

**INFLUENCE OF OIL INJECTION METHOD ON
THRUST BEARING PERFORMANCE AT LOW FLOW CONDITIONS**

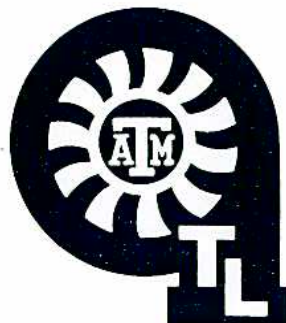
by

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ABSTRACT

The advantage of operating a thrust bearing in an evacuated cavity has been well documented in the literature. By draining the oil from the bottom of the housing and eliminating standing oil around the thrust collar, horsepower reductions of up to 50 percent can be realized. The decision to operate in an evacuated cavity is the first step in designing a low loss bearing. The recent trend in the industry has been to further reduce thrust losses by lowering the quantity of oil supplied to the bearing. This study investigates the impact of the oil injection method on thrust bearing performance when operating in an evacuated housing under low oil supply conditions. Several bearings were tested with identical geometry under duplicate operating conditions with the method of oil injection varied in each test to isolate its influence on performance. Tests were done to determine the starved film flow at various speeds for each lubrication method. Based on the results of the starved flow test, the bearings were evaluated at full flow, incipient starvation, and fully starved conditions. The performance of the bearings tested was nearly identical around a sliding velocity of 12,300 ft/min. At sliding velocities above and below this speed, there was a load at which the temperature of the nozzle, pocket, and groove bearing was identical. The coolest configuration to the left of the crossover tended to be the hottest to the right of this point.

INTRODUCTION

The reliability of tilting pad thrust bearings has made them the bearing of choice for high speed rotating machinery for close to a century. The standard method of lubrication is to feed oil through slots in the back of the bearing retainer toward the bore of the bearing. A tangential drain is normally located at the top of the housing, which allows the housing to fill with oil. Flow is controlled by an orifice located at either the oil inlet to the housing or at the tangential discharge. This method of lubrication is commonly referred to as flooded cavity lubrication and it is still

widely used today. The hydraulic pumping action of the thrust collar pulls the oil from the bearing bore to the thrust pads where a hydrodynamic film is generated. The hydrodynamic film pressure and centrifugal force propel the oil across the bearing and into the discharge area of the thrust cavity.

In the 1970s, the design of tilt pad thrust bearings was greatly influenced by the work of Bielec and Leopard (1970) and New (1974), which illustrated the dramatic impact the thrust bearing can have on the overall performance of rotating equipment. By eliminating the backpressure in the thrust housing through the use of a bottom drain, it was possible to reduce the overall horsepower consumption of the bearing. Parasitic losses associated with the delivery and ejection of the oil in the standard configuration generated heat that did not contribute to the operation of the bearing. Depending on the mean sliding velocity of the bearing, up to a 60 percent reduction of losses was reported.

When the thrust housing incorporates a bottom drain for oil discharge, it is referred to as an evacuated thrust cavity. With this type of system, it is customary to supply oil to the leading edge of each pad through a variety of methods. This is done to ensure that cool lubricant is available in the regions required for development of the hydrodynamic film. When the oil supply method injects oil to the leading edge of each pad, the bearing is said to have directed lubrication. The configuration of the oil inlet method is a function of the bearing design.

The trend in recent years has been to refine the oil injection method to further reduce oil consumption and horsepower losses. This increases the overall performance of the rotating system while reducing the size and cost of the lubrication system. Gardner (1998) showed that reductions in oil flows beyond a threshold point results in sharp increases in pad metal temperatures as the bearing reaches a starved flow condition. A relatively small percentage of the oil flow to the bearing is actually required for lubrication (Elwell, 1971), with the balance of the flow required for heat dissipation. This leads to a concern that as oil flows are reduced, high metal temperatures may compromise the overall reliability of the bearing.

Mikula and Gregory (1983) did a study of thrust bearing lubrication supply methods, followed by additional testing by Mikula (1985, 1988). Various methods of oil injection were evaluated and bore out the conclusions reached by Bielec and Leopard (1970) that an evacuated cavity and directed lubrication contribute to a dramatic reduction in horsepower loss and oil flow without compromising bearing reliability. Their results are somewhat less informative when attempting to compare the relative performance of directed lubrication supply methods. The test configuration used by Mikula and Gregory in 1983 consisted of an active and an inactive thrust bearing with a common drain, making it difficult to isolate the contribution of each bearing to horsepower consumption. Likewise, differences in the geometry of the bearings tested may have contributed to the observed variation in performance.

Testing was done to isolate the influence of four oil injection methods on thrust bearing performance, particularly at low flow conditions. The overall geometry of the bearings evaluated were identical, eliminating the influence of pivot offset and pad aspect ratio.

STARVED FILM FLOW

A generalized schematic of the flow of oil into and out of a thrust pad was presented by Gardner (1998) and is shown in Figure 1. Flow into the leading edge of the pad is designated Q1, while Q3 represents the flow exiting the trailing edge. The Q3 flow mixes with the supply oil to form the Q1 flow for the next pad. Hydrodynamic pressure developed in the film results in inner (Q2) and outer (Q4) edge flows. The pumping action of the thrust collar prevents the Q2 flow from exiting the bearing and it is recirculated to the downstream pads.

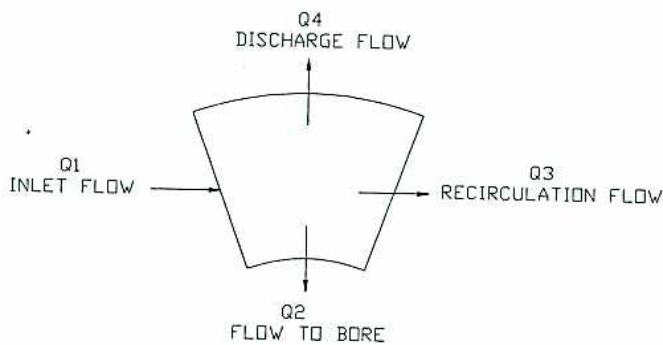


Figure 1. Oil Flow Nomenclature.

