Performance evaluation of the LEG tilting pad journal bearing

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SYNOPSIS This paper discusses the performance characteristics of the leading-edge-groove (LEG) tilting pad journal bearing and presents new experimental temperature data from a bearing operating with "on pad" and "between pads" loading conditions, and with different bearing clearances. This data is then compared with results obtained from a computer model of the LEG bearing. Good correlation between the theoretical and experimental results suggests that the technique used in the model to calculate the temperature of the oil at the leading edge of the LEG journal pad is reasonably accurate.

The experimental data is collected from a 0.098 m diameter, five pad bearing, operating at shaft speeds up to 16500 rev/min and with unit loads up to 3 MN/m^2. The paper also presents experimental results from bidirectional testing of the offset pivot LEG bearing, and finally discusses a technique for achieving substantial reductions in the power loss of the LEG bearing.

1 INTRODUCTION

The circular bore journal bearing may experience self excited subsynchronous vibration during operation, and although other bore modifications such as the multi-lobe and offset halves designs are successful at raising the stability threshold, only the tilting pad journal bearing offers the possibility for eliminating oil film instability. These unique characteristics were confirmed by Brockwell *et al.* (**1**) in a series of experiments on a conventional tilting pad bearing. They found that the cross-coupled coefficients were negligible in comparison to the direct coefficients, providing that there was geometrical and "thermal" symmetry of the bearing. Ettles (**2**,**3**) considered the effect of lack of symmetry on the performance of the tilting pad journal bearing.

Because the tilting pad journal bearing is finding increased usage in high power density machinery, interest is focusing on the bearing's steady state performance. For example, Brockwell and Dmochowski (**4**) found that, during operation, significant changes to the bearing and pad clearances may lead to a pad preload that is substantially different to that specified in the initial design.

In recent years, a new design of hydrodynamic bearing, known as the leading-edge-groove (LEG) tilting pad bearing, has been the subject of further development work. The LEG bearing is so named because the leading edge of each pad is extended to accommodate an axial oil distribution groove that directs a controlled amount of cool lubricant into the hydrodynamic oil film. An earlier experimental study (**5**) focused on a thrust version of the LEG bearing, when tests on a 267 mm outside diameter bearing at speeds up to 13000 rev/min indicated significant reductions in frictional loss and bearing operating temperature. More recently, a journal version of the LEG tilting pad bearing was the subject of preliminary experimental and theoretical studies (**6**). This work showed that the LEG bearing had significantly lower operating temperatures to those of the conventional bearing; a characteristic that was attributed to the reduction in hot oil carry over between one pad and the next.

This paper is an extension of the work described by Dmochowski *et al.* (**6**) and presents new experimental temperature data from the LEG journal bearing for both "on pad" and "between pads" loading conditions, and for different bearing clearances. The experimental results are compared with data from a computer model of the LEG bearing. Furthermore, the paper discusses results obtained from bidirectional testing of the LEG bearing, and describes a technique for reducing the power loss of the LEG bearing.

2 EXPERIMENTAL APPARATUS

2.1 Description of test rig

The test facility is described in detail elsewhere (**7**). Briefly, the 0.098 m diameter shaft is supported on high precision, angular contact ball bearings, and driven by a variable speed electric motor through a belt-pulley system to give a range of speeds between 1800 and 16500 rev/min. The test bearing is mounted in a special housing located midway between the support bearings and moves in response to a vertical load that is applied to the housing by a tensioned cable. Eddy current probes mounted on the ends of the housing measure the horizontal and vertical displacements of the bearing and provide a check of the alignment of the bearing with respect to the shaft. Three axial tensioned wires attached to each end of the housing minimize non-parallel movement of the bearing with

Table 1 Bearing and pad clearances

Bearing group	Bearing radial clearance	Pad radial clearance	Preload
	mm	mm	
1	0.076	0.102	0.25
2	0.102	0.102	0.00
3	0.051	0.102	0.50

respect to the shaft. Because these wires have lateral flexibility, their influence on the radial movement of the bearing housing is small.

The lubricant used in this study was a light turbine oil (ISO VG32) with a viscosity of 0.02325 Pa.s at 40C and 0.0054 Pa.s at 100C. The lubrication system incorporated a feedback control device and a shell and tube heat exchanger to regulate the oil supply temperature to the bearing. The inlet temperature was maintained between 48C and 50C.

