Tribology International 54 (2012) 1–8

Contents lists available at SciVerse ScienceDirect

Tribology International

journal homepage: www.elsevier.com/locate/triboint

Experimental comparison of the performance of a journal bearing with a single and a twin axial groove configuration

F.P. Brito [a,*], A.S. Miranda [a], J.C.P. Claro [a], M. Fillon [b]

[a] Centre for Mechanical and Materials Technologies (CT2M), Universidade do Minho, Campus de Azurém, 4800-058 Guimarães, Portugal
[b] Institut Pprime, CNRS - Université de Poitiers–ENSMA, UPR 3346, Dépt Génie Mécanique et Systèmes Complexes, SP2MI, 11 Boulevard Marie et Pierre Curie, BP 30179, F86962 Futuroscope Chasseneuil Cedex, France

ARTICLE INFO

Article history:
Received 28 October 2011
Received in revised form
20 April 2012
Accepted 24 April 2012
Available online 5 May 2012

Keywords:
Hydrodynamic journal bearings
Feeding grooves
Lubricant feeding conditions
Journal bearing performance

ABSTRACT

It is often assumed that a journal bearing will operate more effectively with a twin groove arrangement than a single groove one, but little evidence has been presented in support of this.

An experimental assessment of a journal bearing with either one or two axial grooves located perpendicularly to the load line was performed.

It was found that under heavy loaded operation the twin groove configuration might actually deteriorate the bearing performance when compared with the single groove arrangement, namely due to uneven lubricant feed through each groove. It is concluded that the knowledge of the feed flow rates through each groove can be used to improve bearing performance under specific regimes by implementing groove deactivation or flow balancing strategies.

1. Introduction

Hydrodynamic journal bearings are widely used in turbomachinery and other equipments subjected to high specific loads and shaft speeds. The lubricant is often supplied at a prescribed pressure through one or two axial grooves. When using single groove journal bearings, it is usual to locate the groove at 90° to the load line, upstream of the position of minimum film thickness. In fact, works such as [1] found that this bearing configuration outperforms the bearing with a groove located along the load line (crown bearing). Twin groove journal bearings are also widely used, especially when the shaft is expected to assume both directions of rotation. In this bearing configuration the two grooves are diametrically opposed and located at $\pm 90^\circ$ to the load line, upstream and downstream of the location of minimum film thickness. In this work these grooves are from now on designated simply as the upstream and the downstream groove, respectively.

Experimental works concerning the twin groove bearing geometry have been reported in the literature but normally with little attention to the lubricant feed conditions such as feed pressure, feed temperature and the resulting lubricant flow rate supplied through each groove. The analysis of these parameters, which significantly affect bearing behavior, would certainly shed additional light over the phenomena taking place within the bearing gap.

In fact, [2,3] carried out a valuable experimental work on these bearings, but with fixed supply conditions. The same can be said of the work by [4], which lacks some other important data such as shaft eccentricity. Finally, [5,6] presented experimental results for this bearing type but omitted the main lubricant properties. Furthermore, the bearing they tested possessed unusually large grooves, spanning 55° each. There is a lack of recent experimental works which assess the influence of feed conditions for this bearing geometry. That is why the authors started assessing the influence of lubricant feed conditions on twin groove journal bearing performance such as lubricant feed pressure and temperature and found that these parameters significantly affect bearing behavior [7,8].

Nevertheless, they found that the lack of knowledge concerning the flow rate partition among the two grooves was limiting the discussion of the results and additional insight into the role of each groove in journal bearing performance was needed.

It would seem commonsense to assume that a twin groove arrangement would provide a more effective lubrication and cooler operation than a bearing with a single groove located at 90° to the load line, upstream of the position of minimum film thickness, but little evidence has been found in support of this. In the particular case of twin groove journal bearings, the study of the individual flow rates in each groove has been almost always disregarded, with the exception, to the authors' knowledge, of a

* Corresponding author. Tel.: +351 253 612 231.
E-mail addresses: francisco@dem.uminho.pt (F.P. Brito), asm@dem.uminho.pt (A.S. Miranda), jcpclaro@dem.uminho.pt (J.C.P. Claro), michel.fillon@univ-poitiers.fr (M. Fillon).

