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Experimental Comparison of Flooded, Directed, and Inlet Orifice Type of Lubrication for a Tilting Pad Thrust Bearing

Introduction

At high speeds the power absorbed by a tilting pad thrust bearing as a result of churning the oil within the bearing casing is greater than that due to shearing the oil beneath the pads. Various methods have been used to try to eliminate, or at least significantly reduce the parasitic churning losses. This problem has been receiving attention for the last 10 years and in Europe a successful method of directed lubrication has been developed, whereas in the United States it is becoming common to reduce the churning losses by throttling the oil supply to the casing, thus reducing the amount of oil contained in the casing and consequently reduce the power loss due to churning.

This paper compares the results of tests carried out in order to assess the advantages of both throttled and directed lubrication systems against the more basic flooded system which has a pressurized casing.

The effectiveness of the two methods in reducing the parasitic churning losses and their effect on the bearing operating parameters and consequential safety margins are discussed.

Operation of Tilting Pad Thrust Bearings at High Speed

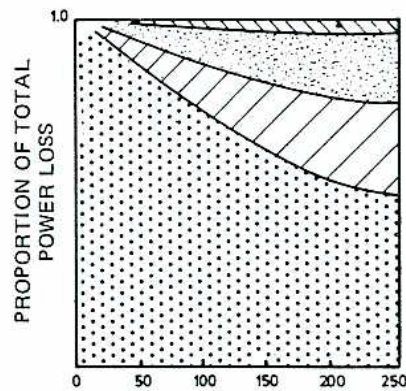
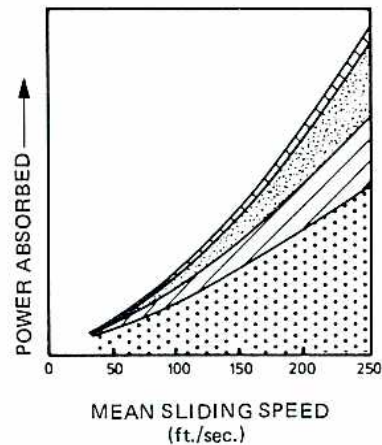
To illustrate the magnitude of the power absorbed by high speed tilting pad thrust bearings and the savings that can be achieved, one may consider a thrust bearing of a turbine generating set with a 30 in. dia collar running at 3000 rpm. Such a bearing may well absorb more than 700 hp of which only about 300 hp is as a result of the inherent shearing of the hydrodynamic oil film between the collar and pads. The remaining 400 hp is due to churning the oil within the casing and does not contribute to the safe operation of the bearings. The reduction of this power loss would naturally result in a commercially more acceptable bearing provided the margin of safety was not impaired. Not only will the reduced power loss increase the efficiency of the machine, especially at part load, but there will also be a reduction in initial plant cost because the cooling plant required to dissipate the heat generated in the bearing will be significantly smaller.

At low speeds the churning loss is not significant when compared with the total power loss. However, it rapidly increases with speed, as illustrated in Fig. 1 and may represent 60 percent of the total power requirement at high speeds (250-300 ft/sec). The contribution that the various parts of the bearing make to the total power absorbed is more fully covered in previous papers [(1 and 2)]¹ and will not therefore be treated here in depth.

To reduce the parasitic churning losses it is necessary to reduce the amount of oil in contact with the collar, but at the same time in no way impair the oil supply to each pad inlet edge.

Throttled Inlet

With this method, power loss is reduced by restricting the oil supply to the bearing. The oil is supplied at reduced pressure to the inner periphery of the bearing and allowed to flow between the pads to the outer periphery of the collar. A relatively unrestricted outlet is provided, preventing a complete filling of the casing. The oil supply is fairly critical in that under-supplying will result in insufficient oil for the bearing to operate satisfactorily, whereas over-supplying will result in complete flooding and little gain in power reduction.



- Churning losses at pad i/dia. & on shaft.
- Churning losses at rim of collar.
- Churning losses between pads.
- Shear losses on surge & main face pads.

¹ Numbers in brackets designate Reference at end of paper.

Directed Lubrication

With a directed lubrication system the oil is supplied to the inlet of each pad through suitable nozzles mounted between the pads. As a consequence of supplying all the oil to the area where it is required it should not be critical over a wide range of flows.