Bearing circumferential temperature distributions were measured by copper-constantan thermocouples imbedded in the babbitt lining to within 0.5 mm of the bearing surface (Figure 1 shows the location of these thermocouples). These measurements, together with all other data from the rig, were stored in the memory of a data acquisition system and passed to a host computer at a later time.

2.2 Description of test bearings

The bearings have a nominal diameter of 0.098 m, a length /diameter ratio of 0.387 and consist of five babbitt lined journal pads supported in a precisely machined aligning ring. This ring is split axially to allow easy assembly of the bearing around the shaft. An annular oil distribution groove is machined into the outside of the aligning ring and, in the case of the conventional bearing, oil is directed from this annulus through radial feed holes into the spaces between adjacent pads. Cool oil to the pads of the LEG bearing is directed to the centre of the leading edge grooves by small diameter pipes fitted with O rings (see Figure 2). The pads of both bearings have an effective angle of 56.1 degrees, although the overall angle of the LEG pad is somewhat larger to accommodate the leading edge groove. The axial length of this groove is 33.5 mm. To allow the pads to adjust to conditions of axial misalignment, their back surface is contoured both circumferentially and axially. The shoes are held axially and circumferentially by retaining plates. Details of bearing and pad clearances, and resulting pad preload values, are given in Table 1.

Each group comprised a 0.6 offset pivot LEG bearing. In addition, group 1 included a 0.6 offset pivot conventional bearing.

2.3 Bearing test conditions

Tests were conducted with loads of 5.18 kN and 11.1 kN, shaft speeds that ranged between 1800 rev/min and 16500 rev/min and nominal flow rates that depended on the actual test condition (see Table 2). The group 2 LEG bearing was tested for both "on pad" (LOP) and "between pads" (LBP) loading configurations. Tests were also performed on bearing groups 1, 2 and 3 to determine the effect, on bearing temperature, of pad radial clearance. Finally, the group 1 LEG bearing was run in reverse rotation.

3 BASIS OF THE COMPUTER MODELS

Computer models that calculate the performance of the conventional and LEG tilting pad journal bearings have been developed and are described elsewhere (**6**). Briefly, the pressure distribution in the oil film is calculated from a two-dimensional version of the Reynolds' equation that considers viscosity variations in the circumferential and radial directions, and assumes the Swift-Steiber boundary condition for a cavitated oil film. Turbulence is accounted for in a manner similar to that described by Constantinescu (**8**) and Frene and Constantinescu (**9**). The temperature distribution in the oil film is governed by the energy equation, accounting for heat conduction across the oil film and heat convection in the circumferential direction. The energy equation is solved assuming the following boundary conditions: 1) a shaft temperature that is equal to the average of the pad temperature distributions; 2) a certain temperature distribution at the pad surface (calculated from the heat transfer equation); and 3) a pad leading edge temperature that is constant through the thickness of the oil film. The third boundary condition is calculated on the basis that hot oil from the preceding pad affects the oil temperature at the leading edge of the next pad. This temperature is calculated from simple heat balance equations which represent pad leading edge conditions in both the conventional and LEG bearings. This matter is discussed later in the paper.

The temperature distribution at the pad surface is calculated from the Laplace heat transfer equation, assuming a constant temperature in the axial direction. The boundary conditions for the heat transfer equation correspond to an equality of heat fluxes at the boundary between the oil film and the pad surface, and heat

Table 2 Nominal oil flow rates, 10^{-4} m^3/s

Load (N)	Shaft speed (rpm)					
	1800	3600	5000	9000	12000	16500
1300	0.15	0.39	0.61	1.37	2.11	3.60
5175	0.21	0.52	0.76	1.56	2.32	3.81
10350	0.28	0.66	0.93	1.82	2.63	4.21
11100	0.21	0.65	0.95	1.85	2.67	4.27

convection on all remaining surfaces. Approximate pad deflection is calculated from the one-dimensional equation for a beam. As in (2), shear forces, bending moments and local differences in temperature across the thickness of the pad are taken into consideration. In the axial direction, the load distribution is assumed constant and is calculated by averaging the pressures in the axial direction. Boundary conditions at the pivot correspond to zero pad surface deflection and zero gradient.

