IX CONGRESSO BRASILEIRO DE ENGENHARIA E CIÊNCIAS TÉRMICAS

9th BRAZILIAN CONGRESS OF THERMAL ENGINEERING AND SCIENCES

Paper CIT02-0420

Experimental Analysis of Thermal Effects on Tilting Pads Hydrodynamic Thrust Bearings - On the Search for Minimum Power Loss Conditions.

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Abstract. *This paper presents an experimental research on the operating behavior of sector pads hydrodynamic thrust bearings. The most important operating parameters, such as temperature distribution, friction torque and power losses are obtained for a wide range of speed, applied load, lubrication and cooling conditions. Temperature distributions are obtained not only for the pads, but also for the rotating collar. The thrust bearing test-rig and instrumentation is described. Three types of sector pads are employed: central pivoted pads, pads with pivots at 60% and pads with pivots at 66% of mean circumferential length. The main conclusion is that bearing friction losses and operating temperatures are about 10% to 13% lower for the 60% and 66% pivoted pads, as compared to the standard central pivoted pads.*

Keywords. *hydrodynamic thrust bearing, pivoted pad, test-rig steady state temperature distributions, friction torque, power loss*

1. Introduction

Extensive literature reviews on theoretical models and experimental works on the behavior of hydrodynamic thrust bearings are given by El-Saie & Fenner (1988) and Glavatskikh (2001), respectively. Comparison between theory and experiments is presented by Almqvist et all (2000) who also presents an interesting review on the literature.

Experimental works are important not only for analyzing bearing reliability under various conditions of load, speed and lubrication, but also to set up some parameters necessary to accurately predict the bearing behavior, through computer models. For example, some boundary conditions assumed by Huebner (1974) and followers are open to question and this is also the case of many classical and recent computer models described in literature.

From the standpoints above, various experimental works have been carried out, most of them dealing with horizontal shaft, flooded thrust bearings with center-pivoted pads. Gregory (1974) measured power losses and pad (shoe) under babbitt surface temperatures of a double six tilting pad thrust bearing with 133.55 mm inner diameter and 266.7 mm outer diameter, pad arc length 51° and a total bearing surface area of 35548 mm^2. Speeds varied from 4000 to 11000 rpm and loads (unit pressure) from 0.7 to 2.1 MPa, oil film thickness being on the order of 25 µm and 45 µm, respectively, for the loaded and the unloaded side of the double thrust bearing. Power losses were computed by Gregory through an energy balance method, whereby power loss (hot oil carry over) is a direct function of measured oil temperature rise, measured oil flow rate and lubricant specific heat. Other means of heat transfer, such as conduction to the machine base, heat convection and radiation, were not taken into account. An ISO VG 32 mineral oil was used and, for a speed of 10000 rpm and "load" of 0.7 MPa the computed power losses were equal to 112 and 150 kW, for oil flow rates of 102 and 205 l/min, respectively. Conversely, average pad temperatures were equal to 97ºC and 90ºC for the oil flow rates of 102 and 205 l/min, respectively. As commented by Gregory, during test runs at 8000 rpm and under certain oil flow conditions, pad surface temperature at the mean diameter rouse from 70ºC, near the leading edge, to about 140ºC, near the trailing edge, oil being supplied at 46ºC for all tests. Although Gregory's data on power losses were not directly measured, they show how significantly power losses increase with increasing lubricant flow rate.

In a further work, Gregory (1979) showed that power losses may change by as much as 150%, when oil flow rate through the bearing is changed.

Mikula (1987) showed that increasing by 25% the oil supply temperature leads to about 10% reduction in power losses in a high speed horizontal shaft flooded thrust bearing with centrally pivoted sector pads.

Dadouch et all. (2000) obtained experimental pad surface temperatures, oil film thickness and pressure distribution for an eight fixed pad vertical shaft small thrust bearing, with loads and speeds up to 8 kN and 2600 rpm respectively. At top speed and load, the temperature difference between the highest and lowest pad surface temperature was about 8ºC. Friction torque was not measured.

Yuan et all. (1999) presented the description of a laboratory test rig for measurement of oil film thickness and distribution of both temperature and pressure on the surfaces of two shoes (pads) of a 12 spring supported sector pads, thrust bearing with inner and outer diameters equal to 711 mm and 1168 mm, respectively. For the highest condition of speed and load, 500 rpm and 4 MPa, respectively, a top temperature of about 100ºC at the mean radius and about 88ºC near the inner radius of a shoe were recorded.

El-Saie and Fenner (1988) presented a combined theoretical/experimental analysis of a 39.5 mm inner diameter and 74.5 mm outer diameter thrust bearing with 8 center pivot (line support) pads. For a thrust load ranging from 200 to 8000 N and a speed of 3000 rpm, some indication is given that for loads lower than 2000 N, the heat conducted to the collar is almost equal to that conducted to the pads; about 70% of the total power losses being convected to the oil. For higher loads the collar conduction of heat is almost twice pads conduction and only 50% of the total losses are convected to the oil. Experimentally, for a thrust load of 22 kN, at 3000 rpm, five thermocouples equally spaced about the mean diameter indicated temperatures about 78ºC, 88ºC and 85ºC, near the leading edge, at 80% of pad circumferential length and near the trailing edge, respectively.

