Influence of oil viscosity grade on thrust pad bearing operation

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Abstract: The effect of oil viscosity grade on the performance of tilting pad thrust bearings is examined in a wide range of shaft speeds and specific bearing loads. Data being discussed were obtained in tests carried out with 228 mm outer diameter (o.d.) and 267 mm o.d. bearings lubricated with ISO VG32 and ISO VG68 mineral oils in a fully flooded mode. In a low-speed range (below 40 m/s), the performance of the 228 mm bearing is analysed in terms of pad and collar temperatures, power loss and oil film thickness. Pad temperature and power loss are employed in the analyses of the 267 mm o.d. bearing operated at high sliding speeds of up to 115 m/s. The results show that there is a significant effect of oil viscosity grade on bearing main operating parameters. The substitution of ISO VG32 oil for ISO VG68 oil results in considerably reduced pad temperatures, lower power loss and thinner oil film thickness. However, no measurable difference in power loss was observed after the onset of turbulence. The influence of oil viscosity grade on pad temperature pattern is analysed. The thermal effect of oil viscosity grade on pad temperature pattern and power loss is also compared with the effect of offset pivot.

Keywords: tilting pad thrust bearing, viscosity grade, temperature, oil film thickness, power loss

1 INTRODUCTION

The performance of a tilting pad thrust bearing can be best quantified in terms of developed power loss, operating temperature and oil film thickness. These parameters are all affected by oil viscosity. Designers and the end-users of rotating machinery are often confronted with the question as to which oil viscosity grade should be chosen to provide the most efficient bearing operation. It is commonly supposed that thinner oils provide lower energy consumption and lower operating temperatures. At the same time, low-viscosity oils develop thinner oil films, which can jeopardize bearing safety. Hence, thicker oils are usually used at low speed and thinner ones at high speed. However, to what extent will the main operating parameters be affected if the thrust bearing operates with an oil of higher or alternatively lower viscosity grade?

Typical mineral oils used for bearing lubrication in turbomachinery are ISO VG32, VG46 and VG68. Most of the data published are for light turbine oil ISO VG32,

and only very few articles have presented data on the influence of oil viscosity on bearing performance. The effect of the grade of oil used on the limits of safe operation for a 196 mm mean diameter eight-pad bearing was presented in reference [1] through a bearing duty diagram. The approximate viscosities of the oils used in that analysis were 15.5, 27.0 and 53.0 mm²/s at 60 °C. The first and the second oils apparently correspond to VG32 and VG68 oils. It follows from the bearing duty diagram that the substitution of the VG68 oil for the VG32 oil significantly extends the limit of safe bearing operation at high and low speeds. As an example, the limit is extended by 50 per cent at a mean sliding speed of 3 m/s and by 40 per cent at a speed of 100 m/s.

Test data for a 267 mm o.d. six-pad bearing were presented in reference [2]. A heavy oil (94 mm²/s at 37.8 °C) was replaced with light turbine oil (VG32). Test data showed that the lighter oil cut power losses by 25 per cent. It should be noted that the bearing lubricated by the lighter oil required lower supply oil flowrate so that some of the reduction in loss is due to less oil.

The influence of oil viscosity on power loss of a spring-supported bearing was studied in reference [3]. Substitution of ISO VG32 oil for ISO VG68 oil reduced power loss by 30 per cent at a specific bearing load of

4 MPa (specific bearing load is defined as the loading force divided by the area of the bearing load-carrying surface). The temperature, measured at the mid-point of pad thickness at a location 25 per cent circumferentially from the trailing edge and 62 per cent radially from the inner edge, was also reduced by 3–6 K. The calculated reduction in minimum oil film thickness was 18–20 per cent. It was concluded that bearing performance could be significantly improved by substitution of the thinner oil and simultaneous increase in oil pot temperature. The minimum film thickness obtained, 10–15 μm, was considered adequate to prevent wiping.

Pad maximum temperature data from 12 test bearings combined with previously published results of other researchers were analysed in reference [4] at sliding speeds of up to 80 m/s and a mean pad pressure of up to 4.1 MPa. The position of the maximum temperature for 12 test bearings was at the centre of the trailing outboard quadrant. It was concluded that the effect of oil viscosity grade was 'of minor importance over the range of typical oils used in tilting pad bearings (ISO grades 32 to 68)'.

Thus, when it comes to typical mineral oils used for bearing lubrication in turbomachinery, none of the published information taken collectively provides a definite answer to the question of the importance of viscosity grade influence on bearing operating charac-teristics. The purpose of the present paper is therefore to clarify this ambiguity and to provide detailed informa-tion necessary to quantify the effect of oil viscosity grade on bearing main operating parameters: temperature, power loss and oil film thickness.

2 TEST FACILITIES

Two test rigs were used in the experimental programme. A test rig with a 228 mm o.d. bearing was employed in the low-speed range, while high-speed runs were carried out in the 267 mm o.d. bearing test rig.

2.1 Test rigs

The test rig with the 228 mm bearing is fully discussed in reference [5]. A brief description only is given here. A general arrangement of the test rig is shown in Fig. 1. Two identical test bearings (1, 2) are mounted in the guiding holders (3), which can slide in the housing (4) positioned on steel wheels (5, 6) with rolling element bearings. Such an arrangement allows direct measure-ment of the bearing power loss since the housing is free for rotation by the action of bearing frictional torque. The rotation is prevented by a load cell (7), which gives the value of torque. The test bearings are loaded back to back against separate collars. The required load is applied by means of four hydraulic cylinders (8, 9)

located between the bearing holders. Bearing load is calculated from the measured pressure in the hydraulic system. The main shaft (10) with two integral collars is supported at each end by journal bearings. Power to the test rig is provided by a 143 kW variable-speed d.c. motor.

A schematic drawing of the test rig for the 267 mm o.d. bearing is shown in Fig. 2. A variable-speed gas turbine with a rated output of 810 kW and a controllable test speed range from 4000 to 14000 r/min is the prime mover. The turbine is connected to the test shaft by means of a flexible coupling. Two identical housings external to the turbine enclosure contain the bearing components. The forward housing adjacent to the turbine is firmly secured to the foundation, while the aft housing is restrained but is free to slide axially. Each housing contains a horizontally mounted double ele-ment thrust bearing and a tilting pad journal bearing for radial support of the test shaft. Each housing is also equipped with separate lubricant supply and drain lines.

Axial load is applied by means of an external hydraulic system, which transmits force directly to the sliding housing. As a result of the single applied force, both thrust bearings A and D experience the same loading, while thrust bearings B and C remain unloaded. A load cell is mounted on the end of the sliding housing for the purpose of measuring the total applied thrust load.