By simultaneously solving these equations for each individual pad, the solution for the entire bearing is obtained in a manner similar to that originally proposed by Ettles (2).

4 PERFORMANCE EVALUATION OF THE LEG BEARING

4.1 Bearing operating temperatures

Dmochowski *et. al.* (6) discuss some preliminary tests of the group 1 LEG bearing and compare it's performance with that of the group 1 conventional tilting pad journal bearing. Both bearings had a pivot location of 0.6. These tests showed that, at shaft speeds above 12000 rev/min, the LEG bearing had substantially lower operating temperatures. For example, at 16500 rev/min with a load of 11.1 kN, the measured maximum temperature of the LEG bearing was 105C, compared to 127C in the case of the conventional bearing. When the performance characteristics of the bearings were analyzed using the computer models described in this paper, it was concluded that the lower operating temperatures of the LEG bearing were the result of reducing the effect of hot oil carry over. Furthermore, it was found that simple heat balance equations could be used when calculating pad leading edge oil temperatures. In the case of the conventional bearing, the authors assumed that all of the flow q_2 (at temperature t_2) leaving the pad trailing edge will enter the leading edge of the next adjacent pad (the limitation is that q_2 cannot exceed 90 per cent of q_1). Thus:

$$t_1=\frac{(q_1-q_2)t_i+q_2t_2}{q_1} \tag{1}$$

where:

t_1 = oil temperature at the leading edge
t_2 = oil temperature at the trailing edge of the preceding pad
t_i = oil inlet temperature to the bearing
q_1 = oil flow at the leading edge
q_2 = oil flow at the trailing edge of the preceding pad
q_i = oil flow to each pad (total flow/number of pads)

In the case of the LEG bearing, it was assumed that all cold oil supplied to the leading edge groove enters the film at the leading edge of the pad, with the balance of the flow being provided by the hot oil carried over from the previous pad. Hence:

$$t_1=\frac{q_it_i+(q_1-q_i)t_2}{q_1} \tag{2}$$

Equations (1) and (2), when used in conjunction with the computer models described in this paper, were found to give calculated bearing temperatures that were in good agreement with the measured temperatures (6).

The work of Dmochowski *et. al.* (6) also featured an investigation of the effect of oil flow rate on bearing temperatures. In the case of the conventional bearing, it was found that halving or doubling the flow rate had little effect on the operating temperature of the bearing, probably because there was always enough oil in the cavity of the bearing to provide the right amount of "make up" flow (q_1) at the leading edge of each pad, even when the oil flow rate was reduced by 50 per cent. Consequently, variations in pad operating temperature were small. On the other hand, the LEG bearing was more sensitive to oil flow, with the result that this bearing had a different operating characteristic from that of the conventional bearing. These tests showed that the operating temperature of the LEG bearing rose slightly when the oil flow was reduced by 50 per cent. Under such operating conditions, it would seem that less cool oil from the oil supply groove, and more hot oil from the previous pad, now enters the oil film, resulting in higher pad temperatures.

4.2 LOP vs. LBP bearing tests

This section of the investigation featured "LOP" and "LBP" tests on the group 2 LEG bearing with loads of 5.18 kN and 11.1 kN, and with shaft speeds that ranged between 1800 and 16500 rev/min. Figure 3 shows measured bearing temperatures from these tests. With regard to the results for the 5.18 kN load (Figure 3(a)), it is of interest to note that bearing temperatures are similar for both loading configurations with shaft speeds up to 16500 rev/min. This is also true of the results from the 11.1 kN load tests for shaft speeds up to approximately 12000 rev/min, although at higher speeds there are differences. For example, at 16500 rev/min, the measured difference was 8C, with the "LOP" operating at the higher maximum temperature of 116C. The results seem to suggest that , for less severe operating conditions, the bottom loaded pad(s) of the bearing operate at a similar temperature, regardless of whether the load is "on" or "between" pads. However, for more severe operating conditions when the load is "on pad", the increase in bearing temperature is thought to be the result of a thinner operating oil film.