The flow that exits the pad radially is designated Q4. When the flow rate to a bearing is less than Q4 multiplied by the number of pads, the bearing is starved. The starved film point can be determined experimentally by plotting pad metal temperature against flow for a given speed and load condition. Above the starved film flow point, pad metal temperatures remain relatively constant versus flow. As flow rates are reduced below the starved film point, pad metal temperatures begin to rise quickly. Additional reduction in flow will result in failure of the bearing.

TEST DESCRIPTION

Test Facility

The rig used has been described in previous literature (Gardner, 1998) and is shown in Figure 2. In brief, the test stand is driven by two 500 hp, variable speed, DC motors through a gearbox. A hydraulic load is applied to the back of the slave thrust bearing running against a 15 inch collar. The load is transmitted through the shaft to the test bearing thrust runner. A pneumatic control valve is used to meter flow to the test cavity. Maximum speed for this unit is 10,000 rpm.

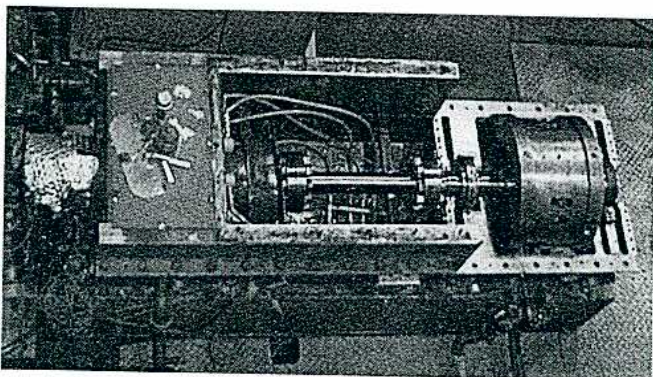


Figure 2. Thrust Rig with Cover Removed Exposing Test Housing.

The test cavity (Figure 3) is evacuated and allows the installation of a single thrust bearing. During testing, flow is supplied only to the test side of the housing. Horsepower loss was determined by calculating the heat balance on the oil flow from inlet to discharge.

Oil outlet temperature is determined with a thermocouple installed in the "A" oil drain. Overflow drains (B and C) located outboard of the test cavity join a common header below the thermocouple installation point. Initial tests on the groove bearing indicated discrepancies in measured horsepower to published data. Inspection of the rig during operation indicated that, under certain operating conditions, flow was being diverted to the outboard drain B. At low speeds and low loads, the bearing was not able to pass all the inlet oil across the thrust face. The balance flowed through the bore of the bearing and out the back of the thrust cavity. A second thermocouple was installed in the header at the point where the A and B drains converged. Heat balances performed with this discharge temperature correlated with the manufacturer's data. All horsepower loss calculations in this report are based on the lower drain temperature.

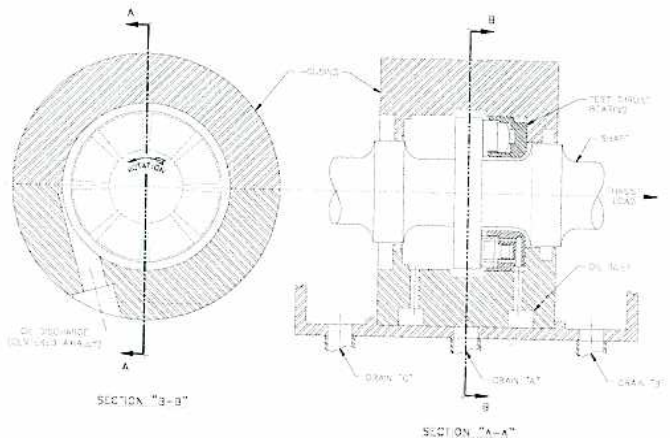


Figure 3. Cross Section of Thrust Test Cavity and Housing Drains.

Test Bearings

In order to isolate the oil injection method, care was taken to ensure that only this feature was varied. The bearings tested were six-pad, self-leveling style with a .6 pivot offset. Pivot offset is the location of the pad support as a percentage of the total pad arc length from leading edge to trailing edge. The outside diameter of the thrust surface was 10.5 inch with an inside diameter of 5.25 inch for an effective thrust area of 55.2 sq in. The pad backing material was steel with an ASTM B-23, Grade 2 babbitt surface.

Thermocouples were installed in three of the six pads, embedded in the babbitt approximately .03 inch below the pad surface. Four thermocouples were placed in each instrumented pad, located at the 60/75, 75/75, 50/85, and 85/85 positions. The position numbers are the percentage of the radial/circumferential pad dimensions from the lower corner of the pad leading edge, respectively.

Four methods of oil injection were evaluated. All the bearings ported supply oil to each pad from an annular groove formed between the bearing retainer and the test cavity. The pocket, directed lube, and nozzle arrangements injected oil into an area upstream of the leading edge of the pad. The groove design fed oil into the leading edge of the pad. All four of the modes evaluated fall under the classification of *directed lubrication*, even though this term was used to describe one of the methods tested.

Pocket

The pocket bearing is shown in Figure 4. A cross section of the bearing is illustrated in Figure 5. Oil is routed from the supply annulus to the front face of the bearing through cross drillings in the retainer. The oil injection point is at the inner diameter (ID) of the bearing. The overall arc length of each pad is greater than a standard six-pad bearing. The leading and trailing edges are

profiled so that when assembled, adjacent pads combine to form a recessed pocket with a single oil inlet hole. The pocket does not contribute to the effective thrust area of the bearing. The oil inlet port in the retainer directs a stream of fresh oil through the hole formed by the pads and into the pocket area. Dams at the inner and outer diameter of the pocket contain the oil and fill the leading edge of the downstream pad.