Glavastskikh (2001) presented a wide range of experimental data obtained from a horizontal shaft flooded thrust bearing with six steel-backed babbitt-faced pads pivoted at 60% of mean circumferential length, the inner and outer diameters being equal to 114.3 mm and 228.6mm, respectively. An ISO VG 46 mineral oil supplied at 30°C, 40°C and 60°C with a constant 15 l /min. flow rate was employed. About 30% reduction in power loss was observed when the oil supply temperature was increased from 30°C to 60°C. It was also concluded that power losses are more significantly affected by shaft speed variation rather than by changing the applied load. Ten thermocouples were conveniently located at about 3.0 mm bellow the pad working surface, in order to obtain the temperature distribution. Axial temperature gradient was not evaluated. For a "load" of 2.0 MPa, rotational speed of 1500 rpm and oil supply temperature equal to 40°C, Glavatskikh obtained a power loss equal to 3.1 kW and recorded temperatures on the order of 53°C, 63°C and 67°C, at about 10%, 50% and 90% of pad arc length, respectively. For a rotational speed of 3000 rpm, the above temperatures were about 57°C, 73°C and 85°C, respectively, whilst power loss was about 7.5 kW. In a similar way, under-working-face temperatures of the collar were measured at 25% and 75% of the radial length and resulted respectively equal to about 60°C and 60.5°C. When speed was increased to 3000 rpm the above temperatures increased up to 73°C and 75°C respectively, for the 25% and 75% radial positions. Axial temperature gradient was not measured for the pads, neither for the collar.

As can be seen from the review of the relevant literature, there is a paucity of experimental data on vertical shaft hydrodynamic thrust bearings with offset pivoted pads.

Theoretically Salles et all (1999) concluded that power loss is lower for thrust bearings with lower number of pads and, power loss is minimum for pads pivoted at about 66% of arc length.

The main goal of the experimental work described in this paper is to find out the working conditions and pivot position that leads to minimum power losses in a sector pads hydrodynamic thrust bearing. Three sets of sector pads were investigated and compared. Radial, circumferential and axial temperature gradients are obtained not only for the pads, but also for the rotating collar. Working with less oil flow rate results in lower power losses and this is also investigated.

2. Descripion of the Experimental Apparatus and Instrumentation

2.1 The Test Bearing

The Kingsbury tilting pad thrust bearing under test is composed basically by a set of six steel/babbitt sector pads positioned on top of six upper leveling plates supported by six lower leveling plates held in a base ring carrier, as shown in Fig. (1). Each pad is loosely constrained, so free pivoting can occur on both the circumferential and radial directions. When subjected to the hydrodynamic forces of the moving lubricant film, each pad inclines, forming a converging flow channel between the top surface of the pad and the bottom lapped surface of the rotating collar, in the circumferential direction of rotation. Pressure is generated as the lubricant is carried through this channel by adhesion to the lapped flat face of the rotating collar. Thrust load is transferred through the oil film, from the collar to the pads.

Three types of pads were employed: a set of centrally pivoted pads and two sets of pads having the pivots positioned respectively at 60% and 66% of the pad mean circumferential length. A set of 60% pivoted pads is shown in Fig. (2), including the main dimensions, which are basically the same, except the 5° angle, which is equal to 8° for the 66% pivoted pads and, obviously 0° for the centrally pivoted pads.