Figure 3 also presents calculated results. These results are in fairly close agreement with the measured bearing temperatures and suggest that the assumptions made in the model (particularly the boundary conditions for the energy equation) are quite reasonable.

4.3 Effect of bearing clearance

Tests were conducted on bearing groups 1 to 3 inclusive to examine the effect of bearing clearance on the operating temperature of the LEG bearing. It should be noted that it was the "bearing" clearance that was changed for this series of tests. Figure 4 presents measured and calculated maximum bearing temperatures for the three bearing groups. The bearings were tested with both load "on pad" and "between pads", for a range of shaft speeds up to 16500 rev/min and a load of 11.1 kN.

Figure 4 has a number of interesting features. Firstly, as with the results presented in the previous section, differences in temperature between LOP and LBP were only noticeable at the higher shaft speeds. This was true of all three bearings. Secondly, reducing the "bearing" clearance from 0.102 mm to 0.076 mm had little effect on bearing operating temperatures, although when the clearance was further reduced to 0.051 mm, quite significant increases in temperature were observed. Since this increase in bearing temperature was particularly rapid at higher shaft speeds, tests on the group 3 bearing were restricted to a maximum speed of 12000 rev/min. Unfortunately, LOP tests on the group 3 bearing were not conducted.

4.4 Bidirectional tests

The leading-edge-groove tilting pad journal bearing features an offset pivot and a leading edge oil distribution groove that are intended to improve the performance of the bearing, but, it is believed, make it suitable for only one direction of rotation. Certain machines, however, require a bearing that is suitable for bi-directional operation. These machines usually operate in one direction for much of their operational life, with only short excursions in the reverse direction of rotation. The conventional centre pivoted pad is suitable for both directions of rotation and is often used in such applications. However, when compared to the performance characteristics of the offset pivoted bearing, the centre pivoted bearing is usually found to operate with a higher babbitt temperature and a thinner oil film.

The purpose of this experimental study was to show that the LEG tilting pad journal bearing is suited to bidirectional operation, and that it can operate in reverse rotation with the maximum rated load of the bearing without causing damage to the babbitt surfaces. Bidirectional testing of the LEG bearing therefore resulted in the pivot being located in the leading half of the pad at the 40 per cent position, and the oil supply groove being positioned at the trailing edge of the pad. Because the position of the thermocouples was unchanged from the "forward" rotation tests, it was thought that this arrangement would not be capable of detecting the maximum temperature of the bearing when operating in reverse. However, it should be possible to estimate the maximum temperature of the bearing and Figure 5 shows how this was done. Essentially, this involved estimating the pad temperature gradient in the direction of rotation from the three measured temperatures on the leading half of the pad and assuming that the maximum temperature will occur at about the 60 per cent location. This would seem to be a reasonable assumption given that the oil film does not extend to the end of the pad, but ends, probably, somewhere in the vicinity of the 60 per cent location.

Figure 6 shows maximum pad temperatures for both the forward and reverse directions of rotation, with a load of 11.1 kN and shaft speeds that ranged between 1800 and 16500 rev/min. As expected, bearing temperatures were higher when running in reverse rotation, although the difference was not that significant at speeds up to 6000 rev/min. At higher speeds the difference was more noticeable, and was approximately 10C at 12000 rev/min. Results from tests with a load of 5.18 kN, while not shown on Figure 6, indicated a maximum temperature of only 95 C at 12000 rev/min, suggesting that higher speeds at lower loads should be possible.

At the conclusion of the reverse rotation tests, the bearing was found to be in a sound condition and suitable for further operation.

Table 3 Calculated power losses and maximum pad temperatures. 11.1 kN load "between pads". Group 1 bearings. Shaft speed = 16500 rpm.