0301-679X/$ - see front matter © 2012 Elsevier Ltd. All rights reserved.
http://dx.doi.org/10.1016/j.triboint.2012.04.026

conference paper by Basri and Neal [9], where they offered a limited set of results which included the separate measurement of the flow rate at each groove. Furthermore, there is no experimental work, to the authors' knowledge, that compares the performance of single and twin axial groove bearings with identical geometry. The lack of information concerning flow rate distribution between grooves seems to be highly limitative for the understanding of the role of the lubricant in bearing cooling and lubrication effectiveness.

In the present work, an experimental comparison between a 50 mm journal bearing with two diametrically opposed grooves located at $\pm 90^{\circ}$ to the load line and a bearing with only one of these grooves (the one located upstream of the position of minimum film thickness) has been undertaken. The comparison was made by using the same bearing for both groove configurations shutting the oil feeding to the downstream groove with a valve in one of the cases, in order to emulate a single groove bearing (this option is further justified in the Experimental Procedure section). The specific load and the feed pressure, which greatly influence the flow rate, have been allowed to vary within a broad range in order to provide additional insight into the subject, while the measurement of each groove flow rate was also undertaken.

The present study might be of special relevance for bearing designers, as some counter-intuitive flow patterns might occur when adding an extra groove to a single groove bearing. As it will be seen, under certain conditions this addition might inclusively be deleterious for bearing performance in ways not normally predicted by many bearing performance prediction tools. In fact, an accurate prediction of the thermohydrodynamic behavior of journal bearings calls for a realistic treatment of the actual lubricant feed conditions, bearing geometry (namely real groove dimensions), film rupture and reformation [1,10].

2. Experimental procedure

The experimental test rig used was the one existing in the Tribology Laboratory of the University of Minho. Although this apparatus has already been used in several works [10,11], it has been significantly improved for the present work in areas such as data acquisition, signal treatment, flow rate measurement and feed temperature regulation.

The rig allows the regulation of rotational speed, load, oil feed pressure and feed temperature. The measured performance parameters were the temperature at the oil-bush interface, the oil outlet temperature, the oil flow rate at each groove, the total oil flow rate, and the shaft locus.

In order to obtain and compare a single and a twin groove bearing configuration, an uncoated bronze twin groove bearing to which the oil feed may be shut at the downstream groove has been used (see details in Fig. 1). This has been made in order to emulate either a single or a twin groove bearing. It is true that the downstream groove has been deactivated but not eliminated when testing the single groove configuration. Nonetheless, its influence should be marginal since this inactive groove is located at the unloaded region of the bearing. On the other hand, this is the only feasible way of testing and compare two different groove configurations (single and twin) with exactly the same geometric features (namely identical clearance), and consecutively during the same test session. In fact, if two different bearings were to be manufactured to perform the single/twin groove comparison, the slight but inevitable differences in gap geometries between the two bearings would compromise more deeply the validity of the comparison than the groove activation/deactivation procedure used in the present work.

The geometric parameters, operating and supply conditions, as well as lubricant properties, are presented in Table 1. The journal, made of stainless steel, was rigidly mounted between two preloaded conical rolling bearings to ensure a suitable stiffness for the system, as seen in Fig. 2a. The bush diameter, the shaft diameter and cylindricity were measured using a coordinate measuring machine with a resolution of 0.1 μm. The oil used was ISO VG 32 (Galp Hidrolep 32—see Table 1 for details).

The loading arrangement relies on a cantilever system on which dead weights are applied. As seen in Fig. 2b, the cantilever acts upon the bush body through a closed loop steel cable.