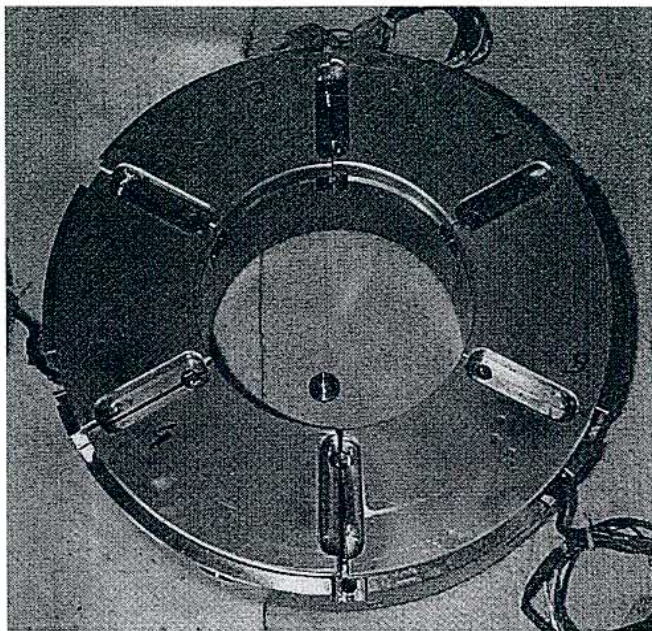


Figure 4. Pocket Thrust Bearing.



Figure 5. Cross Section of Pocket Thrust Bearing.

Nozzle

The nozzle test bearing is shown in Figures 6 and 7. The base ring of the pocket test bearing was modified by installation of nozzle blocks between pads of standard arc length. The nozzle was attached with bolts from the back of the retainer. Cross drilling in the nozzle redirected the lubricant radially from the inside diameter to the outside diameter of the bearing. A series of four oil inlet holes intersect the radial feed hole in the nozzle to spray the oil perpendicular to the thrust collar, injecting oil along the length of the leading edge of the pad.

Directed Lube

The directed lubrication arrangement (Figures 8 and 9) was obtained by removing the nozzles and installing a plug in the through hole for the attachment bolts. The point of oil injection into the bearing was at the same axial location as the pocket configuration. The oil was allowed to spray into the open groove area between each pad without any additional flow direction mechanism. The pads used in this configuration were the same as those used in the nozzle arrangement. The axial distance of the injection point relative to the face of the thrust surface was considerably larger than normal practice to facilitate using the

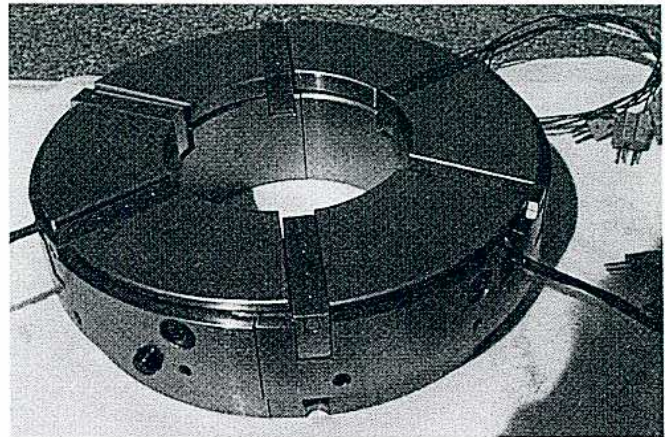


Figure 6. Nozzle Thrust Bearing.

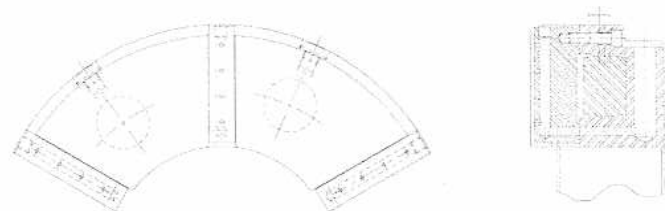


Figure 7. Cross Section of Nozzle Thrust Bearing.

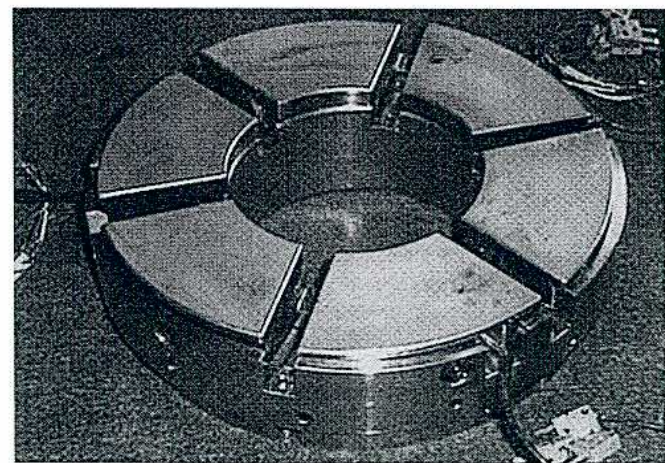


Figure 8. Direct Lube Thrust Bearing.



Figure 9. Cross Section of Direct Lube Thrust Bearing.

Groove

The groove test bearing is shown in Figure 10. A groove was milled in the pad directly behind the leading edge with dams at the inner and outer edges (Figure 11). A through hole at the outer diameter of the groove is provided for oil inlet. A floating nozzle connects the pad to an oil supply hole in the base ring. Lubricant flows from the oil supply annulus, through the nozzle and into the groove in the pad. A shallow slot is milled at the inner edge of the

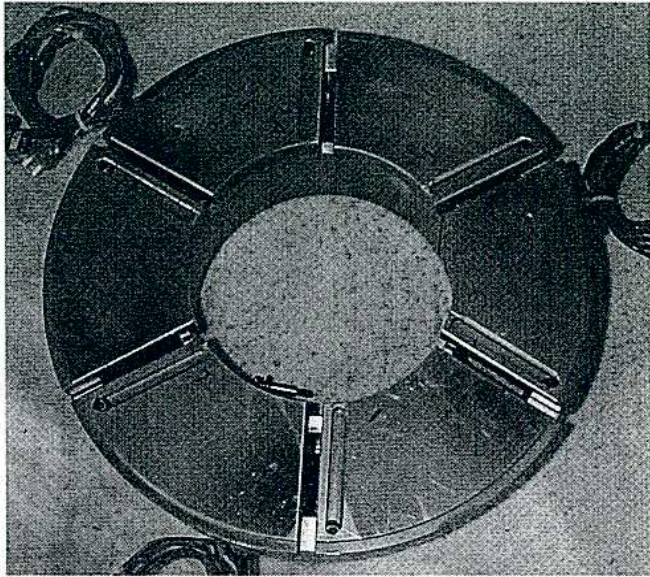


Figure 10. Groove Thrust Bearing.