Bearing	Power loss kW						Max. pad temp. °C
	pad 1	pad 2	pad 3	pad 4	pad 5	total	
conventional	3.48	4.48	1.88	1.55	1.85	13.24	124
LEG	3.36	4.03	2.39	1.72	1.87	13.37	107
LEG (T_i=60°C)	2.90	3.33	1.61	1.16	1.39	10.39	111
LEG (T_i=74°C)	2.36	2.53	1.05	1.19	0.97	8.1	121

5 REDUCING THE POWER LOSS OF THE LEG BEARING

The authors were unable to measure accurately the friction torque of the test bearings, but instead used their computer models to analyze the power loss of the bearings. The purpose of this work was twofold: 1) to identify a method for reducing the power loss of the LEG bearing; and 2) to draw a comparison between the losses of the LEG and conventional bearings. Figure 7a shows that the calculated losses of the LEG and conventional bearings, for an oil inlet temperature of 49C, are similar. However, the maximum operating temperature of the LEG bearing is significantly lower (107C compared to 124C for the conventional bearing). Therefore, it should be possible to raise the oil inlet temperature of the LEG bearing so as to reduce it's power loss, and maintain a maximum bearing temperature that is equal to, or lower than, that of the conventional bearing. As an extreme example, Figure 7a shows calculated power losses for the group 1 LEG bearing where the inlet temperature is raised to the point where the maximum temperature of the LEG and conventional bearings are similar. (Obviously, the rise in inlet temperature is dependent on bearing operating conditions, and details on other shaft speeds are given in Figure 7b). At 16500 rev/min, with a load of 11.1 kN, the oil feed temperature to the LEG bearing must be raised to 74C to give a maximum temperature similar to that of the conventional bearing (with an inlet temperature of 49C). Figure 7a shows that substantial power savings are possible, particularly at higher shaft speeds; for example, with an inlet temperature of 74C the savings are of the order of 40 per cent at 16500 rev/min. Table 3 shows the losses of the individual pads of the conventional and LEG bearings for an inlet temperature of 49C, as well as those of the LEG bearing with elevated inlet temperatures of 60C and 74C. It is worth noting that the minimum film thickness is still acceptable with a 74C inlet temperature (i.e. $1.8*10^{-5}$ m compared to $2.4*10^{-5}$ m with a 49C inlet temperature).

Table 3 also lists power losses for a 60C inlet temperature, as 74C might be considered a rather extreme condition. For this more moderate rise in inlet temperature (11C), the savings in power loss are still substantial (approximately 22 per cent), and the maximum pad temperature is only 111C (still 13C lower than the conventional bearing with a 49C inlet temperature). Table 3 shows that, as a result of increasing the oil inlet temperature, significant savings in power loss are achieved at all pads.

Figure 8 shows the calculated pad temperature distributions of the conventional and LEG bearings. This figure has two interesting features. Firstly, raising the inlet temperature to the LEG bearing has most effect on the operating temperature of the unloaded pads (3, 4 and 5). Since little heat is, in fact, generated at the unloaded pads, this dramatic increase in temperature is probably the result of introducing hot oil at the leading edges of the pads, and not the consequence of hot oil carry over. Secondly, the leading edge temperatures, and therefore the operating temperatures, of the loaded pads of the conventional bearing are similar to those of the LEG bearing (with the raised inlet temperature). In the case of the conventional bearing, leading edge oil temperatures are strongly influenced by hot oil carry over from the preceding pads (see equation 1). However, the effect of hot oil carry over is only minimal in the case of the LEG bearing, although the oil supply temperature is higher. The overall result is that, when comparing LEG and conventional bearing temperatures, differences on the loaded pads are less significant than those of the unloaded pads.

The elevated operating temperature of the unloaded pads has two effects on bearing performance. Firstly, power loss in the upper half of the bearing is lowered as a result of reducing viscous shearing losses. Secondly, the increased temperature gives rise to a hotter running shaft and this translates into further reductions in power loss in the bottom half of the bearing as oil film temperature profiles are modified. In the example given in Figure 8, the calculated shaft temperature for the LEG bearing is 68C with an inlet temperature of 49C. This increases to 85C when the inlet temperature is raised to 74C.

6 CONCLUSIONS

Testing of a 0.098 m diameter five shoe, tilting pad journal bearing, fitted with a leading-edge-groove lubricant supply method, at loads of 5.18 kN and 11.1 kN and shaft speeds up to 16500 rev/min, has resulted in the following conclusions:

1. The maximum temperature of the loaded pad(s) is similar at lower shaft speeds, regardless of whether the load is "on pad" or "between pads". Only at speeds above 12000 rev/min does the difference become noticeable, when LOP is hotter.