2.2 Test bearings

The bearings involved in both test series are geome-trically similar. Each has six babbitted pads and a mechanical equalizing system. The main geometrical characteristics of the bearings are listed in Table 1. The individual active and inactive 'slack' bearing elements in the 267 mm o.d. bearing test rig (Fig. 2) are identical in dimensions and design.

2.3 Instrumentation

In the 267 mm o.d. bearing, small thermocouples are imbedded in the babbitt metal of each thrust bearing pad. The actual thermocouple junction is positioned within 0.76 mm of the babbitt surface. All pads have a thermocouple at the 75/75 location. This is an industry standard location and is defined as the location on the pad face that is 75 per cent of the distance from the leading edge and 75 per cent of the distance from the inside pad radius outwards. Also, an array of nine thermocouples is installed in one pad of each of the loaded bearings, A and D, in order to determine the temperature gradient in both the radial and circum-ferential directions. They are distributed in the same way as in the 228 mm o.d. bearing (Fig. 3a).
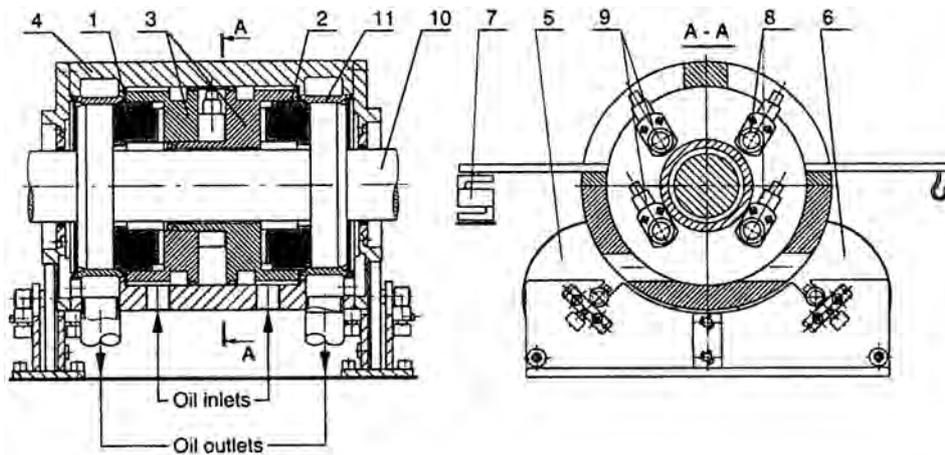
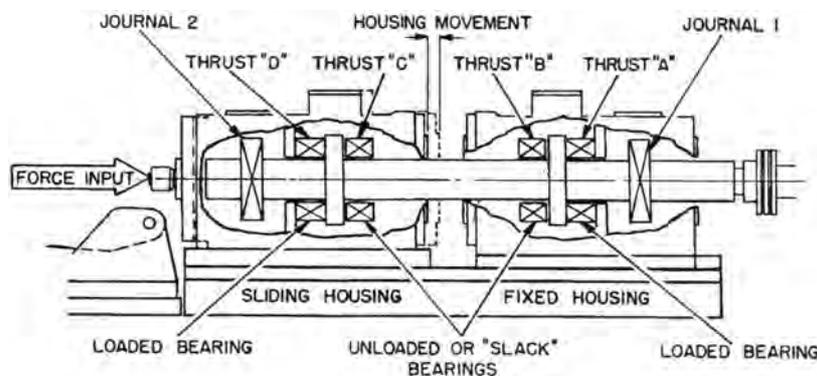**Fig. 1** Schematic of 228 mm o.d. thrust bearing test rig

Fig. 2 Schematic of 267 mm o.d. thrust bearing test rig

Table 1 Test bearing characteristics

Outer diameter (mm)	228.6	266.7
Inner diameter (mm)	114.3	133.35
Number of pads	6	6
Bearing area (mm²)	26 130	35 550
Pad height (mm)	28.58	31.75
Pivot position (offset) (%)	60	50 and 60
Pivot type	Spherical	Spherical
Bearing set clearance (mm)	Variable	0.457

The oil supply flowrate, oil supply and drain temperatures are also measured. They are then utilized for power loss calculation based on the calorimetric technique.

The instrumentation of the 228 mm o.d. bearing is shown in Fig. 3. Thermocouples are mounted in pads 1, 3, 4 and 6. In order to find the temperature pattern over the entire pad surface, an array of nine thermocouples is evenly distributed across the pad surface in pad 3. A tenth thermocouple is located at the 75/75 position. This arrangement is repeated in the diametrically opposite pad 6. All thermocouples are mounted in the backing material as shown in Fig. 3b.

Two capacitive sensors, used for determining the thickness of the oil film during the tests, are located at the leading and trailing edges, as shown in Fig. 3a. A thermistor of negative temperature coefficient (NTC) type was installed 1.5 mm beneath the surface of the collar face in the radial position corresponding to the location of the T75/75 pad thermocouples. Temperature measured by this sensor is denoted as T75 in the subsequent analysis. Uncertainties of the measured quantities are listed in Table 2.

3 TEST CONDITIONS

1. Tests were performed over two ranges of speeds and loads. For the 228 mm o.d. bearing, the specific loads were 0.69, 1.38 and 2.07 MPa, while the mean sliding speeds were in the range 10–30 m/s. For the 267 mm o.d. bearing, specific loads were varied between 0.69 and 3.45 MPa, while mean sliding speeds were varied between 40 and 115 m/s.
2. All bearings were operated in a fully flooded condition.
3. The oil flowrate for the 228 mm o.d. bearing was held constant and equal to 15 l/min for all load/speed combinations. For the 267 mm o.d. bearing, the oil
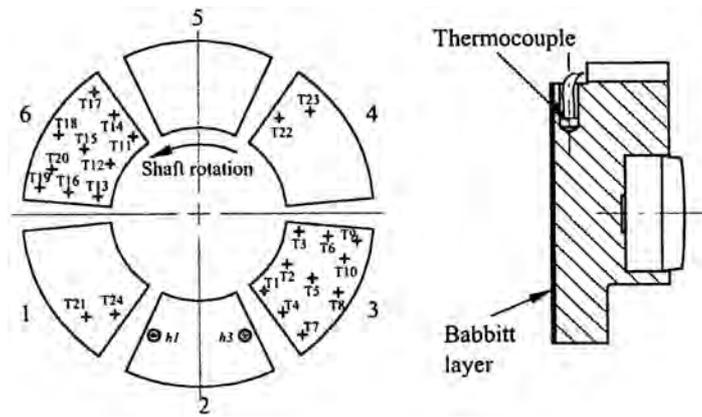
Fig. 3 (a) Instrumentation of the 228 mm o.d. bearing and (b) thermocouple mounting in the bearing pad

Table 2 Uncertainties of the measured quantities

	228 mm o.d. bearing	267 mm o.d. bearing
Temperature (°C)		
Thermocouple	±1	±0.5
NTC thermistor	±0.3	—
Frictional torque (N m)	±0.2% FS	—
Specific bearing load (MPa)	±1% FS	±0.2%
Shaft speed (r/min)	±0.1%	±50%
Supply oil flowrate (l/min)	±0.5%	±1.0%

flowrate was adjusted for each load and speed combination according to recommendations of the bearing manufacturer.