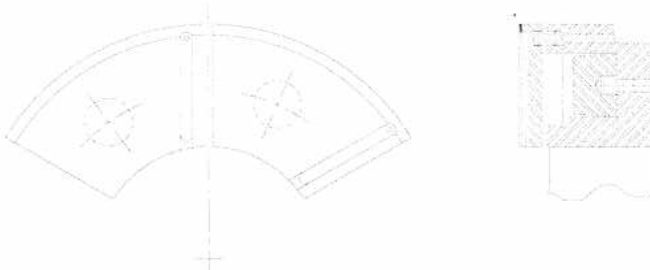


Figure 11. Cross Section of Groove Thrust Bearing.

Test Procedure

Two sets of tests were performed on each bearing. The low flow test was designed to determine the starved flow point for each bearing design at various operating speeds. The rig was operated from 5000 rpm to 9000 rpm in 1000 rpm increments. The load was held constant at 500 psi with an initial oil flow rate of 16 gpm. Flow was reduced in 2 gpm increments and held for 10 min to allow the system to stabilize. Pad temperatures and horsepower loss were then recorded for that flow condition. Testing was halted when a temperature of 280°F was recorded at any thermocouple location or if pad temperatures did not stabilize.

The load tests were done once the starved film point was determined to evaluate the performance of each bearing under full flow, incipient starvation, and fully starved conditions. The speeds for this series of tests were 3600 rpm, 6000 rpm, and 8000 rpm. The load was varied from 100 psi in 25 psi increments with a constant oil flow of 10 gpm. Testing was halted when a temperature of 280°F was recorded at any thermocouple location or if pad temperatures did not stabilize. At loads above 700 psi, the maximum allowable temperature was limited to 250 psi to reduce the chance of failing the bearing.

The lubricant used was an ISO VG 32 mineral oil. Inlet temperature was maintained at 120°F for all tests.

TEST RESULTS

Low Flow Test

Plots of the maximum 75/75 temperature versus flow are shown in Figures 12 through 14 for 5000, 7000, and 9000 rpm. The calculated starved film flows are shown as a vertical line in each chart. The measured starved flow point for each inlet configuration is indicated by an upturn in the temperature curve. The directed

lube bearing exhibited high operating temperatures during the low flow tests and was not run above a speed of 7000 rpm. The discussion for this series of tests will be limited to the nozzle, pocket, and groove bearing due to the limited output from the directed lube configuration.

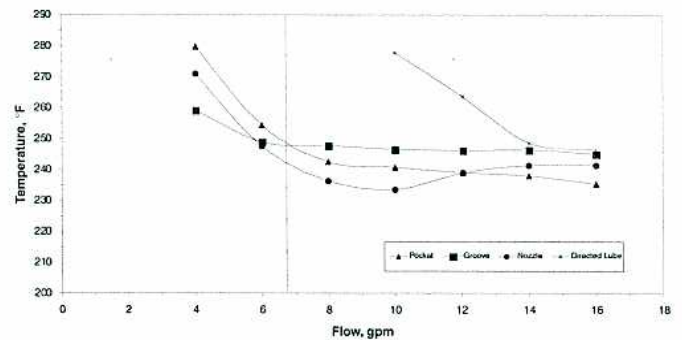


Figure 12. Maximum 75/75 Pad Temperature Versus Flow (5000 rpm - 500 psi Load).

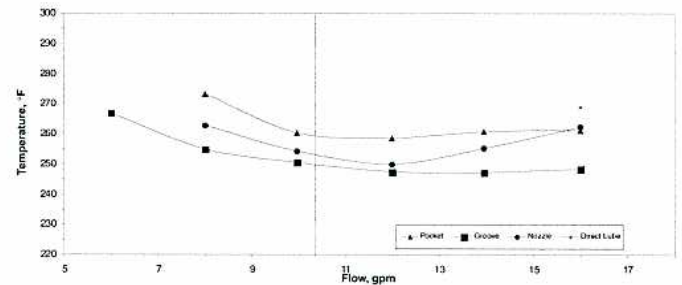


Figure 13. Maximum 75/75 Pad Temperature Versus Flow (7000 rpm - 500 psi Load).

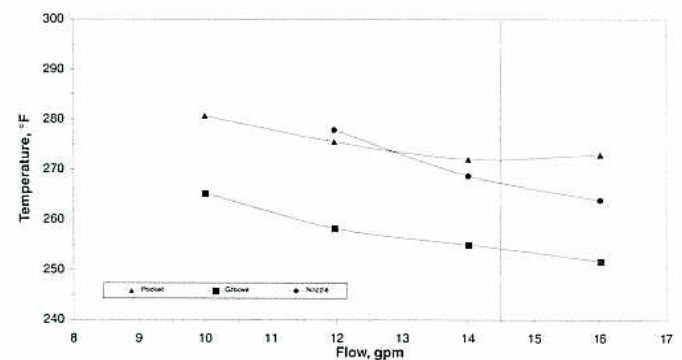


Figure 14. Maximum 75/75 Pad Temperature Versus Flow (9000 rpm - 500 psi Load).

At speeds below 9000 rpm, starved film point for the nozzle, pocket, and groove bearings occurred at flows within a range of 2 gpm. The predicted starvation point fell within the 2 gpm gap for all cases except 6000 rpm, where all bearings exhibited starvation prior to prediction. At all speeds, the hottest bearing starved at lower flows than the coolest operating bearing, even though the maximum metal temperature variation was only 21°F in the starved flow region at a speed of 9000 rpm. The starvation point was identical for all three bearings at a speed of 6000 rpm.

The temperature curve for the nozzle bearing at 5000 rpm (Figure 12) shows a reduction in metal temperature prior to the starved film point, a characteristic of turbulence in the film (Capitao, 1976). The pocket and nozzle bearings exhibited turbulence during the low flow testing, with a more pronounced dip in the temperature curve for the nozzle inlet. Turbulence was not observed with the groove bearing at any speed.