In high speed applications it is temperature rather than film thickness that is the limiting criterion. The pad temperature is greatly influenced by the temperature of the oil entering the pad, which in turn depends upon the proportion of that oil that has been carried over on the collar as hot oil from the previous pad. The mean temperature of the oil entering the pad can therefore be considerably higher than the oil supplied to the bearing assembly.

One method of reducing the hot oil carry over is illustrated in Fig. 2. A scraper is used to remove the hot oil from the collar before fresh oil is supplied at the leading edge of the next pad. The scraper must be placed close enough to the collar to remove this thin layer of oil. There are two major drawbacks to this design. Firstly, either the oil supply nozzles and scrapers have

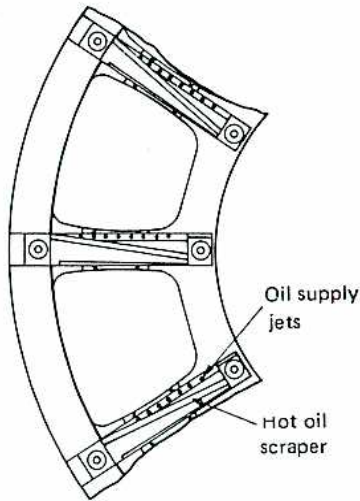


Fig. 2 Early design of directed lubrication using babbit lined scrapers to remove hot oil carry over

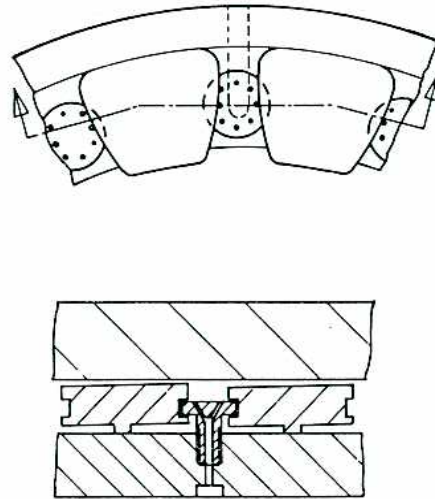


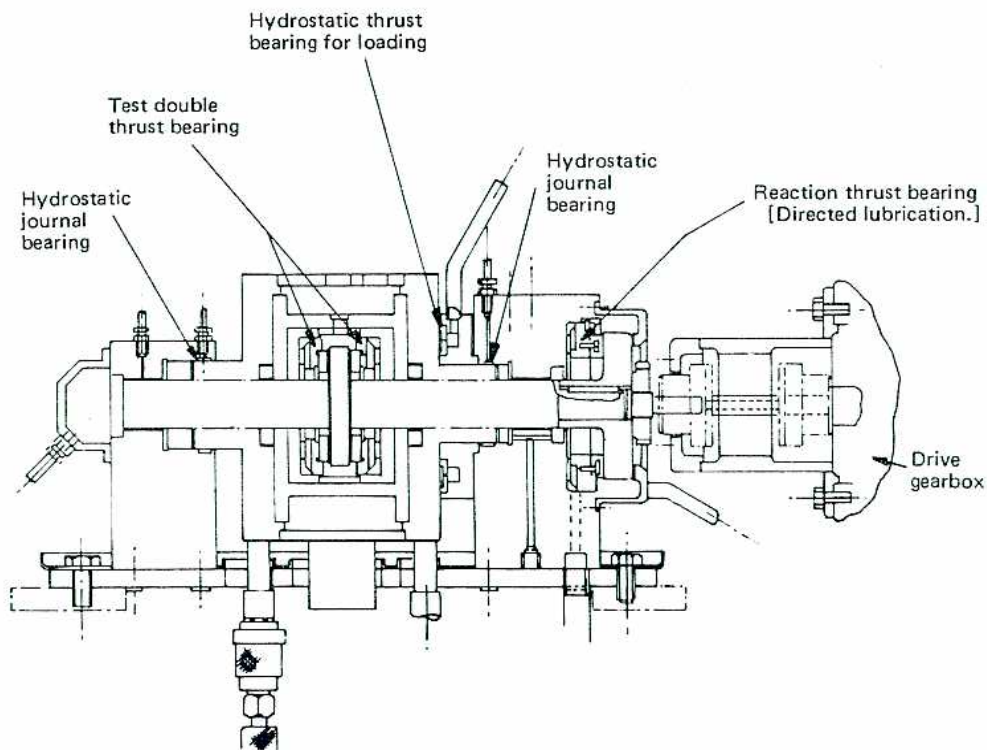
Fig. 3 Directed lubrication system using a conical array of oil jets