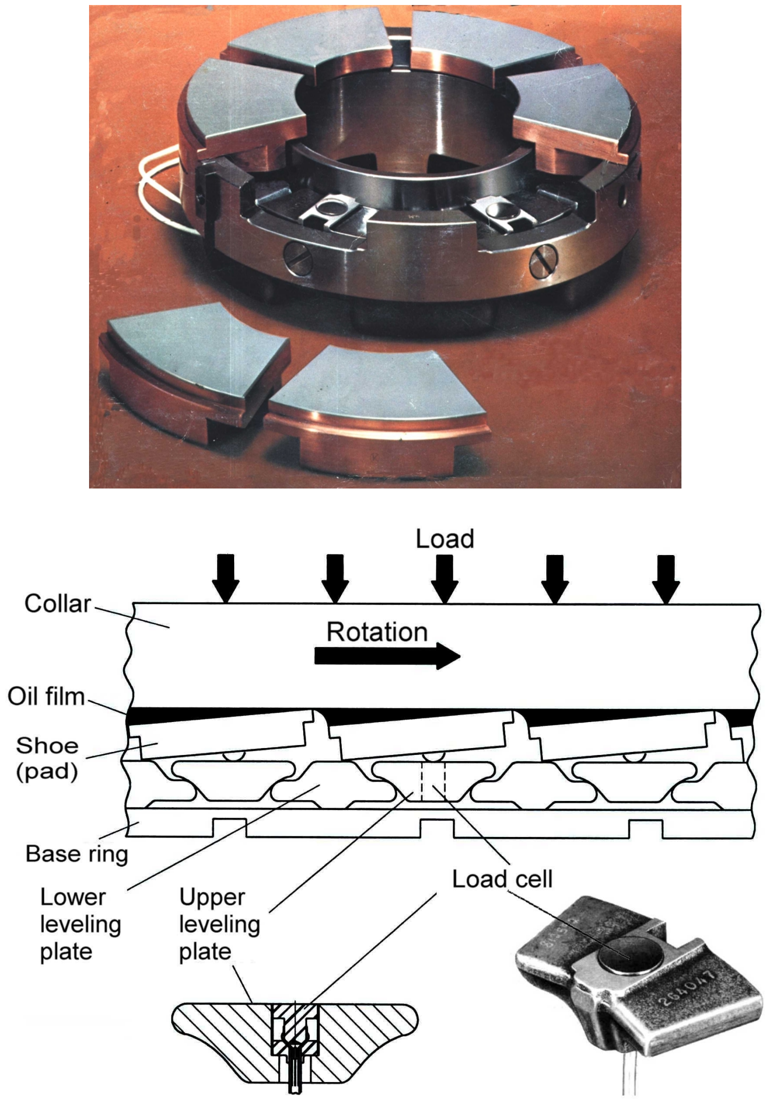

Figure 1. Working principle of the tilting pad thrust bearing

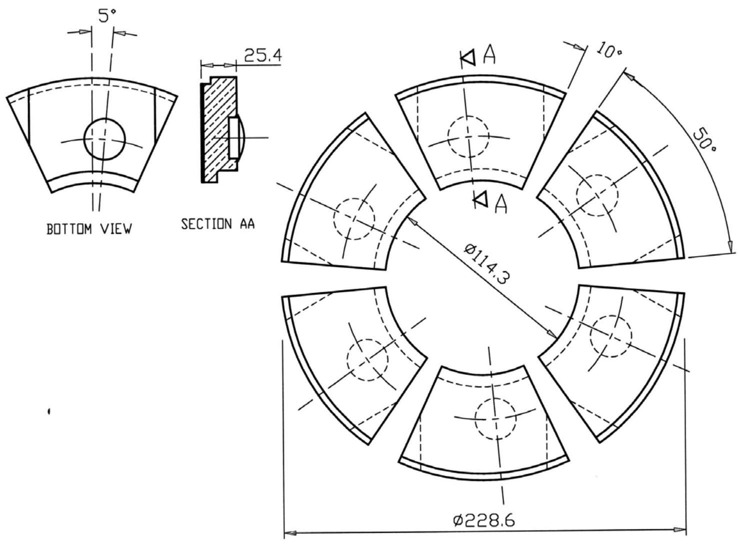

Figure 2. Thrust bearing with 6 shoes 60% offset pivot.

2.2 The Test-Rig

The general arrangement of the test rig is show in Fig. (3). Power is supplied by a 5kW variable speed d.c. electric motor, allowing a step less speed variation from 0 to 3500 rpm. Just below the electric motor, a torquemeter HBM

T10F allows continuos monitoring of torques from 0 to 100 N.m. Right down, a flexible coupling Antares AT50 is used to connect the main shaft/rotating collar to the torquemeter/electric motor.

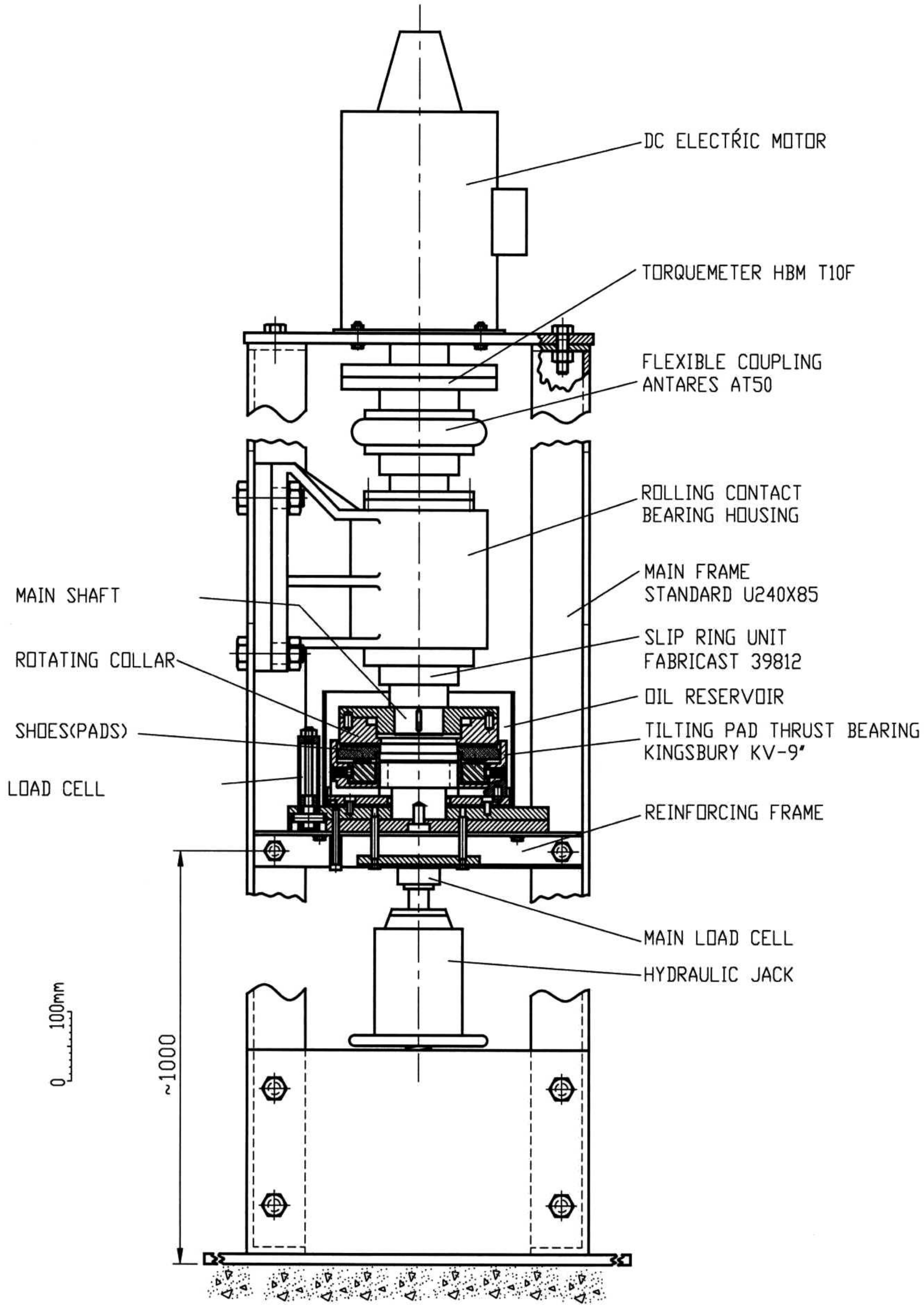

Figure 3. General arrangement of the test-rig.

The test module proper, i.e., the Kingsbury KV9" thrust bearing, rigidly clamped to the oil reservoir, is lubricated by an ISO VG 32 mineral oil supplied at 45°C by means of an hydraulic unit, and is submitted to the upward thrust applied through the hydraulic jack and measured by the central main load cell, on top of the jack.