The temperature of the nozzle and pocket bearings rose quickly at flows below the starved film point. The slope of the temperature curve for the groove inlet was flatter than the other configurations in the starved flow region. At speeds above 7000 rpm, the groove bearing was able to operate at a slightly lower flow (2 gpm), which is consistent with data published by Mikula in 1988.

The horsepower loss curves for the low flow tests at 5000, 7000, and 9000 are presented in Figures 15 through 17. Below the starved flow point, the horsepower loss decreased with a reduction in flow. The variation in horsepower loss at the lowest flow conditions ranged from a minimum of .4 hp to a maximum of 2.7 hp.

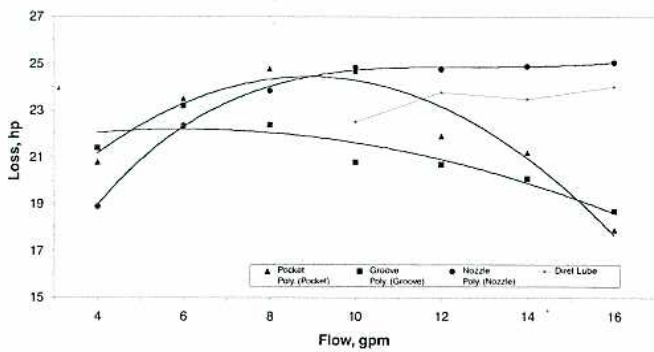


Figure 15. Loss Versus Flow (5000 rpm – 500 psi Load).

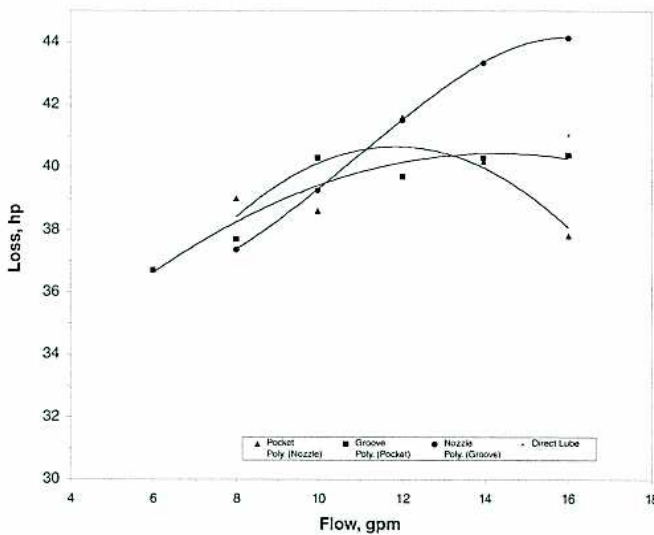


Figure 16. Loss Versus Flow (7000 rpm – 500 psi Load).

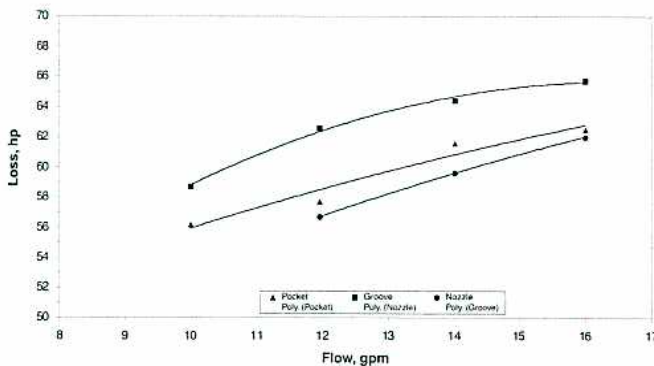


Figure 17. Loss Versus Flow (9000 rpm – 500 psi Load).

The power loss of the nozzle bearing increased with flow at all speeds. Below 8000 rpm, the groove and the pocket bearing exhibited a reduction in the horsepower loss as the flow was

bypassing the bearing cavity, lowering the oil discharge temperature used in the heat balance equation. The groove and pocket bearings injected oil into features that were machined within the pad that contained dams at the outer diameter. At the lower speeds, the volume of oil that could physically pass through these two bearings was limited by the backpressure developed in the film. The pumping action of the collar was not sufficient to recirculate the Q2 flow, and it exited the thrust cavity to the outboard drain through the bore of the bearing. This bypass was observed during tests at the lower speeds, and decreased as the speed was increased or the flow was lowered. An oil trap and thermocouple were installed at the back discharge of the test cavity in an attempt to determine the quantity of the oil bypass; however the results were not consistent. Since the focus of the testing was operation at the lower flow rates, it was felt that the bypass at the higher flows would not impact the study.

The relationship between operating temperature and horsepower loss is evident in the test results. Below the starved film flow, the cooler operating bearing resulted in the higher horsepower loss, although the difference was modest.

Load Test

Figures 18 through 20 show the maximum 75/75 pad temperature versus load for the constant speed load tests. The oil flow to each bearing was held constant at 10 gpm for all test speeds. This flow was chosen to observe the performance of the bearings at full film, incipient starvation, and fully starved film as loading increased. At each speed, there was a load at which the temperature of the nozzle, pocket, and groove bearing was identical. The coolest configuration to the left of the crossover tended to be the hottest bearing to the right of this point. The directed lube bearing ran hotter than the other bearings and worse than calculated in all the load tests. The discussion for this series of tests will focus on the nozzle, pocket, and groove bearing.

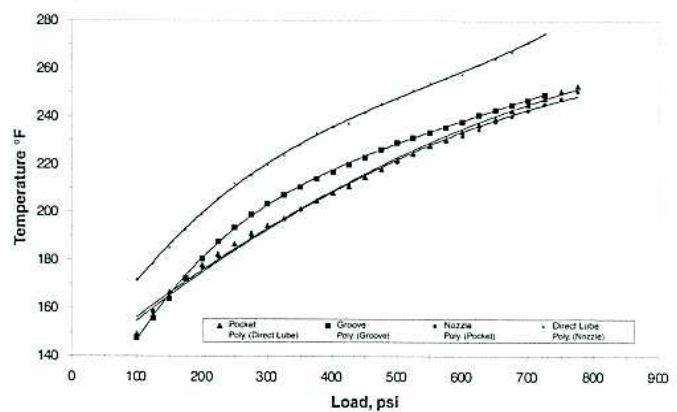


Figure 18. Maximum 75/75 Pad Temperature Versus Load (3600 rpm – 10 gpm).