to be set up extremely accurately giving no allowance for wear or a spring loaded scraper system is required, resulting in an extremely complicated oil feed arrangement. Secondly, the scrapers themselves contribute to the power absorbed by the bearing. A further disadvantage is that the system can only be used for one direction of rotation.

To avoid these problems a system [3] of directed lubrication has oil jets which impinge on the trailing edge of the preceding pad to help to reduce local high temperatures, also scour hot oil from the collar as well as spraying oil to the inlet edge of the pad. The nozzles take the form of "umbrella" sprays fitted between the pads in place of the normal pad stops, which produce a controlled conical array of oil jets, see Fig. 3. For very small pads (below 1 in.) it has been shown that a single oil jet between the pads will operate successfully.

Test Rig and Instrumentation

A standard double acting thrust assembly [4] having a 5 in. dia collar was used for this series of tests mounted in the "Orion"



test rig. Each bearing had 8 white metal faced center pivoted steel pads, each pad being 1.125 in. wide.

The Orion test rig is shown in Fig. 4. The test shaft may be driven at speeds up to 12,000 rpm by a 75 hp variable speed motor through a gearbox. To enable the torque to be accurately measured the test head is mounted on hydrostatic bearings, and the load is applied through a hydrostatic pad. Torque is measured by a spring balance, speed is displayed by an electronic frequency meter, load by a pressure gauge in the hydraulic loading circuit, and oil flow by a fixed displacement meter.

The instrumentation in the main face pads is shown in Fig. 5. Two of the pads are fitted with capacitive film thickness transducers and a further two pads are fitted with 9 copper-constantan thermocouples fitted in the white metal surface. The pads with film thickness transducers were placed opposite each other and a thermocoupled pad was placed next to each of them. The tests were carried out using a mineral oil with a viscosity of 41cSt at 50 deg C (and 84cSt at 100 deg C). The oil inlet temperature was maintained at 50 deg C throughout the tests. The lubrication arrangement for the three systems tested are shown diagrammatically in Fig. 6.

Comparison of Power Loss

Fig. 7 shows the effect of oil flow on the power absorbed by the bearing with various specific loads for the three systems; pressurized flooded, throttled, and directed lubrication at a speed of 12000 rpm. The curves show that the bearing with the pressurized casing absorbs the greatest power and the directed lubricated

bearing the least. In the case of pressurized flooded and directed lubrication the oil flow rate has a small effect on the power absorbed, whereas in the bearing operating with a throttled supply power loss is dependent upon the oil flow rate through the bearing. At very low flows (below those that would normally be recommended for safe operation of the bearing) the saving in power with throttled inlet was approximately 30 percent whereas