2.Reducing the "bearing" clearance from 0.102 mm to 0.076 mm had little effect on the operating temperature of the bearing. However, a further reduction to 0.051 mm resulted in a steep rise in the maximum temperature of the babbitt lining.

3. With a load of 11.1 kN, the bearing operated satisfactorily in reverse rotation at speeds up to 12000 rev/min. The tests indicated that higher speeds, at lower loads, are possible.

4. Calculations have shown that substantial reductions in the power loss of the LEG bearing are possible when the oil inlet temperature is raised. In the example cited in this paper, the savings can be as large as 39 per cent in comparison to the losses of the conventional bearing. This assumes that the maximum temperature of both bearings is similar.

5. The test data has been compared with results from a computer model and correlation has been found to be acceptable. This would seem to confirm that the lower operating temperature of the LEG bearing is the result of reducing hot oil carry over.

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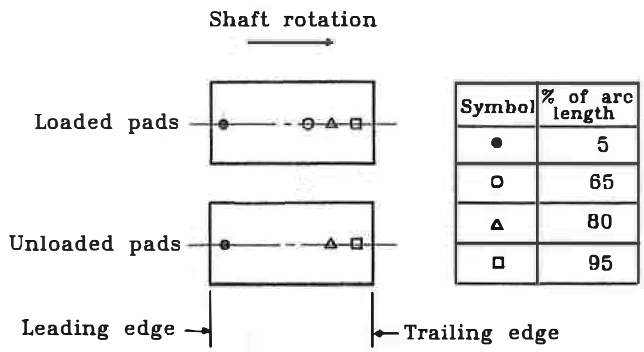