At 3600 rpm (Figure 18), the bearings are operating well above the starved film point. The groove bearing ran the hottest in this test with a maximum temperature difference of 10°F at 325 psi. The temperatures converged at loads above and below this point.

The temperatures of the pocket, nozzle, and groove bearings were nearly identical for all loads when operating at 6000 rpm (Figure 19). At this speed, the low flow tests indicated that the bearings were operating at starved film point. The maximum temperature difference was 8°F at a load of 700 psi.

The crossover load point for the fully starved test at 8000 rpm occurred at 200 psi. The temperatures of the pocket and the nozzle configurations are nearly identical up to a load of 450 psi. The groove bearing ran cooler at the higher loads with a maximum

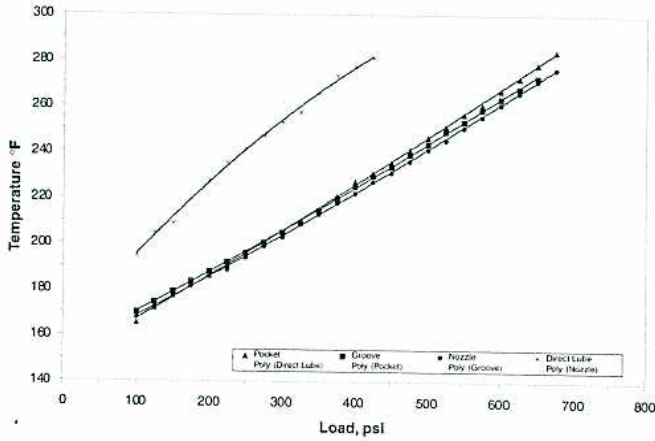


Figure 19. Maximum 75/75 Pad Temperature Versus Load (6000 rpm - 10 gpm).

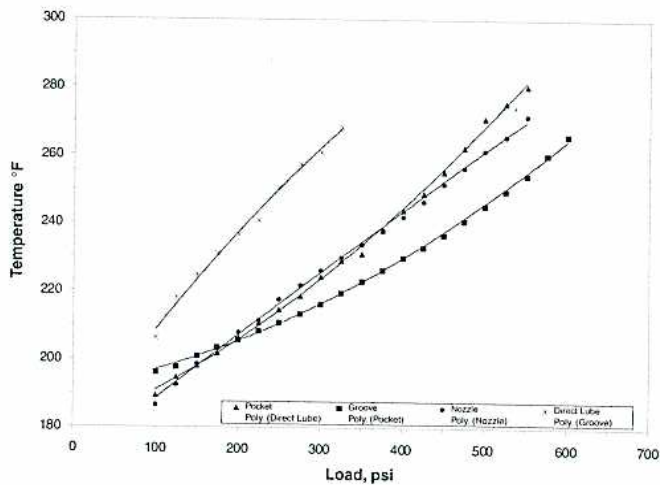


Figure 20. Maximum 75/75 Pad Temperature Versus Load (8000 rpm - 10 gpm).

The horsepower losses for the three tests are shown in Figures 21 through 23. The hotter bearing in each test resulted in the lowest horsepower, although the difference on average was less than 3 hp.

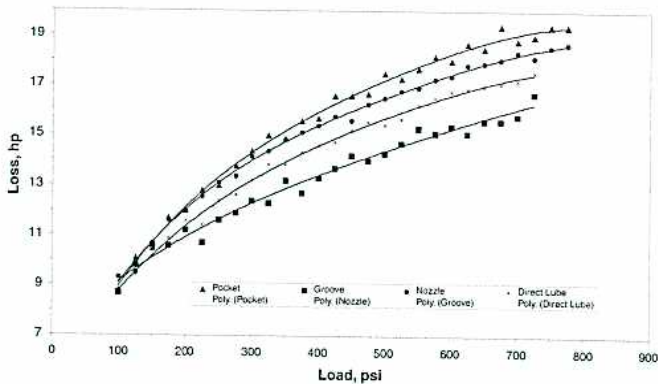


Figure 21. Loss Versus Load (3600 rpm - 10 gpm).

Figures 24 through 26 show the maximum 75/75 temperature versus mean sliding velocity for 100 psi, 300 psi, and 500 psi, respectively, for the pocket, nozzle, and groove bearings. This is a consolidation of data obtained from the constant speed, variable load tests. The mean sliding velocity is obtained by calculating the mean circumference of the bearing in feet and multiplying by the rotational speed of the collar. The flow for all test points was 10

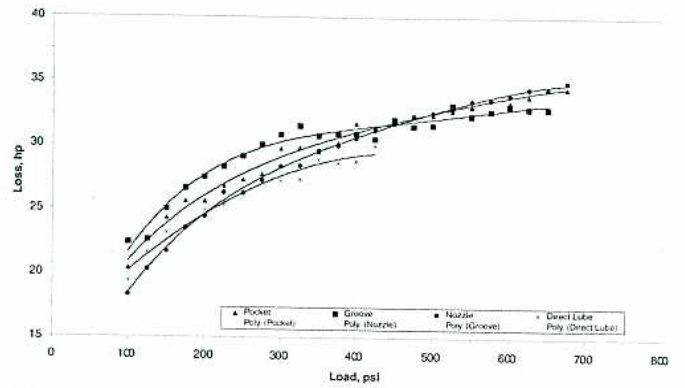


Figure 22. Loss Versus Load (6000 rpm - 10 gpm).