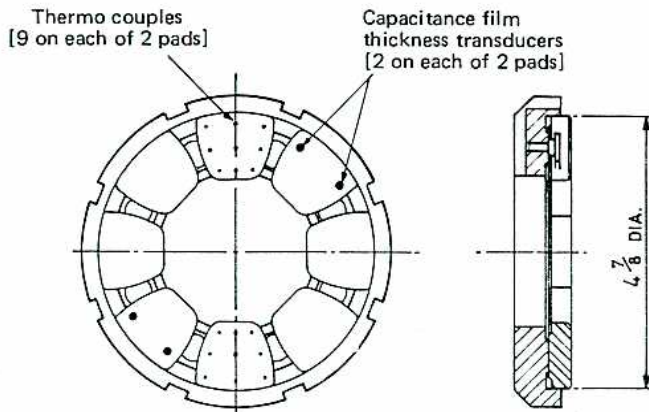


Fig. 5 Layout of temperature and film thickness measurement instrumentation

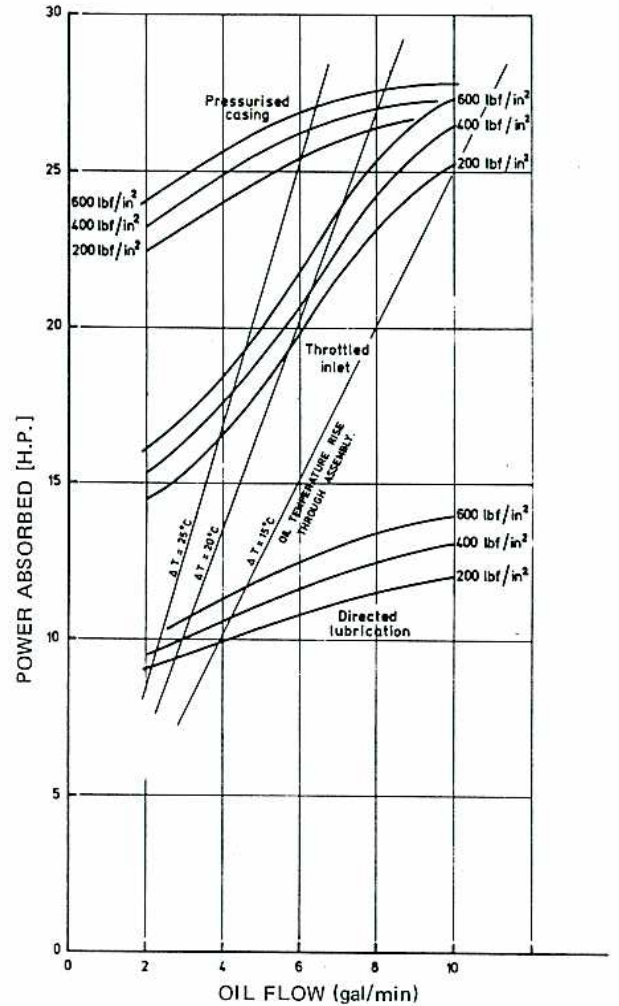


Fig. 7 Relationship between oil flow and power absorbed by the bearing at 12,000 rev/min for the lubrication systems at different bearing specific loads

