds approaching 20,000 rpm [mean sliding velocity approaching 100 m/sec (325 ft/sec)].

Oil Film Thickness

The film thickness results are presented in Figures 5a and 5b. These results confirm the field experience, in that the synthetic oils showed a considerable increase in film thickness over that of the mineral oil. The surprising result that the SYN 10 ran with a thicker film than the MIN 32 is not explicable using conventional hydrodynamic theory, assuming the oil to be Newtonian and isoviscous. In an attempt to understand this phenomenon, the pad tilt was calculated. This calculation revealed that the synthetic oils consistently cause the pad to operate with a lower tilt ratio; i.e., the pad is more nearly parallel to the collar surface than would be the case with the mineral oil. This indicates that the higher specific heat of the synthetic lubricants results in a smaller temperature gradient across the pad, and thus the viscosity ratio from front to back of the pad is lower. Under these circumstances, the tilt ratio must be lower in order to permit the center of pressure to remain over the pivot.

Although this partially explains the differences in load

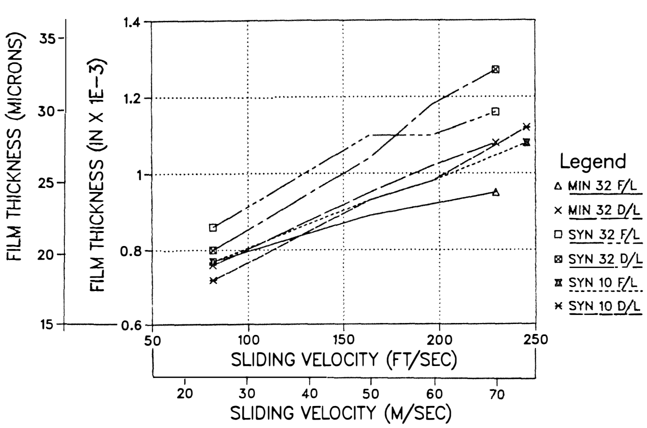


Figure 5a. Film Thickness Vs. Sliding Velocity at 1.42 Mpa (206 psi) Pad Load.

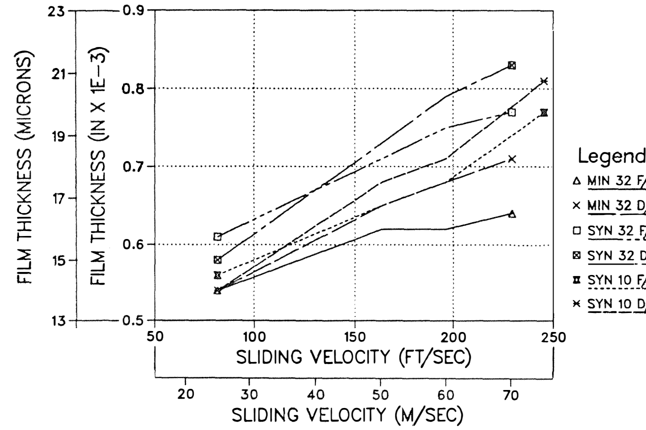


Figure 5b. Film Thickness Vs. Sliding Velocity at 2.84 MPa (412 psi) Pad Load.

capacity, it is by no means a total explanation. The increase in film thickness is far too great for this simple explanation to be totally satisfactory. Furthermore, in the directed lubrication tests, there is firm evidence that the pads require more synthetic lubricant than the equivalent mineral oil. This, paradoxically, would indicate a higher tilt ratio. One is left with the possibilities that the synthetic oils either become highly non-Newtonian or that there are significant boundary layer effects under bearing operating conditions. Such theories might be tested in a specially designed test rig, in which the shear rate and net hydrodynamic effects may be varied with respect to each other. This could be achieved if a journal bearing test rig were designed to allow the bush and/or the load to be rotated independent of the shaft at varying speeds.

In addition, directed lubrication provides a small increase in film thickness over that measured for flooded operation. This results from reduced bearing power losses, which lower the oil operating temperature and thereby increase its viscosity in the bearing. It is this increase in viscosity that yields the thicker lubricant films measured in the directed lubrication mode.