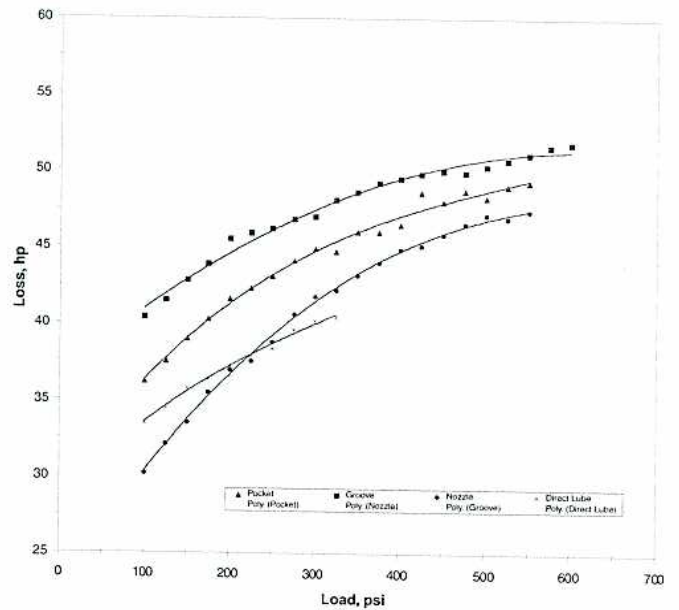


Figure 23. Loss Versus Load (8000 rpm - 10 gpm).

gpm. At velocities below 12,000 ft/min, all the test bearings are operating with full flow. Above this point, the bearings are operating under starved flow conditions.

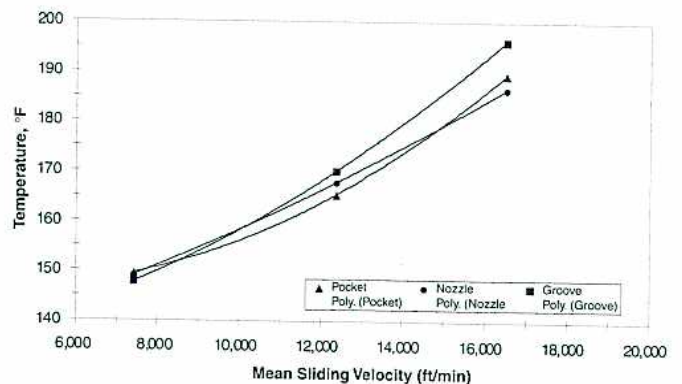


Figure 24. Maximum 75/75 Pad Temperature Versus Mean Sliding Velocity (100 psi Load - 10 gpm).

The temperature performance of the three bearings is very similar at sliding velocities below the starved film point of 12,370 ft/min (6000 rpm) with a loading of 100 psi. The groove bearing was the hottest configuration beyond this speed, with a maximum temperature difference of less than 10°F. At 300 psi load, the temperatures of the test bearings converged at 12,370 ft/min.

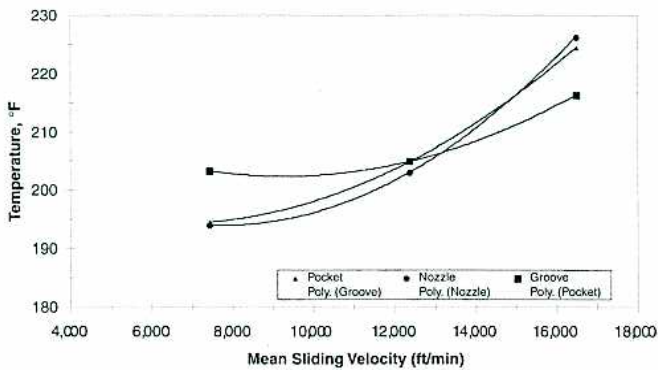


Figure 25. Maximum 75/75 Pad Temperature Versus Mean Sliding Velocity (300 psi Load – 10 gpm).

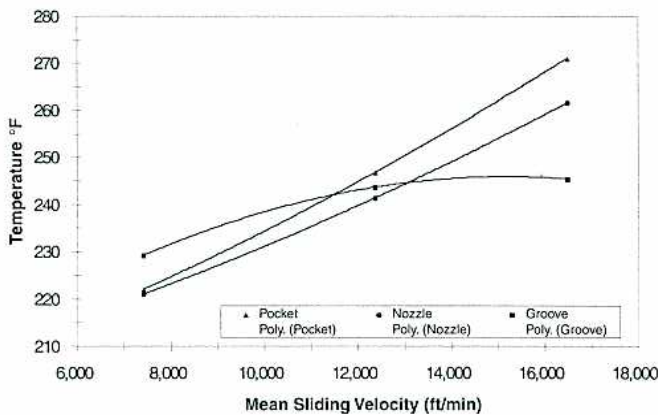


Figure 26. Maximum 75/75 Pad Temperature Versus Mean Sliding Velocity (500 psi Load – 10 gpm).

groove bearing ran the hottest below this velocity with a maximum temperature difference of 10°F. In the starved flow region, the temperature differential increased to 19°F with the pocket configuration recording the highest temperature. The largest temperature differential of 26°F occurred at a load of 500 psi and a sliding velocity of 16,493 ft/min (8000 rpm).

The horsepower loss as a function of sliding velocity for various loads is presented in Figures 27 through 29. The largest differences in losses (10.2 hp) between the three bearings occurred at the lowest load and highest velocity (Figure 27). At this loading, the pocket bearing exhibited the coolest temperatures and lowest losses. The maximum difference in horsepower loss between the test bearings reduced as the load was increased under starved flow conditions. At 300 psi, the separation was 5.2 hp, while at 500 psi the difference was 3.2 hp. The nozzle pocket inlet recorded the lowest loss of the three bearings under all starved flow load conditions.

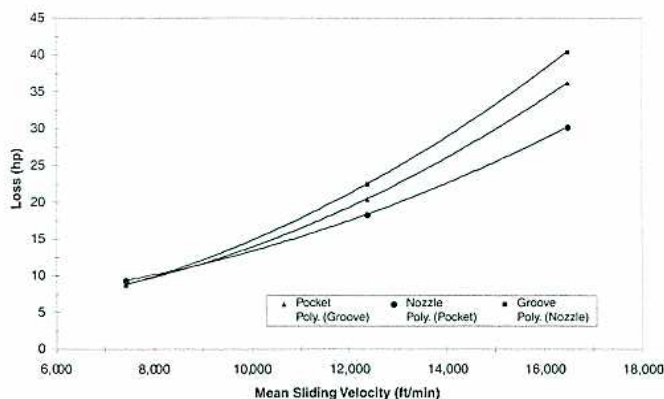


Figure 27. Loss Versus Mean Sliding Velocity (100 psi – 10 gpm).