Reaction to the applied load is provided by the rolling contact bearing (r.c.b) housing seen right bellow the flexible coupling. A 6010Z deep groove ball bearing, on top end of the r.c.b. housing, has the function of a radial guide bearing, whilst a NSK 7313 angular contact ball bearing, on the bottom end of the housing, carries the upward axial load and keeps the main shaft and the rotating collar in a constant vertical position.

Right bellow the r.c.b. housing a Fabricast 39812 silver/silver-graphite slip-ring unit picks up the rotating signals of the collar thermocouples, enabling a continuous monitoring of the corresponding temperatures

2.3 The Measurement Systems and Instrumentation

The main objective of the experimental testing was to monitor/evaluate bearing friction torque and temperature distribution throughout the tilting pad thrust bearing, for a wide range of speed, load and lubrication conditions. This was achieved through a high-speed data acquisition system which logs out the signals from the torquemeter, thermocouples and load cells.

Rotating speed was adjusted by the current converter and accurately measured by a Minipa MDT2244 optical digital tachometer.

The thrust load was applied through the hydraulic jack and measured by the central load cell [01], Fig. (4). The reinforcing frame [05] together with the counterdiscs [08,09] is firmly bolted to the vertical main frame of the test rig, leveled and square to the collar. Three holes equally speed by 120° in the counter disks freely guide the three vertical rods [04] which are screwed to the base disc [10]. Therefore, as can be realized from Fig. (4), the whole assembly consisted by items [02], [03], [04], [10], [11], [15], [12] and [13] plus the pads [14] on top, is free to move vertically up to the rotating collar [16], when the thrust load is applied by action of the hydraulic jack.

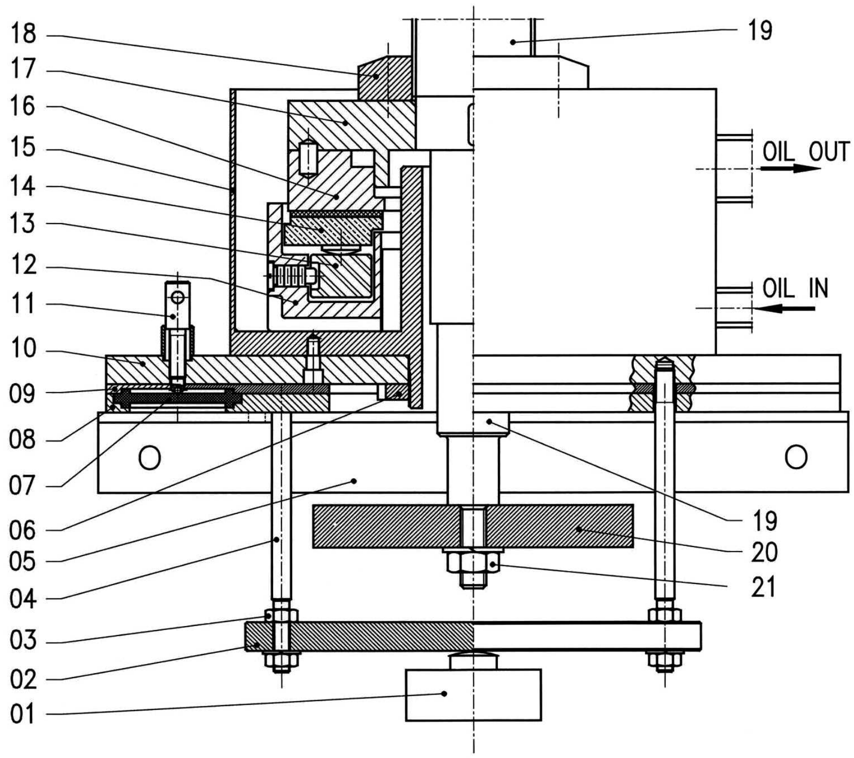

Figure 4. Details of the thrust bearing and loading system.

The rotating collar is rigidly fixed to the main shaft [19] through the disc [17], nut [18], six allen bolt and a square key, the whole unit being kept at a constant vertical position by means of the rolling contact bearing housing, as described before, with the help of Fig. (3)

Referring again to Fig. (4), three strain-gauged discs [07] are clamped between the counterdiscs [08] and [09]. Previously to the thrust load application, the three load transfers [11] are screwed up from the base disc [10]. After application of the desired thrust load by means of the hydraulic jack, the load transfers [11] are screwed down, in the base disc [10], till their spherical ends touch the respective disc [07]. Then, the drain valve of the hydraulic jack may be opened, transferring part or the total thrust load to the three load cells composed by the strain-gauged discs [07] and the load transfers [11]. This procedure allows excellent stiffness to the whole thrust bearing assembly. Furthermore, the upper leveling plate load cell shown in Fig. (1) allows continuous monitoring of the individual load applied to the instrumented tilting pad described bellow. All the load cells were calibrated by using a universal testing machine and showed excellent repeatability and less than 1% hysteresis.

Temperature measurement throughout the tests was accomplished by employing 27 type K (chromel/alumel) thermocouples embedded at several points within the collar, two pads and lubricating oil. Both wires of each thermocouple were passed separately through a twin hole ceramic tube 25 mm long and 2.5 mm diameter to facilitate manufacturing and positioning to the corresponding point within the pad, or the collar, where the temperature has to be measured or monitored. Figure (5) shows a 66% pivoted pad with 12 perforations for insertion of the thermocouples. Similarly, six axial holes were drilled in the collar, for insertion of six thermocouples, allowing the accessment of the collar temperature distribution. In this way, having obtained the steady state temperature of each thermocouple position and, consequently, the radial/axial temperature gradient within both the collar and pads, it is possible to estimate the heat dissipation parcels through these components of the whole bearing assembly. Further, by measuring both the inlet and outlet oil temperatures, the amount of heat carried out by the circulating oil can be determined, enabling one, therefore, to estimate the percentage of heat (friction power losses) being dissipated through the bearing components/lubricating oil.

Figure 5. A 66% pivoted pad, prepared for thermocouple insertions.