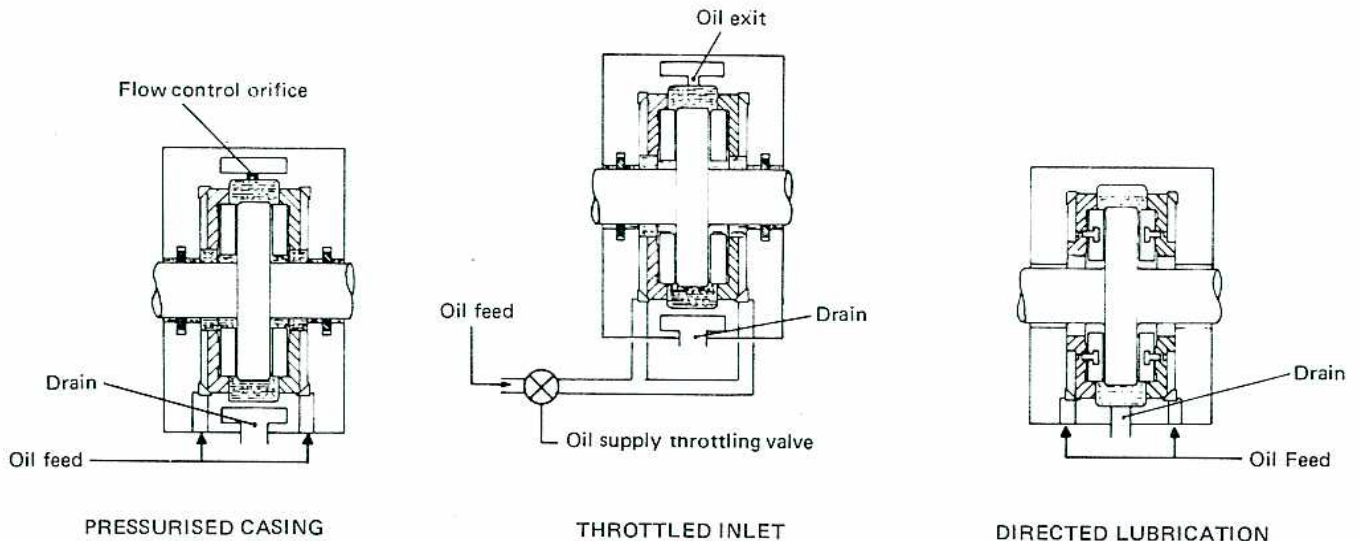


Fig. 6 Diagrammatic layout of the three lubrication systems tested

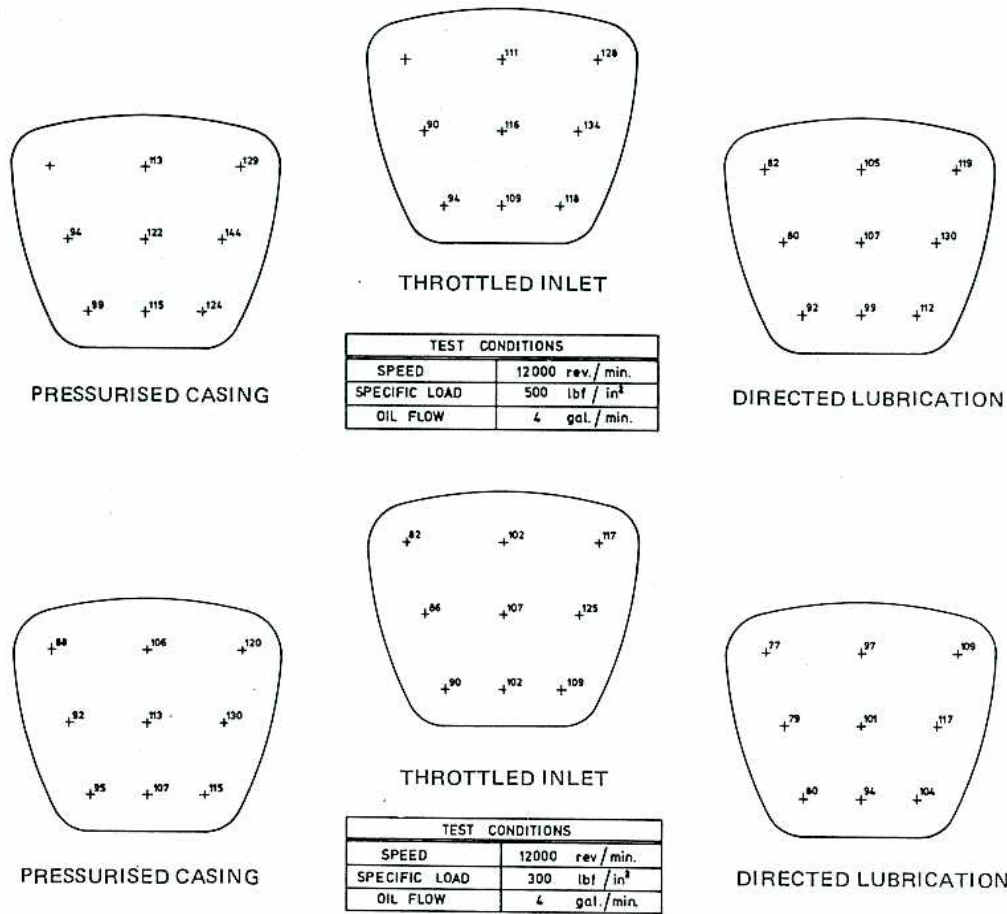


Fig. 9 Typical measured pad surface temperatures

with directed lubrication it was approximately 55 percent. However, as the oil flow increased the advantage of throttling decreased until, at very high flows, the power loss almost reached that of the bearing with the pressurized casing. In contrast to this the power loss of the directed lubricated bearing increased in a similar way to that of the pressurized flooded bearing; and at these high flows the power saving was still of the order of 50 percent.

Comparison of Pad Temperatures

Figs. 8 and 9 show typical pad surface temperature results at 12,000 rpm and specific loads of 300 and 500 lbf/in.². The oil flow was approximately 4 imperial gallon/minute. From a comparison of the thermocouple readings of pad surface temperatures it can be seen that the temperatures at the pad inlet edge are slightly lower for the throttled supply than for the pressurized flooded case. This is to be expected since the churning losses have been reduced; consequently, for the same oil flow the oil temperature rise across the bearing and thus the oil inlet temperature to the pad has been reduced. The pad inlet temperature under directed lubrication was significantly lower than either of the other two cases and approached the oil supply temperature. This demonstrates the effectiveness of the directed lubrication system in reducing the amount of hot oil carry over.