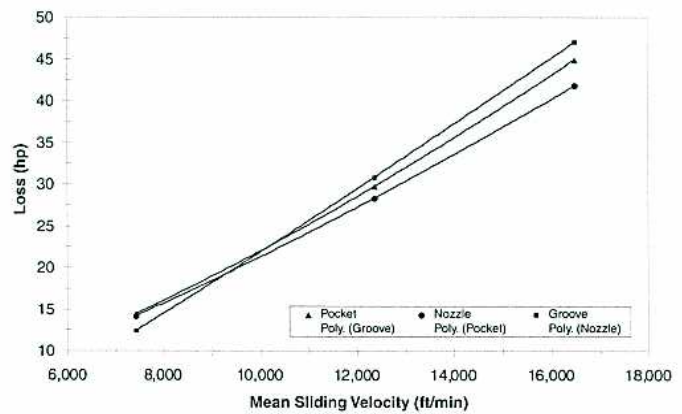


Figure 28. Loss Versus Mean Sliding Velocity (300 psi – 10 gpm).

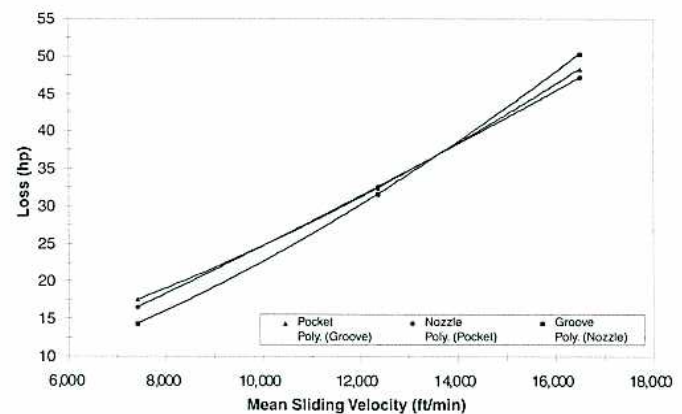


Figure 29. Loss Versus Mean Sliding Velocity (500 psi – 10 gpm).

A relative ranking of three of the oil inlet methods is shown in Table 1, based on speed, load, and level of oil supply. The rankings are based on the data presented in Figures 24 through 29 for both temperature and horsepower loss. The following classifications are presented with a caution that there is an inherent danger in applying such general terms to operating conditions:

- Medium speed: 3000 to 12,000 ft/min
- High speed: above 12,000 ft/min
- Light load: 100 psi or less
- Medium load: 100 psi to 400 psi
- High load: above 400 psi

The table is not intended to be a selection guide, but to show the relative performance of each oil inlet under identical service conditions. *Operation of thrust bearings under the test conditions is not recommended for field equipment.* The table should be used in conjunction with the experimental data since the separation margins were minimal in many instances.

CONCLUSIONS

Only a relatively small quantity of the total oil flow to a thrust bearing is required for lubrication. Additional flow must be supplied to carry away the heat associated with the work the bearing is performing for the rotating system. When the flow to a bearing is reduced, horsepower losses will decrease. Below a threshold flow point, the temperature of the babbitted surface will increase rapidly with small reductions in oil flow rate. At elevated operating temperatures, the bearings' capacity to absorb additional load is limited. There is a delicate balance between reductions in bearing losses through attenuated oil flows and overall machine reliability. Rotating equipment in the field behaves differently than

Table 1. Relative Performance of Oil Injection Methods under Various Load, Speed, and Flow Conditions.

Operating Conditions	Inlet Configuration	Temperature	Loss
Medium Speed Light Load Full Flow	Pocket	①	①
	Nozzle	①	①
	Groove	①	①
High Speed Light Load Starved Flow	Pocket	②	②
	Nozzle	①	①
	Groove	③	③
Medium Speed Medium Load Full Flow	Pocket	①	②
	Nozzle	①	②
	Groove	③	①
High Speed Medium Load Starved Flow	Pocket	②	②
	Nozzle	③	①
	Groove	①	③
Medium Speed High Load Full Flow	Pocket	①	③
	Nozzle	①	②
	Groove	③	①
High Speed High Load Starved Flow	Pocket	③	②
	Nozzle	②	①
	Groove	①	③

in the controlled environment of the test stand. If a bearing is designed to operate close to the starved film point, normal oil flow fluctuations in the field could lead to large excursions in bearing operating temperatures.

Reductions in horsepower loss and oil flow requirements are dramatic when comparing a flooded bearing design with one operating in an evacuated cavity. Once these gains have been realized, further reductions are somewhat more elusive.

This testing was designed to isolate the relative effectiveness of several oil inlet configurations on thrust bearing performance at low flow conditions. Four configurations were evaluated, each with identical geometry under identical operating conditions. The directed lubrication bearing operated poorly, primarily due to compromises in the design to support interchangeability of the base ring with two of the other configurations. Additional testing will be done in the future with a more conventional configuration to confirm this conclusion.

Based on the results of this investigation, the following conclusions may be summarized as follows:

- The difference in the starved flow point for the pocket, nozzle, and groove bearing were minimal at the load and speeds evaluated.
- The performance of the pocket, nozzle, and groove bearing were nearly identical at a mean sliding velocity of 12,000 ft/min and 10 gpm.

- Under starved flow conditions, reductions in the metal operating temperatures of the groove bearing over the pocket and nozzle arrangements were observed for sliding velocities above 12,000 ft/min and loads greater than 300 psi.

- The directed lube arrangement performed poorly in all tests conducted. This was due in part to compromises in this configuration to facilitate the reuse of the retainer.

- The pocket arrangement used the same oil injection configuration as the directed lube without a degradation in performance, indicating that this configuration offers an advantage in channeling of the oil to the leading edge of the pad.

- As load and speed increase under starved flow, the difference in horsepower loss between the pocket, nozzle, and groove bearing were minimal.

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