Before insertion of the thermocouples, the pads and collar perforations were filled in with a silicone base heat sink compound which presents the advantages of having simultaneously good thermal conductivity and electrical insulation, avoiding, therefore, any interference between two or more neighbouring thermocouples.

The stationary thermocouple wires were directly connected to the data acquisition system. On the other hand, the rotating thermocouples were connected to the silver rings of the slip-ring unit and then, through the silver graphite brushes, to the data acquisition system. A slight air cooling to the ring/brush contacts was used, as indicated in books and papers on temperature measurement.

For calibration, both types of thermocouples were immersed together with a certified PT 100 thermistor in a glass tube charged with mineral oil, the whole assembly being placed in a beaker of oil that was then heated up to 150^{o}C. On cooling down, in steps of 10^{o}C, the outputs from the thermocouples/data acquisition system were as follows: - 0.1^{o}C for the directly connected thermocouples and +0.4^{o}C for the thermocouples connected to the slip-ring unit and then to the data acquisition system.

Bearing friction torque was measured through the HBM T10F torquemeter, which access the total torque from both the tilting pad thrust bearing and the two rolling contact bearing in the housing below the flexible coupling. Therefore the later torque has to be subtracted from the first. This was done by using plenty of formula and tables given in literature. Calibration of the torquemeter was carried out throught a system consisted essentially by a torque-arm clamped to the torquemeter upper flange, a flexible rope fixed to the arm outer end and passing over a deep groove ball bearing positioned on a protruding pin in the test-rig main frame as shown in Fig. (6) The lower flange of the torquemeter was prevented to turn, while the load was applied in increments and decrements at the vertical end of the rope and the corresponding outputs were recorded. Excellent repeatability/linearity was observed, in total agreement with the information given in the torquemeter manufacturer catalogue, i.e. a 1.0 mV output results from a 10 N.m applied torque. Further, insignificant hysteresis, lower than 1% was observed during the calibration procedure.

Oil flow rate was regulated and measured through a unidirectional valve installed immediately after the oil filter of the hydraulic unit. Complementary, an orifice type, variable reluctance, linearized, bi-directional flow meter connected to a Nashua Model 736 Read-out unit allows oil flow measurement ranging from 1.8 to 75 l/min. Calibration was carried out by using a high precision chronometer to measure the time taken to fill 4 liters of oil in a graduated recipient, for different opening positions of the regulator valve and, simultaneously, the reading in the Model 736 Read-out unit.

Figure 6 – Torquemeter Calibration

Oil flow rate was regulated and measured through a unidirectional valve installed immediately after the oil filter of the hydraulic unit. Complementary, an orifice type, variable reluctance, linearized, bi-directional flow meter connected to a Nashua Model 736 Read-out unit allows oil flow measurement ranging from 1.8 to 75 l/min. Calibration was carried out by using a high precision chronometer to measure the time taken to fill 4 liters of oil in a graduated recipient, for different opening positions of the regulator valve and, simultaneously, the reading in the Model 736 Read-out unit.

3 Results

The experimental results were obtained for three tilting pad bearings consisted by six sector pads with pivots located respectively at 50%, 60% and 66% of the pad mean length, as described before.

Rotational speeds varied from 500 to 3500 rpm, applied load from 12 to 24 kN and supplied oil flow rate from 1.7 to 4.5 l/min; an ISO VG 32 mineral oil, with viscosities of 27.2 mPa.s at 40 ^{0}C and 4.6 mPa.s at 100 ^{0}C was employed and supplied at 45 ^{0}C in most of the tests.

The presentation and discussion of the results is made on the basis of Fig. (7), which shows the thermocouple positions (dimensions in mm) within the collar and a pad (shoe). These positions are indicated by C1, C2, C3, C4, C5 and C6 for the collar and by S4 to S16 for the pad. Positions 1,2,3 and 9 are indicated without the prefix S in cross section AA, due to lack of space. In fact, two pads were instrumented, such that thermocouples S14, S15 and S16 were positioned in the second pad and measure the oil temperatures at the leading edge, outer radius and the trailing edge of the pad, respectively. Two additional thermocouples in positions S5 and S13 are repeated in both pads, enabling a comparison among the corresponding temperatures.

In Tab. (1), the five thermocouples positioned in the second pad are denoted by an asterisc (*). Table (1) shows the steady state temperatures for a 12 kN thrust load and at 1500 rpm rotational speed, for the 6 positions within the collar and 18 position within the two pads for the there thrust bearings, i.e., with centrally pivoted pads, pads with pivots at 60% and 66% of the circumferential length. Three additional thermocouples give the ambient temperature, Ta, the oil supply temperature, Ti, and the outlet oil temperatures, To. Steady state friction torques are also given in Tab. (1) for the three bearings.

It can be concluded from Tab. (1) an also from Figs. (8) and (9) that, the steady state temperatures and friction torques were about 10% and 13% lower for the 66% and the 66% pivoted pads, respectively, as compared to the bearing with centrally pivoted pads.

It may also be seen from Tab. (1) that the temperatures corresponding to positions S6 and S12 were, respectively, the maximum and minimum pad temperatures. Further, it can be noted that the temperatures corresponding to positions S4, S5, S6 and S9 are essentially the same, for the three bearings.

Another evidence from Tab. (1) is that, for the bearing consisted by six centrally pivoted pads, the maximum temperature occurs in position S6, the so called 75/75 position (at 75% of the pad radial length and at 75% of the pad arc length, in the direction of rotation), as had also been observed by Gregory (1974). However, for the two offset pivoted pads, the temperatures S4 and S5 were slightly higher than the temperature S6.