The maximum temperature of the pads operating under identical conditions of load, speed, and oil flow will be mainly controlled by the pad oil inlet temperature. This is confirmed by these tests, however, there is a noticeable difference in the radial temperature distribution between the throttled case and the pressurized case. The former shows a much greater radial change than the latter. This is presumably because there is little mixing of the supply oil as it enters the bearing but more mixing takes place at the periphery with both recirculating oil and with the hot oil carried on the collar from the preceding pad.

The effect of increasing pad specific load on the maximum pad

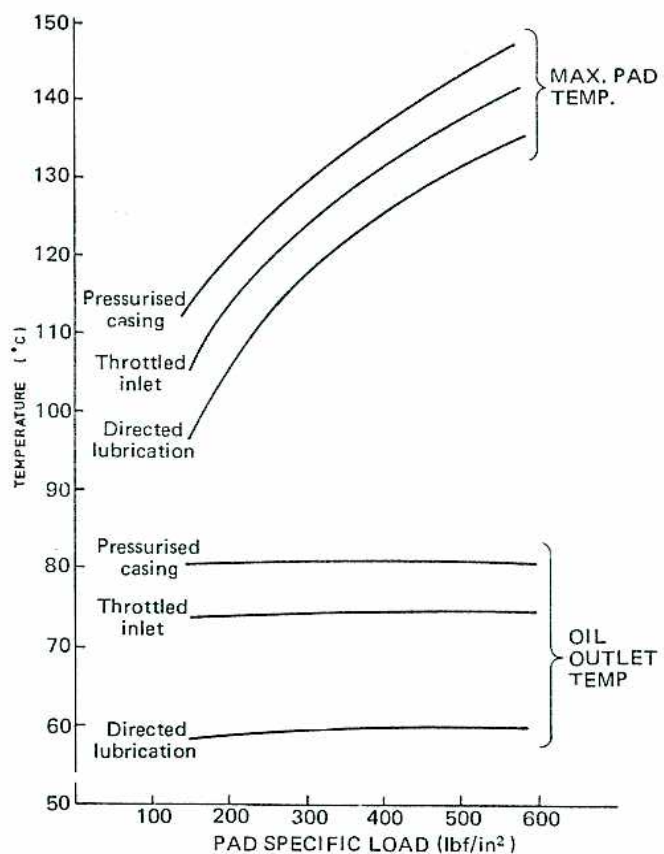


Fig. 10 Variation in maximum pad temperature and oil outlet temperature at 12,000 rev/min and 4 gal/min oil flow

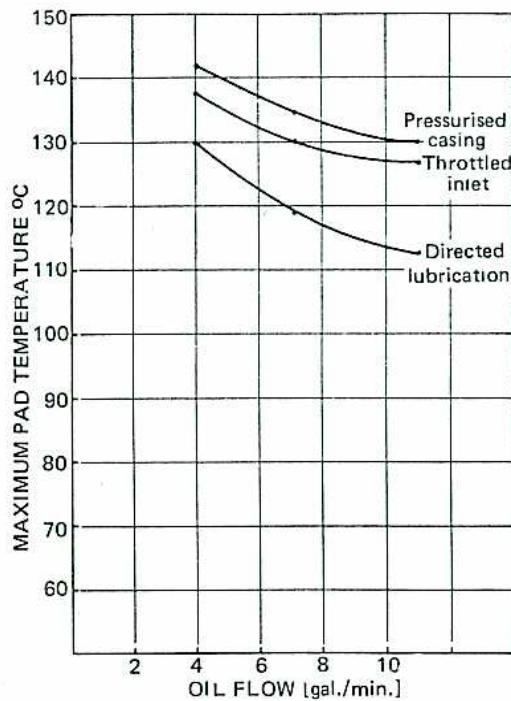


Fig. 11 Variation in maximum pad temperature with oil flow at 12,000 rev/min and 500 lbf/in² pad specific load

surface temperature is given in Fig. 10 and shows that the trends already discussed above are true for the range of pad specific loadings of 150 lbf/in.² to 600 lbf/in.². Fig. 11 shows the effect of oil flow on maximum pad temperatures for the three systems of lubrication. The reduction in pad temperature with increased oil flow is most effective with directed lubrication. This results from the higher jet velocities which are more efficient in reducing the hot oil carry over.