Maximum Pad Temperatures

The trend of the synthetic oils to run at lower maximum pad temperatures is shown in Figures 6a and 6b and is a result of the higher specific heat of these oils than that of the mineral oil. The SYN 32 typically operated at 1-5°C (1.8-9.0°F) lower maximum pad temperature than MIN 32. This difference is small, as the mechanism tends to be a self-stabilizing one. A

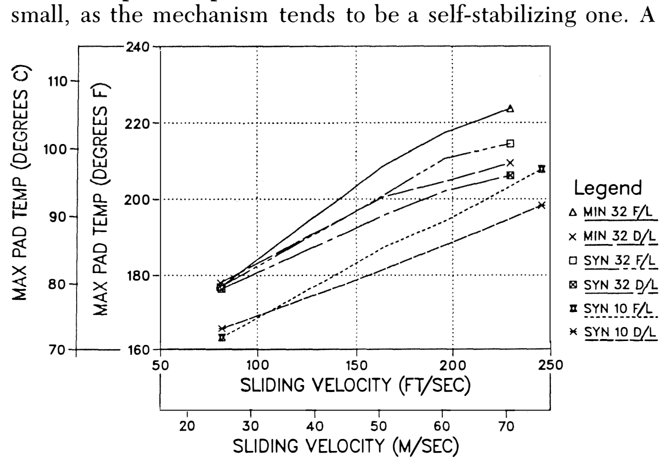


Figure 6a. Maximum Pad Temperature Vs. Sliding Velocity at 1.42 MPa (206 psi) Pad Load.

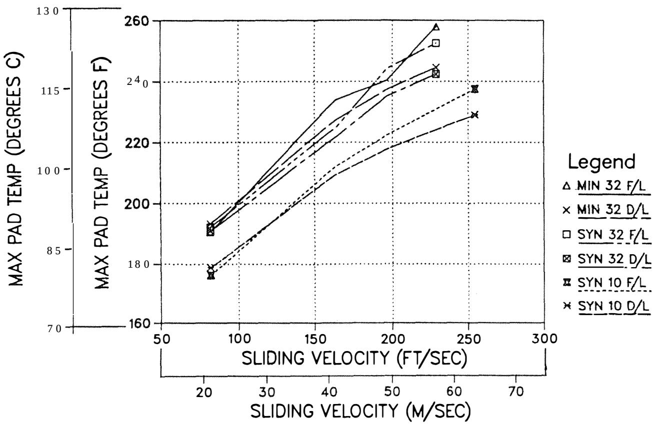


Figure 6b. Maximum Pad Temperature Vs. Sliding Velocity at 2.84 MPa (412 psi) Pad Load.

smaller temperature rise in the synthetic lubricant results in a higher viscosity, and consequently in more shearing work being done in the hydrodynamic film. This tends to raise the temperature to a stable position close to that of the mineral oil.

Clearly, the thinner oil (SYN 10) will tend to run cooler for the reasons stated above. Maximum pad temperature reductions of 7-12°C (12-22°F) were observed with SYN 10, relative to MIN 32.

As noted earlier, directed lubrication also lowers the maximum pad temperature. Reduced bearing power loss and the consequent reduction in heat generation within the bearing are responsible for the cooler operation.

CHOICE OF LUBRICANT AND LUBRICATING SYSTEM

Designers incorporating a thrust bearing into a mechanical system are required to satisfy both safety and short- and long-term economic requirements (i.e., acceptable purchase and running costs). It is often necessary to make compromises between these requirements. The parameters that affect the bearing's ability to provide extended safe operation are film thickness and temperature, and any action that will lower the temperature or increase the film thickness will tend to improve the bearing life. The economics are affected by initial cost (including pumps and the cooling system) and by running costs, including both power consumption and maintenance requirements.

Therefore, a case for considering the use of an adipate diester synthetic lubricant may be made on three grounds:

1. Increased reliability due to increased film thickness;
2. Reduced maintenance cost due to extended periods between oil changes (3-5 times as long), because of the increased stability of the lubricant; and
3. Reduced power consumption by the use of a thinner grade of synthetic lubricant.