4. The oil supply temperature was 46 °C.
5. Two turbine oils, ISO VG32 and ISO VG68, were used in the tests. Their properties are listed in Table 3.
6. Steady state mode for the 228 mm o.d. bearings was ensured by checking temperature variation over a period of 15 min. If the variation did not exceed 0.2 K, the thermal equilibrium was judged to be reached. For the 267 mm o.d. bearings, two sets of temperature readings were taken 10 min apart and compared until the variation did not exceed 1.0 K.

4 RESULTS

The pad temperature and power loss are recorded for both bearings. In addition, collar temperature and oil film thickness are given for the 228 mm o.d. bearing.

Test results are presented as a function of mean sliding speed, which is defined at the pad middle radius. Data are grouped in two ranges of mean sliding speed: 10–30 and 40–120 m/s. Performance comparisons for the two oil grades are based on identical operating conditions of speed, load, oil flow and oil supply temperatures.

4.1 Temperature

A typical set of results for the 228 mm o.d. bearing is shown in Fig. 4. The T75/75 temperature, an average value for four pads, is plotted as a function of speed for three specific bearing loads. It can be observed that the bearing lubricated by thinner oil runs approximately 5 K cooler. This difference is only slightly affected by shaft speed and load. A similar trend can also be noted by examining Fig. 5 which depicts collar temperature T75 for the same operating conditions as in the previous figure. The bearing lubricated by ISO VG32 oil runs with a collar temperature up to 7 K lower.

The T75/75 temperature for the 267 mm o.d. bearing is presented in Fig. 6. It is an average value for six pads. Once again, the temperature data show that the bearing runs cooler when lubricated by thinner oil. The difference increases with rise in load at all sliding speeds. At the same time, increasing the speed has only a slight effect on the temperature difference. An inflection that can be observed on some temperature curves indicates a transition in flow regime from laminar to turbulent. Comparison of the temperature curves for the highest load, 3.45 MPa, indicates that oil viscosity grade

Table 3 Oil properties

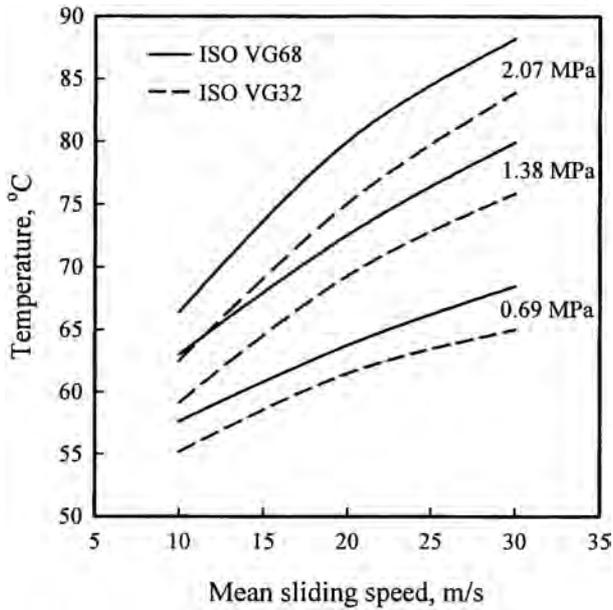|	Viscosity at 40 °C, (mm²/s)	Viscosity at 100 °C, (mm²/s)	Density at 15 °C, (kg/m³)	Specific heat (J/K m³)	Heat conductivity (J/m K)
ISO VG32	30	5.2	867	1.74 × 10⁶	0.13
ISO VG68	67	8.9	877	1.74 × 10⁶	0.13

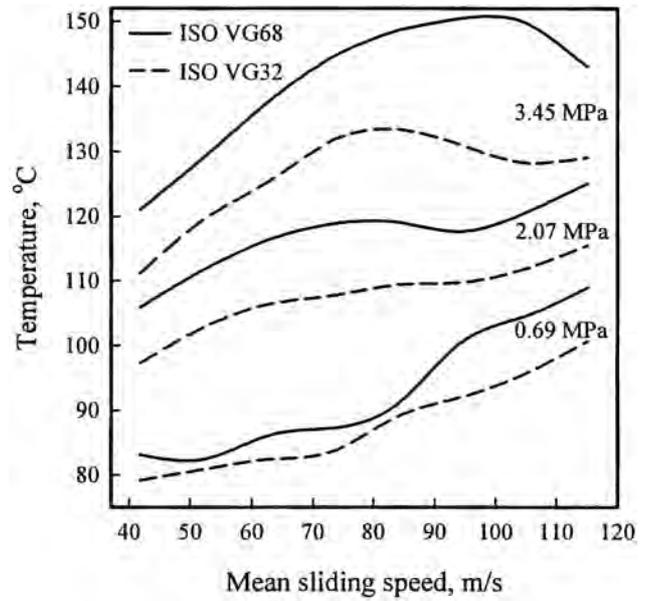**Fig. 4** Averaged pad temperature T75/75, 228 mm o.d. bearing

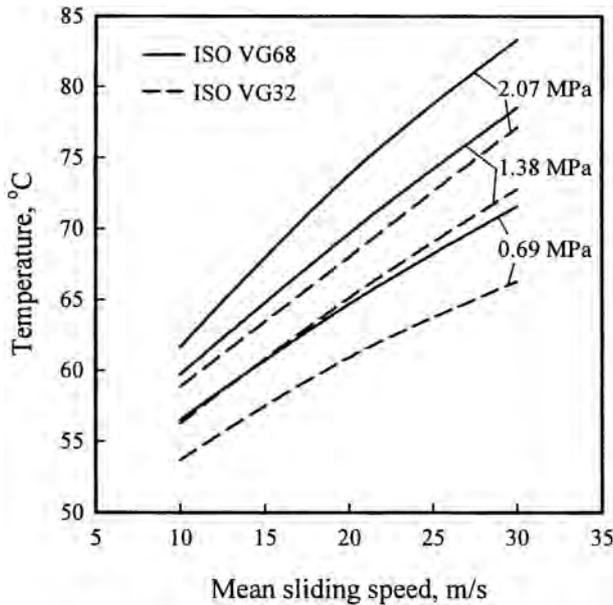**Fig. 6** Averaged pad temperature T75/75, 267 mm o.d. bearing