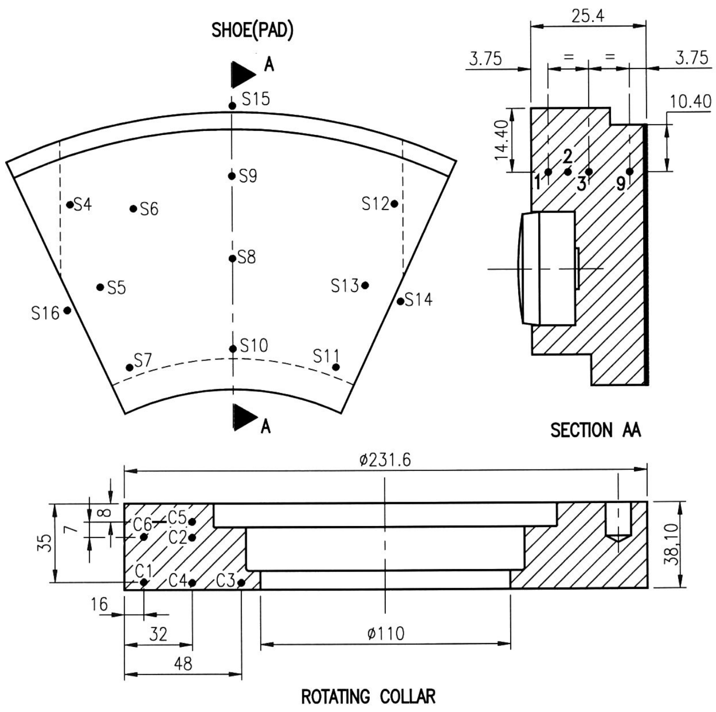

Figure 7. Thermocouple positions within the pads and collar.

From the collar and pad temperature gradients given in Tab. (1) and by applying the basic conduction heat transfer equations, it may be concluded that about 23% and 17% of the bearing friction power loss are transferred to the collar and to the pads, respectively, in general agreement with the distribution proposed by El-Saie & Fenner (1988).

Table 1. Temperature distributions within the pads and collar, for the pads with three different pivot positions.

Rotation: 1500 rpm Axial Load: 12 kN				Pivot position: 50% 60% 66% Friction Torque: (N.m) 8.14 7.14 6.94							
Temperatures °C											
	Pivot position				Pivot position				Pivot position		
Pos.	50%	60%	66%	Pos.	50%	60%	66%	Pos.	50%	60%	66%
S1	75.1	-	66.1	S2	76.4	-	-	S3	76.5	-	67.2
S4	74.5	71.5	68.6	S5	79.6	71.3	69.1	S7	76.1	68.8	67.7
S8	79.8	69.8	66.6	S9	79.6	70.2	66.4	S10	77.1	68.0	66.0
S11	73.9	66.3	64.2	S12	73.8	64.6	61.8	S13	-	65.4	63.1
S14*	68.7	63.7	61.9	S15*	68.9	63.3	61.8	S16*	69.3	66.8	66.7
C2	81.8	72.3	69.7	C4	84.5	75.5	71.6	C5	80.0	71.7	68.6
C1	79.3	71.1	68.9	C3	83.4	74.3	68.4	C6	78.3	70.2	67.7
Ta	24.0	21.8	24.2	Ti	45.0	45.1	45.4	To	63.7	57.3	56.0
S6	79.9	71.4	68.5	S5*	78.4	69.7	68.9	S13*	72.3	65.2	63.4

* thermocouples embedded in a second pad

T_a = ambient air temperature; T_i = inlet oil temperature; T_o = outlet oil temperature

It was a glad surprise to realize that the sector pads of the horizontal shaft, double thrust bearing investigated by Glavatskikh (2001) are essentially identical to the 60% pivoted pads of the present work. However, bearing friction torque measured by Glavatskikh was about 2.5 times the friction torque obtained in the present work. This large difference is certainly due to the existence of two sets of pads in Glavatskikh bearing and also to its flooded lubrication system. Conversely, Glavatskikh bearing operating temperatures were about 10% lower. Surface temperature distribution observed from Tab. (1) is more uniform than Glavatskikh data, and this may be advantageous as far as thermal distortions are concerned. Glavatskikh also measured under surface temperatures at two radial positions in the collar, respectively at 25% and 75% of the radial working length, and obtained an insignificant difference between these temperatures. However, in the present work, collar surface temperatures were measured also at the mean radius, position C4, and, as can be seen from Tab. (1), this temperature is on the order of 10% higher than the temperatures C1 and C3. It can also be concluded from Tab. (1) that the maximum collar temperature, at position C4, is about 10% higher than the pad maximum temperature.