Comparison of Oil Film Thickness

Fig. 12 shows that there is little difference in the oil film thicknesses for all three methods of lubrication. The differences that do occur are of the order that would be expected from measured differences in oil inlet temperature to the pads.

Bearing Safety

Tests have been carried out in the USSR [5] to investigate bearing safety with both a throttled supply and pressurized casing. In each case the bearing was run with the lubrication system operating and load increased until failure occurred. The throttled bearing was only capable of carrying 84 percent of that of the pressurized casing bearing. It was concluded that the most likely reason for this was that the oil entering the pads of the throttled bearing was partly foamed due to the subatmospheric pressures that exist in the casing with this type of lubrication. (Depressions had been measured as low as half an atmosphere within the casing.)

Although no specific figures are available, tests have shown that the load capacity of a directed lubricated bearing at high speed is higher than that of one with the pressurized casing. This is generally as a result of such a unit running cooler.

Preference is often expressed in favor of the pressurized flooded bearing on the grounds of safety. However, such thrust bearings usually operate in conjunction with journal bearings. Therefore there is no advantage in having a thrust bearing capable of extended operating after the journal bearings have seized solid due to an oil supply failure. Tests have shown that the directed lubricated bearing is capable of operation after the oil supply

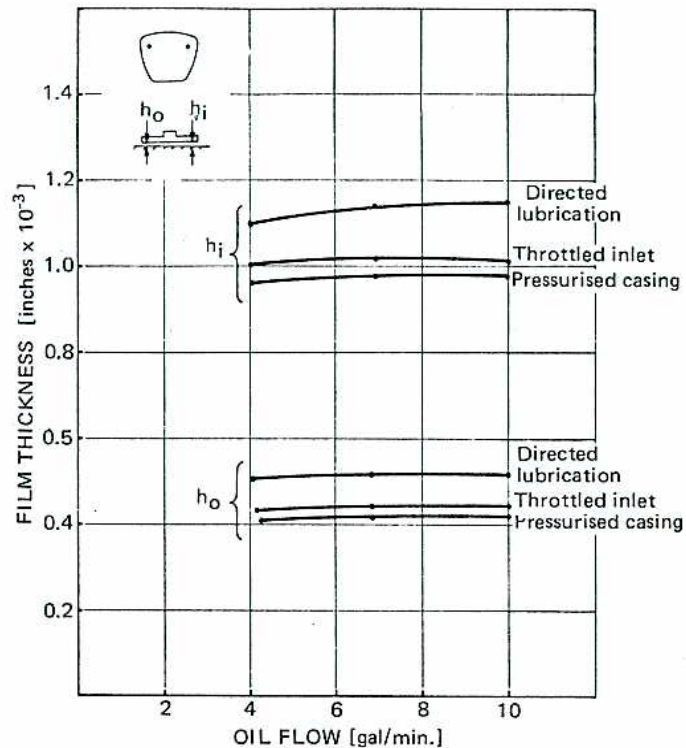


Fig. 12 Relationship between oil flow and film thickness at 12,000 rev/min for the three lubrication systems tested

A bearing with throttled inlet similar to the one tested will retain some oil in the casing if the oil supply fails. This being so there is little doubt that such a bearing will behave in this respect at least as well as the one with directed lubrication. What is probably more important is the ability of the bearing to cope with the momentary overload or extended operation with high pad surface temperatures. If the bearing operating temperature be used as a criterion (which is usually the limiting factor at high speed) then one would expect that the bearing with the lowest temperature would be the safest bearing in operation. This view is well supported by industrial experience.

Further, it is often possible to use a smaller bearing to carry the load with directed lubrication than would be the case if either of the two other systems are adopted. This in turn will further reduce the power consumption of the unit.

Field experience resulting from the increasing use of both lubrication methods by industry indicate that these limited tests are representative of the characteristics of larger units.