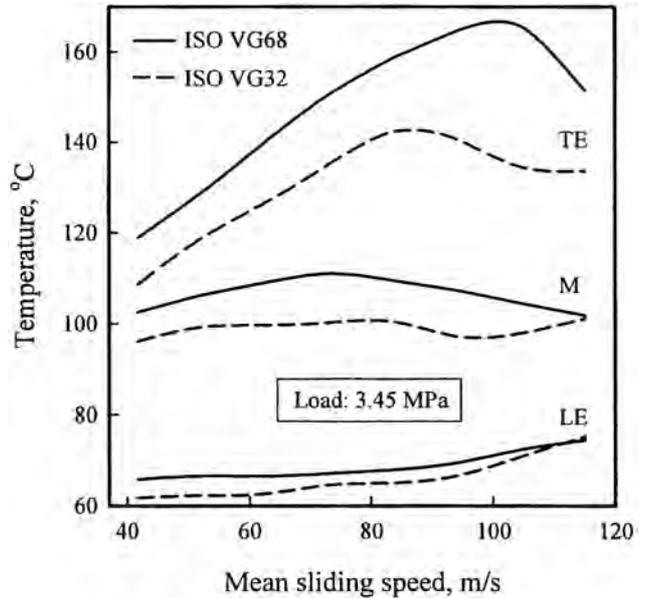**Fig. 5** Collar temperature T75, 228 mm o.d. bearing

Fig. 7 Centre-line leading edge (LE), middle (M) and trailing edge (TE) temperatures, 267 mm o.d. bearing

affects the transition point, namely sliding speed, at which the temperature trend changes direction. Substitution of the thicker ISO VG68 oil shifts the transition to a higher speed, increasing the difference in the T75/75 temperatures up to 22 K. Figure 7 shows temperatures measured at the circumferential centre-line of the pad near the leading edge (LE), in the middle (M) and near the trailing edge (TE). It can be observed that only the trailing edge temperature is greatly affected by sliding speed and by the transition in flow regime. The leading edge and middle temperatures are only slightly affected.

4.2 Power loss

Power loss for the 228 mm o.d. bearing, calculated from the measured frictional torque and angular speed, is depicted in Fig. 8. As would be expected, use of the thicker oil increases power loss. The difference in power loss is noticeably affected by shaft speed. Since the amount of oil supplied to the bearing is kept constant, the increase in power loss is greater at 30 m/s owing to the increased contribution of churning losses. At this sliding speed the amount of oil supplied to the bearing is at the recommended level, and increase in power loss is
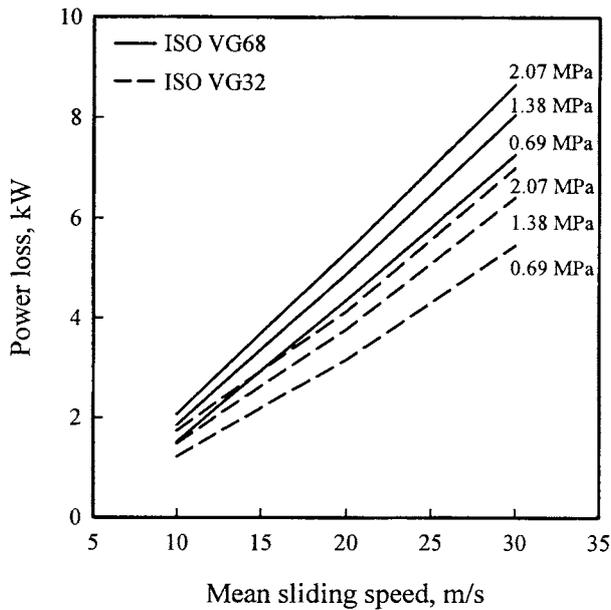
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Fig. 8   Power loss, 228 mm o.d. bearing

Fig. 10   Power loss, 267 mm o.d. bearing

approximately 20–30 per cent depending on bearing load.

Power consumed by the 267 mm o.d. bearing is calculated by the calorimetric technique from the flowrate of the oil, its specific heat and the rise in temperature between supply and drain. Power loss values presented in Fig. 9 apply to the complete bearing assembly which includes active and inactive 'slack' bearings. At lower speeds, the power loss for ISO VG68 oil is clearly higher, which is consistent with 228 mm data. There is, however, no measurable difference in power loss at high sliding speeds. This is further illustrated by Fig. 10 which gives power loss plotted against specific bearing load at four sliding
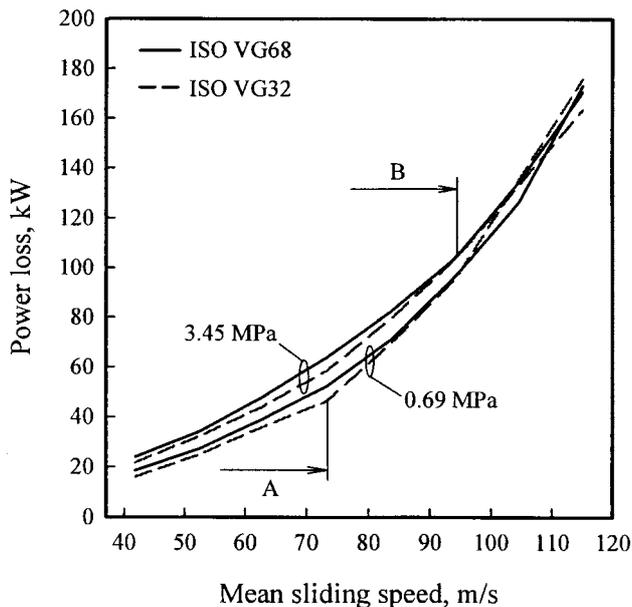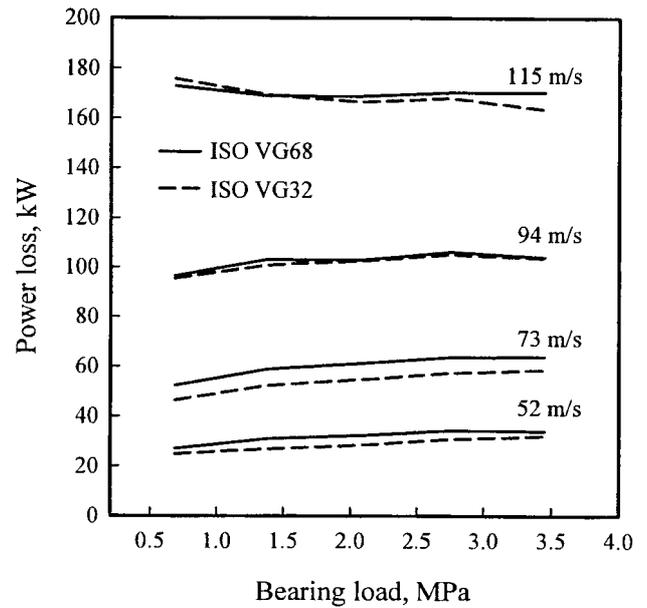
speeds. The reason for similar energy consumption is a transition from a laminar to a turbulent flow regime. This transition starts first in the inactive bearing, point A, owing to its thicker film thickness. The difference in power loss then gradually decreases until it vanishes at B, which is the transitional point for the active bearing. It is interesting to note that load variation in the active bearing does not affect the point of transition in the inactive bearing.