Table 1
Main bearing characteristics, lubricant properties and operating conditions.

Parameters		Units	Value/range
Geometrical bearing characteristics			
Bearing diameter (nominal)	d	mm	50
Outer Bush diameter	D	mm	100
Bush length/diameter ratio	b/d		0.8
Groove lenght/bush length ratio	a/b		0.5
Groove width/diameter ratio	w/d		0.2
Bearing diametral clearance (at 20 °C)	C_d	μm	107
Oil properties			ISO VG 32
Dynamic viscosity at 30 °C	μ_{30}	Pa.s	0.0467
Dynamic viscosity at 75 °C	μ_{75}	Pa.s	0.0083
Specific Heat	C_p	J/kgK	1943
Specific mass	ρ	kg/m^3	875
Thermal conductivity	k	W/mK	0.13
Operating conditions			
Rotational speed	N	rpm	3000
Applied load	W	kN	0.4–5
Specific Load	Ws	MPa	0.2–2.5
Supply conditions			
Oil supply pressure	P_f	kPa	70–250
Oil supply temperature	T_f	°C	29.5

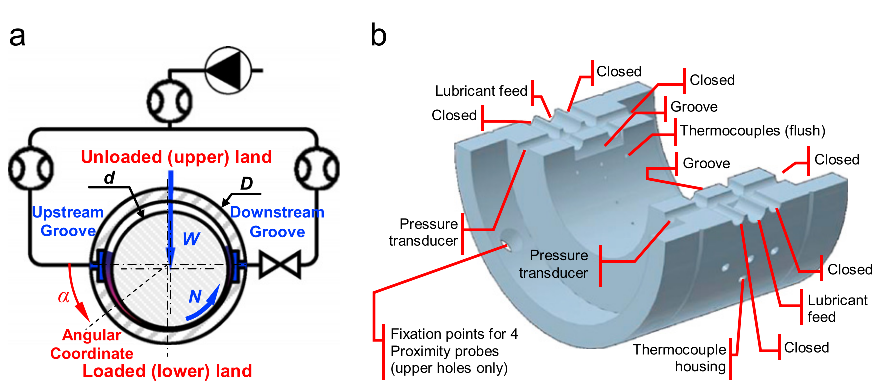

Fig. 1. (a) Schematic overview of the bearing system, including flow meter system and (b) Drawing of the lower half of the bush body.

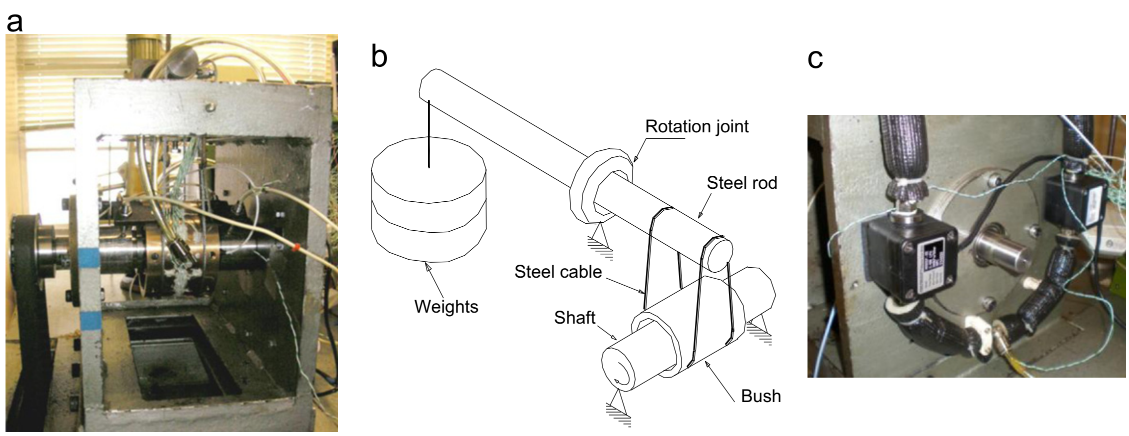

Fig. 2. (a) Detail of the test rig showing the bearing system, (b) outline of the loading system and (c) detail of the gear flow meters used for measuring the flow rate of lubricant feeding each groove, including thermocouple for feed temperature measurement at the main feed pipe.