Symbol	% of arc length
●	5
○	65
△	80
□	95

Fig.1 Thermocouple locations

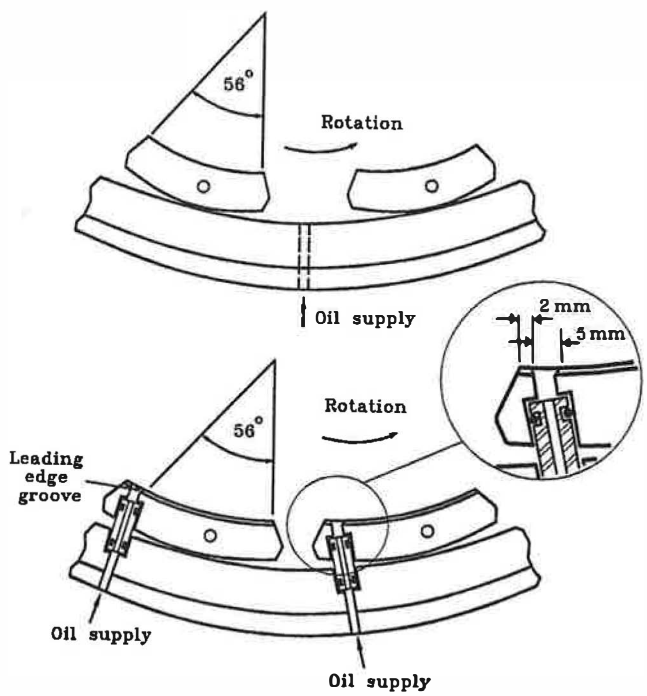

Fig.2 Pad arrangements:
a) conventional b) LEG

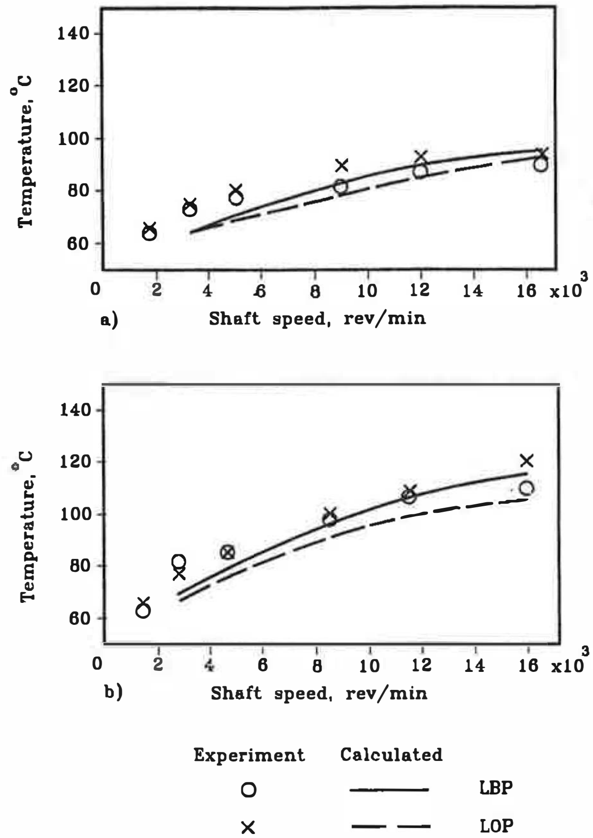

Fig.3 Maximum pad temperatures for LBP and LOP
Group 2 bearing, nominal flow rates
a) 5.18 kN (1163 lb) load
b) 11.1 kN (2500 lb) load

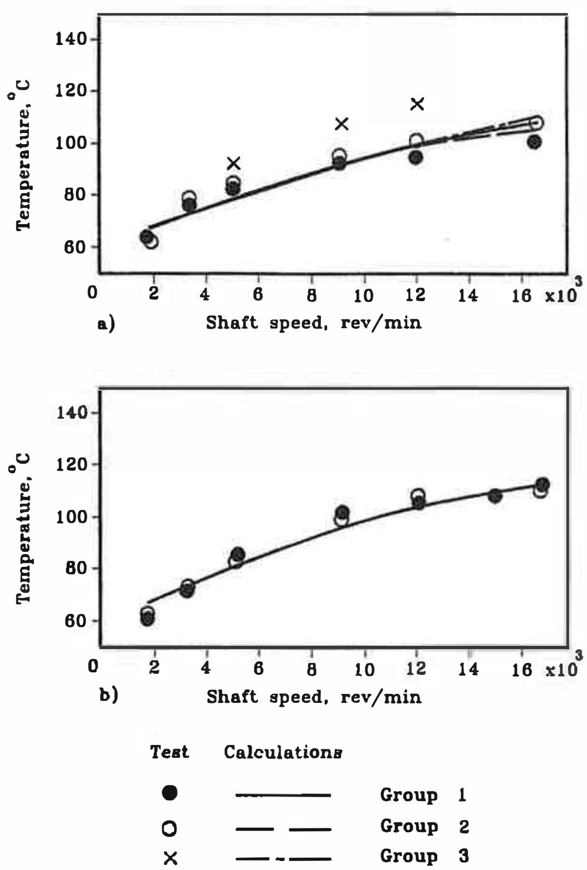

Fig.4 Maximum pad temperatures for different bearing clearances
Nominal flow rates, 11.1 kN (2500 lb) load
a) LBP b) LOP