Industrial plant development is creating a need for bearings of high capacity to run at greater speeds where churning would be the major source of power loss if a flooded bearing with pressurized casing were used. It is important therefore, that the performance of assemblies with these different lubrication systems is known since it is expected that there will be an increased demand for them.

References

- 1 Martin, F. A., "Tilting Pad Thrust Bearings: Rapid Design Aids," The Institution of Mechanical Engineers, Tribology Convention, 1970.
- 2 Bielec, M. K., and Leopard, A. J., "Tilting Pad Thrust Bearings: Factors Affecting Performance and Improvements With Directed Lubrication," The Institution of Mechanical Engineers, Tribology Convention, Vol. 184, Part 3L, 1970.
- 3 Features of the directed lubrication described in this paper are covered by U. S. Patent Nos. 3378319, 3454312, 3494680, and corresponding foreign patents and applications.
- 4 Glacier Standard tilting pad bearings assembly part no. 8112/2P.
- 5 Zaretskii, E. I., et al. *Energomashinostroenie*, Vol. 16, No. 7,

DISCUSSION

A. B. Harbage²

Mr. New discusses two methods of reducing friction in tilting pad thrust bearings. The system used by Glacier Metal Company of England which sprays oil at the pad leading edges in an otherwise empty bearing cavity, and the common method used in the United States of throttling the supply reducing the oil volume in the bearing cavity. Not covered by Mr. New is a third method for providing oil to the pads in an otherwise empty bearing cavity. This method is fully discussed in U. S. Patent 3,512,854 of May 19, 1970. In this scheme the thrust shoes are attached to the rotating element. The gaps between the shoes thus form channels not unlike those of a radial centrifugal pump. By shaping the cross section of these channels such that the distance between pads is greater at the pad runner side than pad back side, and incorporating a restraining dam at the channel OD, a moving pocket of oil exists all along the interface of the moving pad leading edge and runner. A rotating ring distributes the oil to the channel ID. The oil within the pocket at the interface operates in a force field having a large component in both the radial and tangential direction, providing a high positive head exactly where needed.

This method has been applied to a very large bearing in U. S.

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Navy use. A transparent working model showed that the bearing cavity became entirely empty except for a film covering the runner and the moving pockets at the shoe leading edges. The bearing has performed satisfactorily for five years. No laboratory evaluation or analysis has been made to determine film thickness or frictional loss, but due to the observed lack of oil at anywhere within the bearing housing except at the shoes leading edges and the film on the collar, reduced frictional losses must exist.³

Authors' Closure

The contribution by Mr. Harbage shows an interesting solution to the problems of reacting thrust between two counter-rotating shafts.

I would, however, imagine that such a design was limited to low and moderate speed. At high speeds where churning losses contribute significantly to the power loss, there may be a hazard resulting from attaching pads to a collar which itself may be approaching its bursting speed. Further at high speeds the energy required to provide the high positive head in the oil pockets will itself form an additional loss.

Nevertheless, the design no doubt forms an adequate and economic engineering solution to the problem for which it was designed.

³ The opinions expressed herein are those of the discussor and do not necessarily reflect the views of the Navy Department.

May 19, 1970

A. B. HARBAGE, JR.
LOW FRICTION THRUST BEARING
Filed Dec. 27, 1967

3,512,854

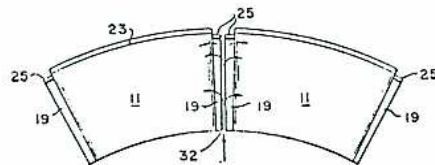
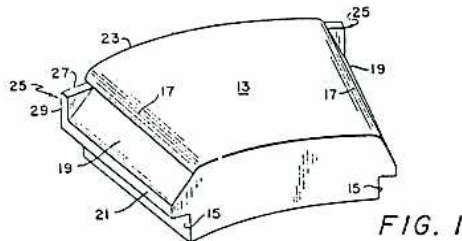


FIG. 2.

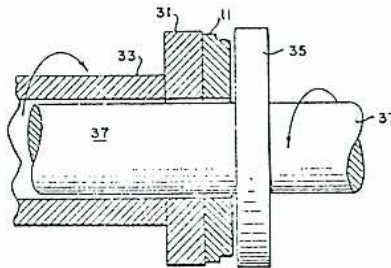


FIG. 3.

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