4.3   Oil film thickness

Oil film thickness for the 228 mm o.d. bearing is plotted in Fig. 11. It is measured in pad 2 close to the leading ($h1$) and trailing ($h3$) edges, as shown in Fig. 3. Both leading and trailing edge films are considerably reduced after substitution of the thinner VG32 oil. Depending on load and speed, this reduction varies between approximately 16 and 22 per cent. Decrease in the leading edge film thickness is greater than that in the trailing edge film, indicating that pad inclination is lower when a thinner lubricant is used. Pad inclination is also affected by shaft speed, being greater at higher speed. At the same time, the reduction in oil film thickness at the trailing edge is not noticeably affected by sliding speed.

5   DISCUSSION

Over the range of operating conditions tested, a distinct effect of oil viscosity grade on the bearing main operating parameters has been established. There is then a question of whether this effect is of importance compared with conventional ways of improving bearing
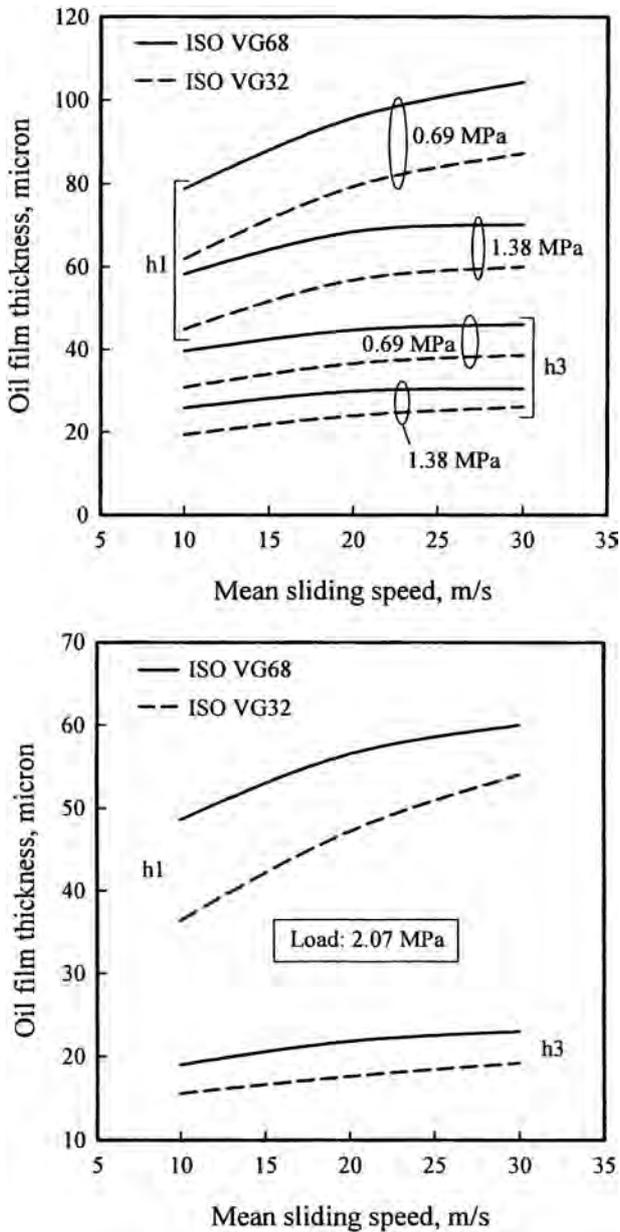

Fig. 9   Power loss, 267 mm o.d. bearing

Fig. 11 Oil film thickness

However, lower temperatures and a reduction in energy consumption of up to 18 per cent, shown in the tests, are at the expense of a thinner oil film which means a lower load-carrying capacity. A 16–22 per cent reduction in oil film thickness near the trailing edge was observed in the tests. It must thus be decided in each particular case whether it is worth substituting a thinner oil for a thicker one. In low-speed bearings, power loss is usually not of primary concern compared with minimum allowable oil film thickness, so a thicker oil is desired when resulting pad temperatures are acceptable. A thinner oil can be substituted, giving pad temperature and power loss advantages in cases where loads are low such that the resulting film thickness is acceptable. If reduced energy consumption and load capacity are of importance, a synthetic lubricant of lower viscosity grade can successfully be used without affecting the margin of safety, as shown in reference [8].

At high sliding speeds the main concern is to reduce energy consumption and to decrease maximum pad temperature, which imposes a limit of safe operation. As for oil film thickness, it is usually always adequate owing to high speed. Data on the magnitude of oil film thickness in this region can be found in references [9] and [10].

A common solution to reduce power loss, and at the same time pad temperature, is to use direct lubrication such as the leading edge groove [11–13] or spray jet [14, 15] system. This measure allows up to 70 per cent reduction in power loss [15] and up to 20 K reduction in temperature [12, 13], depending on operating conditions and the system used. In the present study a smaller reduction in energy consumption is achieved and only before the onset of turbulence. On the other hand, the 267 mm o.d. bearing lubricated by the ISO VG32 oil operates with T75/75 temperatures 14–24 K lower than when lubricated by the ISO VG68 oil. This difference is of the same magnitude as the one obtained with direct lubrication.

As has been previously mentioned, reduced temperatures are favourable for longer service life of the lubricant. Depending on the operating conditions, temperature T75/75 can be lower than pad maximum temperature as the hot spot may change its location. It is thus equally important to analyse the temperature pattern over the entire white metal face in order to reveal the effect of oil viscosity grade on pad maximum temperature. This can conveniently be done by using isotherms. For this purpose, data recorded from ten thermocouples mounted in one of the 267 mm o.d. bearing pads are curve fitted in the circumferential and radial directions. In addition, temperature values at other locations of the pad are also obtained by this procedure. These combined measured as well as interpolated and extrapolated temperatures are plotted as isotherms.