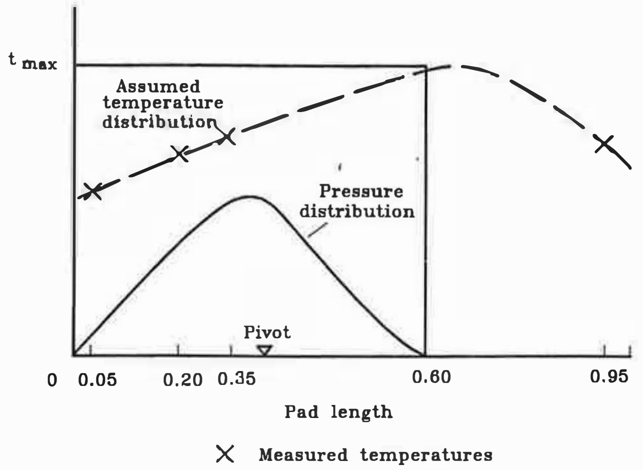

Fig.5 Estimation of maximum pad temperature - reverse rotation tests

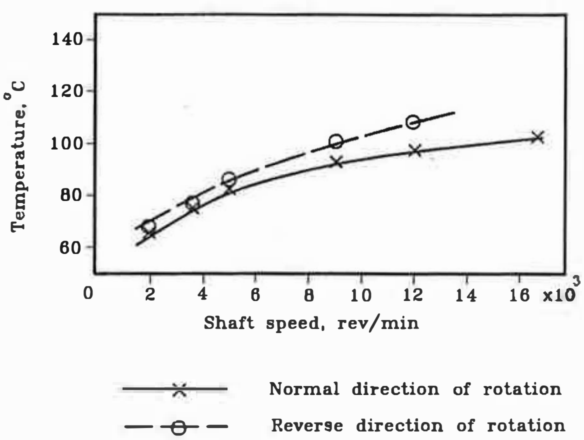

Fig.6 Maximum pad temperatures for forward and reverse directions of rotation
Group 1 LEG bearing, nominal flow rates, 11.1 kN (2500 lb) load, LBP

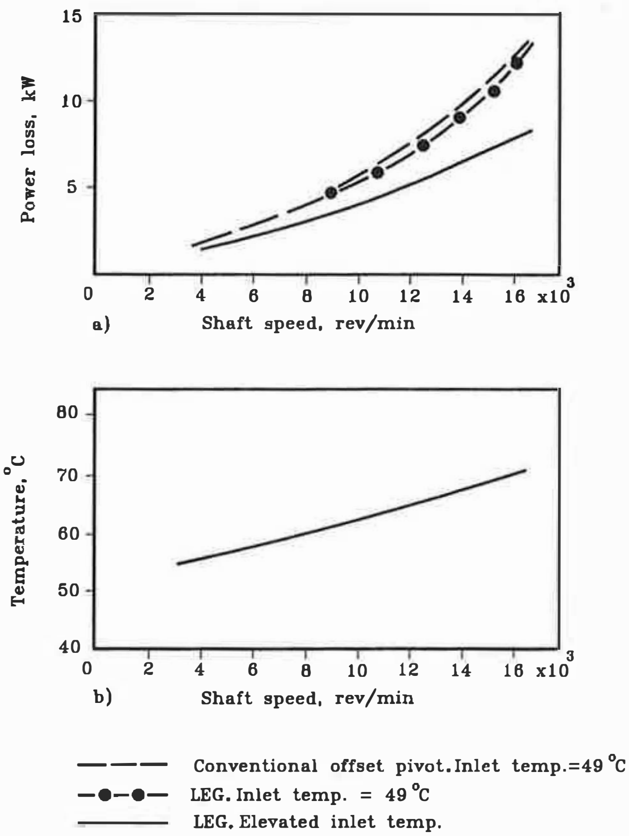

Fig.7 Power loss analysis
Group 1 bearings, nominal flow rates, 11.1 kN (2500 lb) load, LBP
a) calculated power losses
b) elevated oil supply temperatures - LEG bearing

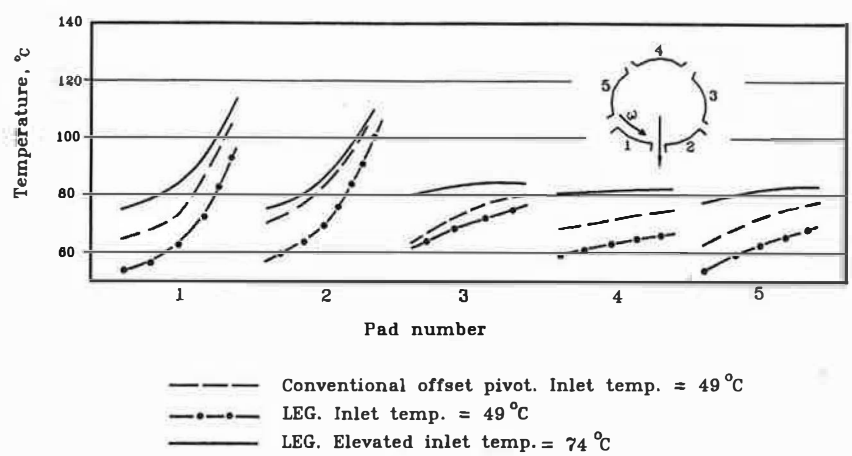

Fig.8 Calculated pad temperature profiles for 16500 rev/min
Group 1 bearings, nominal flow rates, 11.1 kN (2500 lb) load, LBP