Furthermore, the directed lubrication system should also be considered, as it offers the following advantages:

1. At high speeds the bearing power consumption (and therefore the running cost) is much lower than for an equivalent flooded bearing.
2. At high speeds the bearing operates with thicker films and lower temperatures, resulting in greater bearing reliability.
3. The total bearing package cost, including the pumps and coolers, frequently is lower than that for an equivalent flooded bearing system.

Another problem that a designer frequently faces is the uprating of a bearing because of a change in design or because of lack of precision in the original design procedure (which led to inaccurate load calculations). This is a significant problem, especially in balanced pumps and turbines, where small changes in design parameters may lead to large changes in thrust load.

Let us now consider some specific examples of operating conditions, in order to draw more general conclusions.

First, consider a low speed bearing running at synchronous speed (3600 rpm) and being lubricated with mineral oil. A lightly loaded bearing operates with a large film thickness and low pad temperature. In addition, the power loss is small and would not be significantly affected by a change to a synthetic oil of similar grade. There are therefore two possible actions which may reduce operating costs. First, a change to a synthetic lubricant of the equivalent grade might reduce the total maintenance costs by significantly extending the oil drain interval. Alternately, a thinner synthetic oil could be used to provide an improvement in plant efficiency by reducing the power absorbed by the bearing.

As another example, consider a bearing operating at low speed, but with a high load (a condition that might occur as a result of uprating, plant malfunction, discrepancy between calculated and actual loads, or even a decision to heavily load a bearing in order to reduce its size and power loss). In this case, a significantly higher load might be carried by the bearing (up to 30% higher) without reducing the film thickness, if a mineral oil were replaced by a synthetic oil of the same grade.

At high speed (for example, 18,000 to 20,000 rpm), the bearing limitation is usually temperature, rather than film thickness. In addition, the power absorbed by a bearing increases dramatically with an increase in speed. The use of the directed lubrication system tested would significantly reduce the bearing power consumption (by up to 50%), the operating temperature, and the oil flow requirement in such an application. These benefits would combine to yield an increased operating margin of safety while reducing operating costs.

SUMMARY AND CONCLUSIONS

Field experience with fully formulated adipate diester synthetic lubricants indicates that these oils produce stronger, thicker films than do their mineral oil counterparts. To verify and quantify the benefits of using these synthetic lubricants, an experimental investigation has been performed in a tilting pad thrust bearing. The primary results and conclusions of this investigation follow.

We can conclude, based upon the 1-5°C (1.8-9.0°F) lower pad temperatures and 5-25% greater film thickness provided by SYN 32 relative to MIN 32, that a greater margin of safety is afforded by replacing mineral oil with a comparable grade of adipate diester synthetic lubricant in hydrodynamic thrust bearings. Similar improvements in the margin of safety in turbomachinery journal bearings can be expected when SYN 32 is substituted for MIN 32. Unfortunately, monetary value cannot be easily assigned to this greater margin of safety; therefore, economic justification for replacing MIN 32 with SYN 32 cannot be readily calculated.

The energy savings associated with the use of SYN 10 in place of MIN 32 is significant—a 20-30% reduction in bearing power loss was demonstrated. Furthermore, oil film thickness measurements indicate that SYN 10, despite its lower viscosity, produces film thickness equal to or greater than that of MIN 32. Based on these test results, the use of SYN 10 in tilting pad thrust bearings can be recommended, and the user will gain energy savings and lower operating temperatures without a sacrifice in bearing safety.

Due to the unusual behavior of synthetic lubricants in terms of developed oil film thickness, the scale-up of test results to larger size bearings is not recommended without further confirmatory tests. However, based upon all previous work with synthetic lubes, we expect that similar behavior of synthetic lubricants will occur in larger size bearings.

In conclusion, significant economic and technical benefits may be obtained by choosing the correct synthetic lubricant, and further advantages may be obtained with directed lubrication. The appropriate choice of bearing can lead to a lower cost, more compact unit, running with a higher level of reliability and consuming less power without any obvious drawbacks.

ACKNOWLEDGEMENTS

The authors would like to thank the following people for their assistance in this project:

A. J. Leopard
Glacier Metal Company Limited

I. W. Miyoshi
Exxon Chemical Company
Central Engineering Division

J. Zielinski
Exxon Chemical Company
Paramins Technology Division