Figure 12 shows isotherms for the centre-pivot pad for a number of sliding speeds and a specific load of
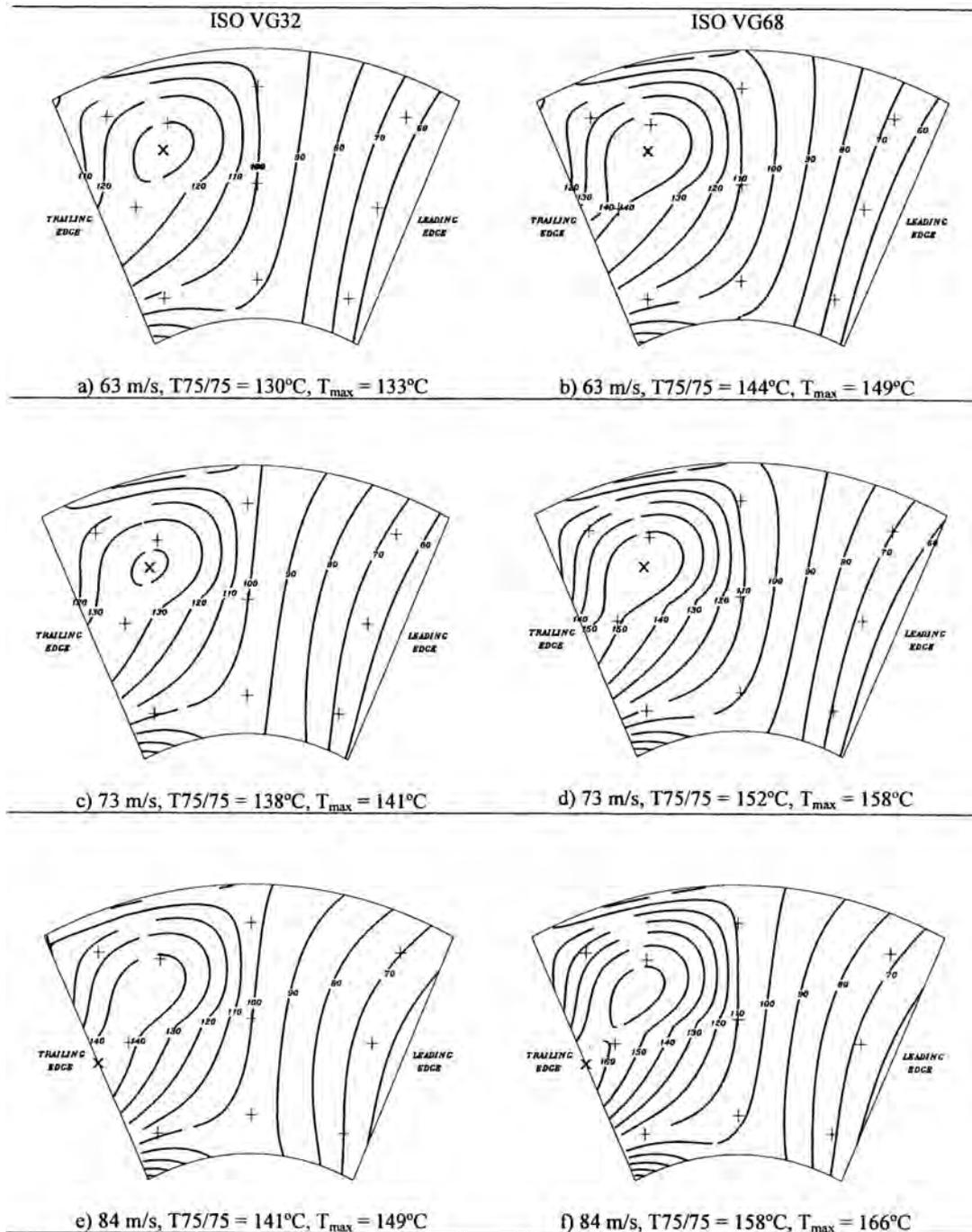
operation. In the low-speed range, which is typical for large bearings, the limit of safe operation is set by the minimum allowable film thickness [6] while the main design challenge is to control pad thermal deflection. This is achieved by various means. For example, application of a double-layer pad design with cooling ducts, as proposed in reference [7], allows a significant reduction in pad crowning and a 4–10 K reduction in pad maximum temperature. In the present study, substitution of ISO VG32 oil decreased the bearing temperature by up to 7 K. The effect of oil viscosity grade is thus of importance.

Reduced temperature is also of advantage from the point of view of a lower rate of oil degradation.

a) 63 m/s, T75/75 = 130°C, T$_{max}$ = 133°C

b) 63 m/s, T75/75 = 144°C, T$_{max}$ = 149°C

c) 73 m/s, T75/75 = 138°C, T$_{max}$ = 141°C

d) 73 m/s, T75/75 = 152°C, T$_{max}$ = 158°C

e) 84 m/s, T75/75 = 141°C, T$_{max}$ = 149°C

f) 84 m/s, T75/75 = 158°C, T$_{max}$ = 166°C

Fig. 12   (continued)

3.45 MPa, for the VG32 and VG68 oils respectively. It can be observed that the temperature rise from the leading to the trailing edge is greatly affected by speed and is on average 20 K lower for the ISO VG32 oil. Comparisons at the leading edge of the pads show that the lubricants are close in temperature. Moving further across the pad to the centre, it can be seen that the isotherms are widely and uniformly spaced in a near-radial pattern. At the pad centre, ISO VG68 is 10 K hotter at all sliding speeds. The temperature rise for both oils in the first half of the pad is not affected by speed. This is in contrast to the second half, where a strong effect of speed on temperature gradient can be observed.