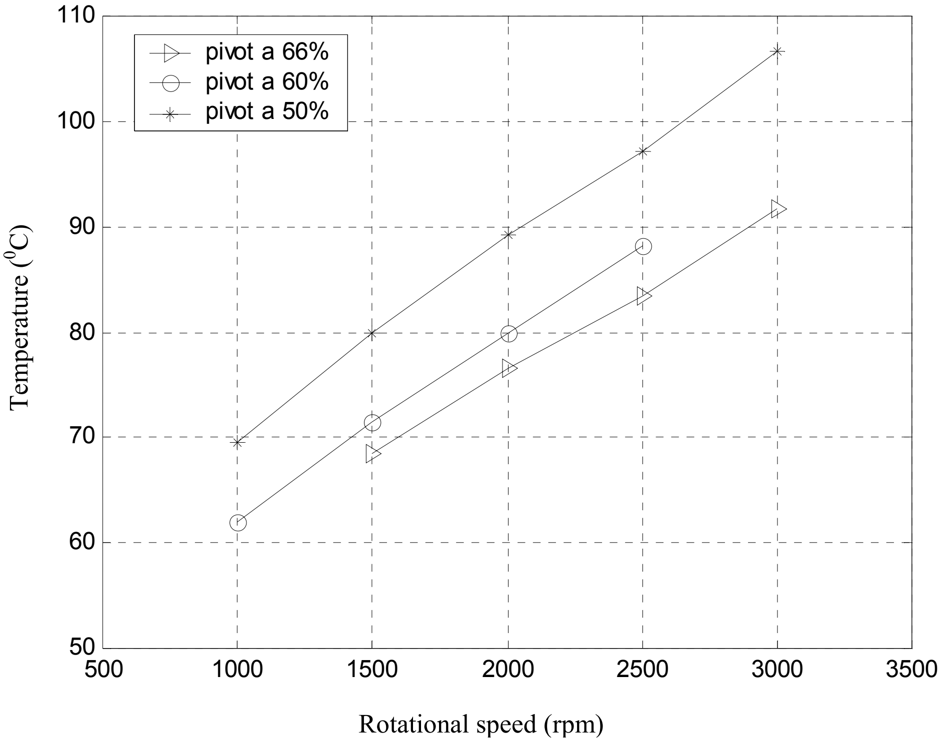

Figure 8. Variation of temperature S6 with speed, for a thrust load of 12 kN.

Figures (8) and (9) show temperatures and friction torque versus rotational speed for the three types of pads. It can be seen that the temperatures increase almost linearly with speed, while friction torque increases less significantly up to a 3000 rpm rotational speed, and may tend to present a constant value for higher speeds.

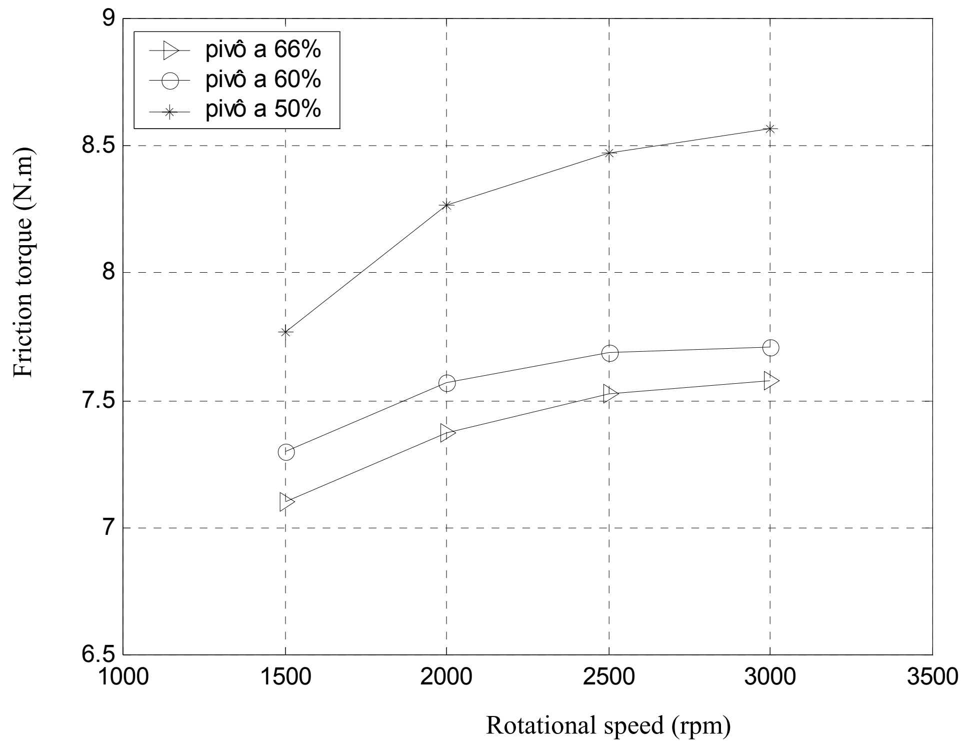

Figure 9. Variation of torque with rotational speed for a 18 kN thrust load and for a supplied oil flow rate of 1.9 l/min.

Figure (10) show the variation of the temperatures S6 and S1 and also the friction torque with applied load for the bearing consisted by 66% offset pivoted pad. It can be seen that the axial temperature gradient (S6-S1) increase almost linearly with thrust load and the friction torque shows a similar trend.

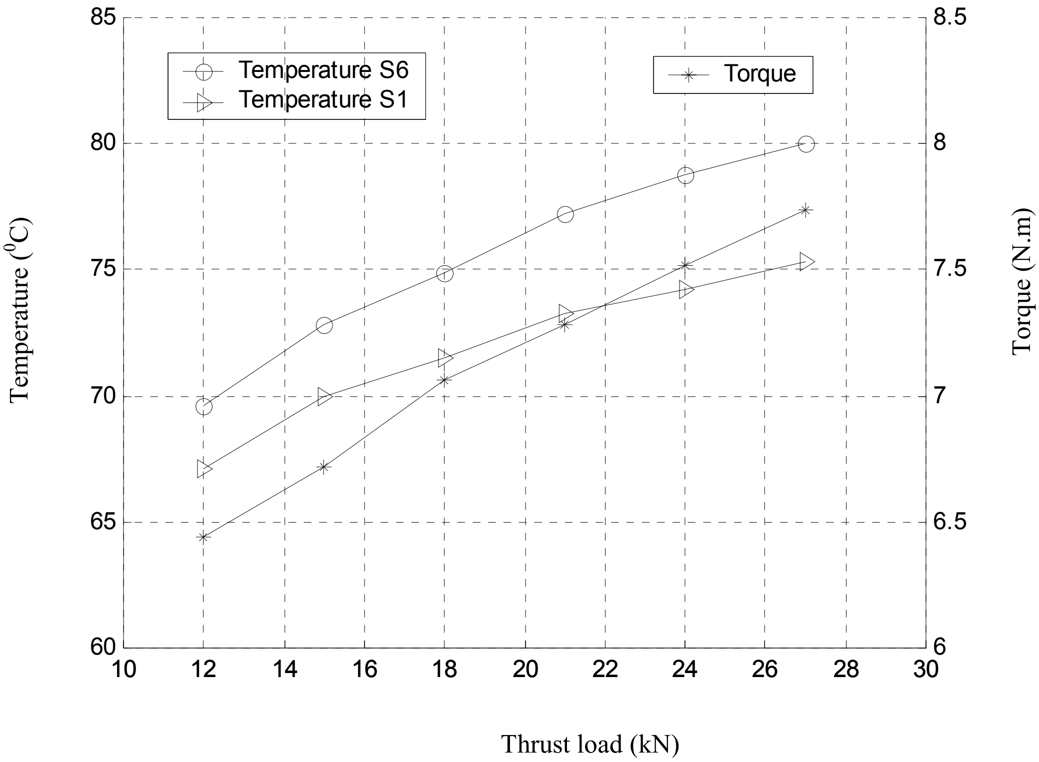

Figure 10. Variation of temperatures and torque with thrust load for a 1.8 l/min supplied oil flow rate and at 1500 rpm rotational speed.