A distinct hot spot is produced near the 75/75 location with a temperature close to T75/75 (Figs 12a to d). At sliding speeds above 73 m/s (Figs 12e to h) the hot spot moves towards the trailing edge, mid-radius. For the ISO VG68 oil this move is slightly more prolonged and is completed at 94 m/s. Thereafter, the difference
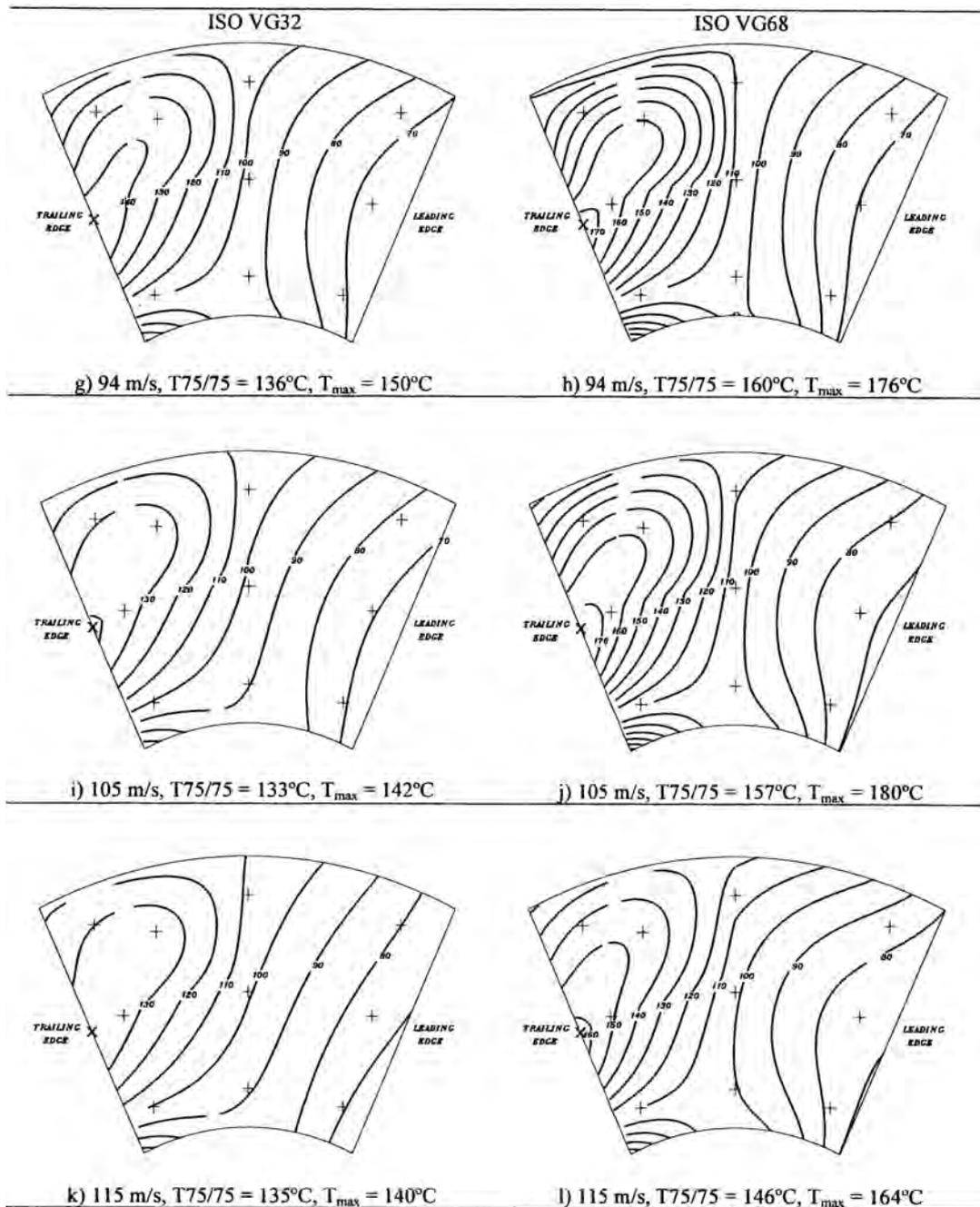
Fig. 12 Pad temperature patterns, 267 mm o.d. bearing, centre-pivot pad

between T75/75 and maximum pad temperature becomes greater. Movement of the hot spot occurs before the transition in the flow regime. The transition, which is also affected by oil viscosity grade, starts after 94 m/s in the bearing lubricated by ISO VG32 (Fig. 12g). For the bearing lubricated by ISO VG68 the transition speed is around 105 m/s (Fig. 12j). After the transition, the temperature gradient in the trailing part of the pad is significantly reduced, as manifested by more widely spaced isotherms (Figs 12i, k and l).

In order to estimate the significance of the oil viscosity grade thermal effect, it is of practical interest to compare this effect with that of an offset pivot. The use of a 60 per cent offset pivot instead of a centre pivot is a common way of reducing temperature in unidirectional bearings. Figure 13 shows averaged pad temperatures T75/75 for the 267 mm o.d. bearing lubricated by the ISO VG32 oil. It can be observed that offset pads operate at a 10–20 K lower temperature compared with centre-pivot pads. This difference is of the same magnitude as in the case of oil viscosity grade.

The pad temperature pattern is, however, affected to a much greater extent when an offset pivot is employed. Using the same technique as mentioned
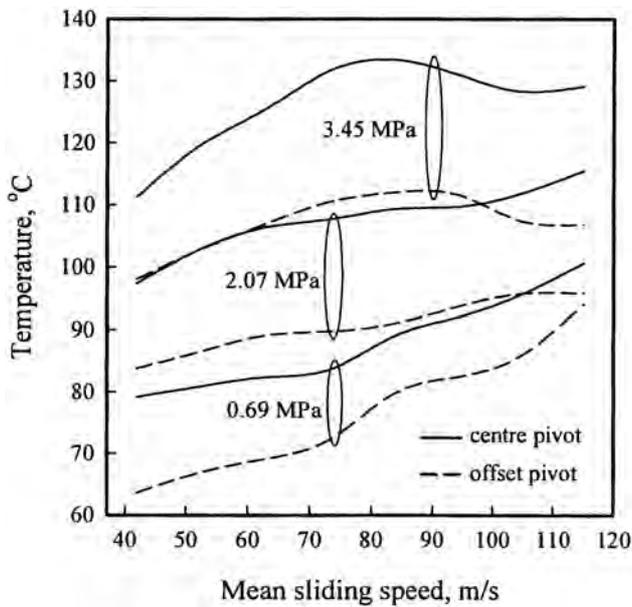
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Fig. 13 Averaged pad temperature T75/75, 267 mm o.d. bearing with offset and centre-pivot pads, ISO VG32 oil

a) 73 m/s, T75/75 = 114°C, T_{max} = 164°C

b) 94 m/s, T75/75 = 113°C, T_{max} = 182°C

c) 115 m/s, T75/75 = 109°C, T_{max} = 147°C

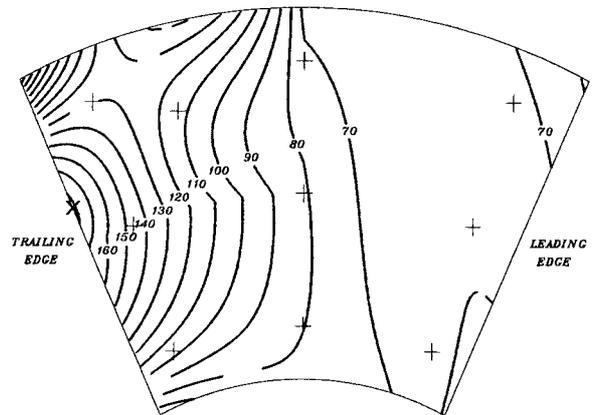Fig. 14 Pad temperature patterns, 267 mm o.d. bearing with offset pads, ISO VG32 oil

above, isotherms can be plotted over the working face of an offset pad, as shown in Fig. 14.

In comparing Figs 12c, g and k with Figs 14a to c, the following can be observed. In the inlet part of the pad, leading edge to centre, the temperature gradient is much lower for the offset pivot, whereas in the trailing part, after the pivot point, it is steeper. For example, at 94 m/s sliding speed, the temperature rise within the inlet part is about 40 K for the centre-pivot pad (Fig. 12g), whereas it is only 10 K for the offset pad (Fig. 14b). In the trailing part the difference in temperature is much higher.

The offset pivot hot spot is located at the trailing edge and does not change its position with speed. This, combined with the steeper temperature gradient, makes the difference between T75/75 and T_{max} much greater. It should be noted that values of T_{max} are obtained by extrapolation and so have error. Maximum temperature T_{max} is used here more for judgement of location. Nevertheless, even the measured mid-radius trailing edge temperature is higher for the offset pads in some cases. This is an important finding and needs some comment, as it was never previously addressed in the literature. It is generally accepted that the offset pads run cooler compared with centre-pivot pads. This is based on the typical thermocouple location in real applications like the 75/75 position, and, indeed, the present tests show that this location is much cooler for all operating conditions. At the same time, pad maximum temperature can be higher. Judgement is needed to assess the two parameters. On the one hand, high temperature accelerates the process of oil degradation and increases the risk of thermal fatigue. On the other hand, high trailing
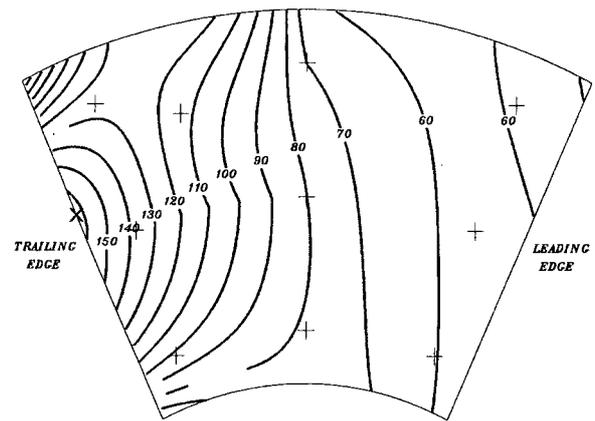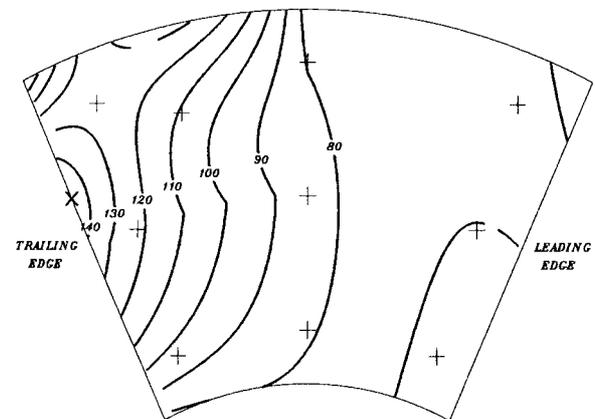
edge temperatures are in a lower film pressure region, which is less critical regarding white metal strength compared with higher film pressure regions inboard. It should be noted that these comparisons and comments are for the specific geometry tested. For example,
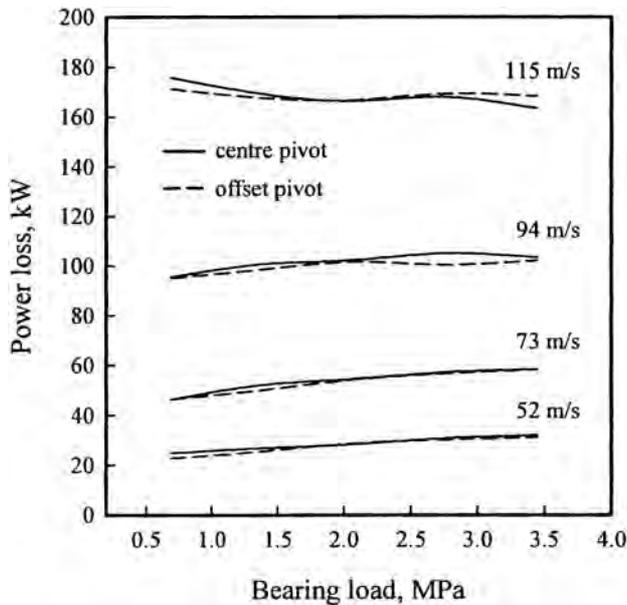
Fig. 15 Power loss, 267 mm o.d. bearing with offset and centre-pivot pads, ISO VG32 oil

increasing the offset to 65 per cent can further reduce trailing edge temperatures.

With regard to energy consumption, there is no measurable effect of offset pivot on power loss, as follows from Fig. 15. Power loss values presented in Fig. 15 apply to the complete bearing assembly which includes active and inactive 'slack' bearings. It can be observed that energy consumed by the offset pad bearing is essentially the same as the centre-pivot bearing for each speed and load. This is in contrast to the effect of oil viscosity grade when power loss for the thinner oil is lower before the onset of turbulence.

6 CONCLUSIONS

A comparative experimental study of the influence of oil viscosity grade on tilting pad thrust bearing performance in a wide range of shaft speeds and loads has been carried out. The thermal effect of the oil viscosity grade has been compared with the corresponding effects obtained by alternative techniques for improving bearing performance such as double-layer pad design, direct lubrication and 60 per cent offset pads. The following conclusions can be drawn:

1. Bearing temperature is significantly affected by oil viscosity grade. For comparison, at low sliding speeds the temperature decrease obtained by substitution of the thinner oil is similar to that realized with a double-layer pad design. At high sliding speeds the reduction in temperature as a result of substitution of ISO VG32 for ISO VG68 oil is comparable with that achieved by using direct lubrication or 60 per cent offset pads.

2. ISO VG68 oil provides thicker oil films which means a higher load-carrying capacity. This is of importance at low sliding speeds where capacity is not also restricted by pad temperature.
3. The energy consumed by ISO VG68 oil is greater than that consumed by ISO VG32 oil. However, after the onset of turbulence, no measurable difference in power loss is observed.
4. The onset of turbulence is distinctly affected by oil viscosity grade. Thicker oil shifts the transition to a higher sliding speed.
5. In some cases, offset pad maximum temperature is higher than that for the centre-pivot pad. This is in contrast to T75/75 temperature, which is much lower for the offset pads.
6. There is no measurable difference between 60 per cent offset and centre-pivot pad power loss.
7. Pad temperature patterns are similar for both ISO VG32 and ISO VG68 oils. However, the pad temperature pattern is significantly affected by pivot position.

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