Figure (11) shows, for the 66% offset pivoted pads, the variation of temperatures S6 and S12 and the friction torque with the oil flow rate supplied to the bearing. As expected, friction torque increases significantly with oil flow rate,

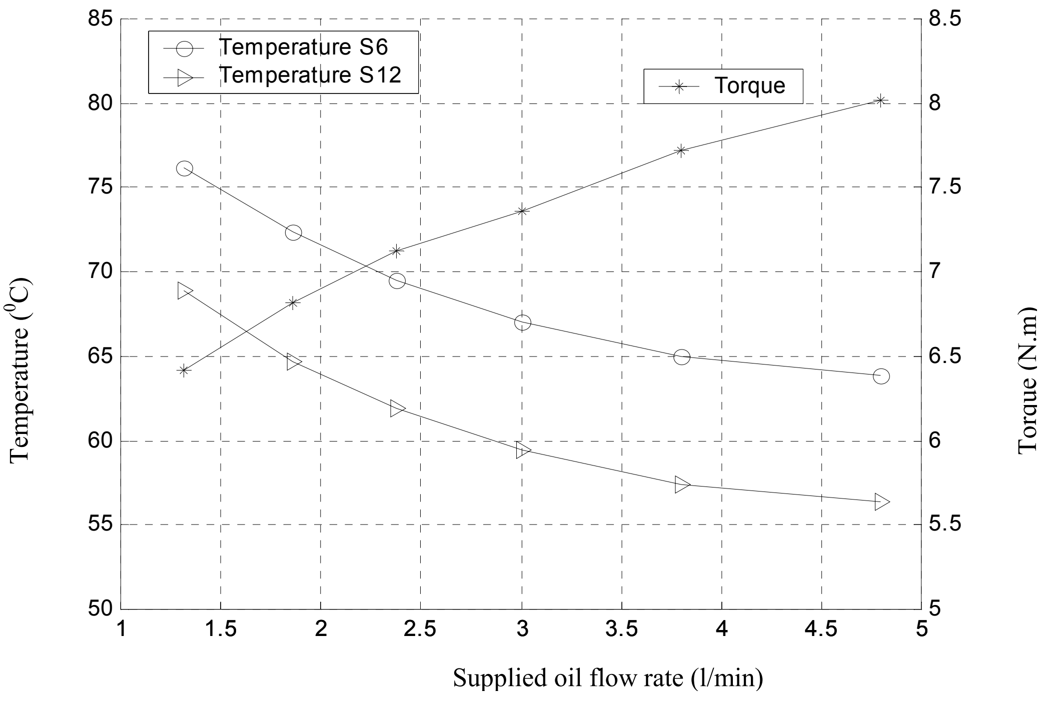

Figure 11. Variation of temperatures and torque with supplied oil flow rate, for a 15 kN thrust load and at 1500 rpm rotational speed.

while the temperatures decrease. It can be observed that the temperature gradient (S6-S12) remains almost constant as the oil flow rate is increased. Apparently, the steady state temperatures tend to a minimum value for an oil flow rate about 5.0 to 5.5 l/min. Increasing the supplied oil flow rate, will probably continue to increase the bearing operating torque, without lowering the steady state temperatures. Therefore, for the operating conditions indicated in Fig. (11), the supplied oil flow rate of about 5.0 l/min may be considered as the ideal value from the standpoint of minimum operating temperature condition. Unfortunately, it was not possible to carry out tests with oil flow rates higher than 4.8 l/min., due to oil leakage problems in the test-rig.

4. Conclusions and Suggestions for Further Work

The most important conclusion from the experimental data obtained from the test apparatus described in this paper is regarded to the pad pivot position. It is clear from Tab. (1) and Figs. (8) and (9) that the bearing with pads pivoted at 66% of the pad circumferential length, in the direction of rotation, operates with lower temperatures and friction torque. However, one must keep in mind that rotation cannot be inverted for a bearing with offset pivoted pads and, in this respect, centrally pivoted pads are appropriated for any machine that operates in either direction of rotation.

Another important aspect shown in the paper is the significant effect of decreasing the power loss in the bearing, by lowering the supplied oil flow rate, or increasing the oil supply temperature. However, this procedure brings up higher operating temperatures. It must be emphasized that most of the experimental data presented in this paper were obtained for low oil flow rate supplied to the bearing, with consequent lower operating torques and higher temperatures than other data available in literature.

Further work in the test -rig is unlimited, the following aspects being envisaged:

- Based on the trend shown in Fig. (11), the ideal oil flow rate supply to the bearing that corresponds to minimal operating temperatures can be obtained, for the ranges of rotational speeds and applied load,
- Different lubricants and lubrication systems may be employed,
- Different bearing sizes and materials may be investigated,
- A wider range of speed and thrust load may be considered, by introducing some modifications to the test-rig.

4. Acknowledgement

The authors gratefully acknowledge the financial support from FAPEMIG – Fundação de Amparo à Pesquisa do Estado de Minas Gerais, Proc. TEC 855/97 and technical and financial support from Kingsbury Inc.

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