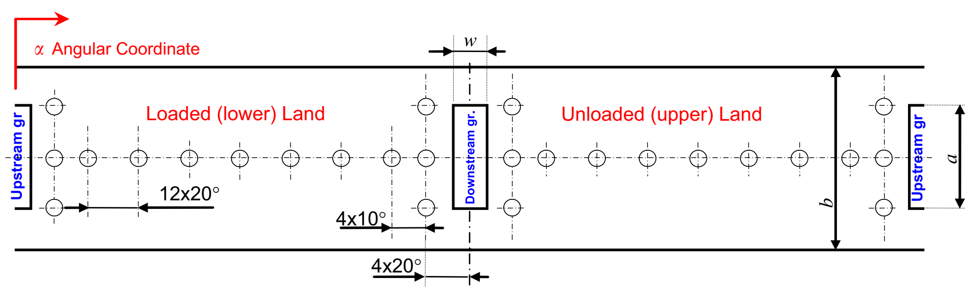

Fig. 3. Angular location of the thermocouples at the inner surface of the bush (unwrapped view).

The loading system was calibrated using a high precision load cell with an error of less than ± 0.5 N. The shaft was driven by a 0.95 kW variable speed motor via a transmission belt. The speed was regulated through an inverter drive and kept within a range of ± 10 rpm of the nominal speed.

The feed pressure (P_f) was regulated by a restrictor valve and monitored by pressure transducers located at the interior of each groove as seen in Fig. 1b. P_f was kept within an interval of variation of ± 4 kPa.

The feed temperature (T_f) was regulated via a thermostatic bath with outer circulation passing through a plate heat exchanger in order to heat the oil that was being supplied to the bearing. The temperature of the bath was regulated so that T_f was kept within a range of $\pm 0.4\,^{\circ}$C from the set point. T_f was monitored by three thermocouples, one located in the main feeding pipe, just upstream of the point where the flow is separated in two branches to feed each groove, as seen in Fig. 2a. Two other thermocouples, one for each groove, were also located at the groove entrance.

The measurements were always performed under steady-state. In order to achieve this, start-up times were set for thermal stabilization. Between tests, parameters such as temperature and flow rate were monitored until stabilization was achieved.

The oil flow rate was measured by three gear flow meters (repeatability 0.03%), suitable for low flow rate measurements, linked to the data acquisition system. As outlined in Fig. 1a, one flow meter was attached to the main feed line, while the other two (seen in Fig. 2c) were positioned along each branch in order to measure partial flow rates. To ensure accurate flow rates, electric pulse measurements were performed during 35 s so that even feeble flow rates could be recorded. The difference between the total flow rate and the sum of the partial flow rates measured was below 1.5% in most of the cases.

The temperature field was monitored through type K thermocouples attached to a data acquisition system. The repeatability of the measured values was within $\pm 1\,^{\circ}$C. The temperature at the oil-bush interface was measured at the locations depicted in Fig. 3. The thermocouples were placed inside fully drilled holes, flush with the inner bush surface as depicted in Fig. 1b. Another set of thermocouples was positioned so as to measure the oil outlet temperature and the environment temperature.

The relative shaft position was obtained with the help of two pairs of Eddy current proximity probes located at $\pm 45^{\circ}$ to the load line, on both sides of the bearing. The system was calibrated, and sensitivities around 7 mV/μm were obtained. The accuracy of the measurements was found to be strongly affected by thermal and elastic deformations of the various components. Estimations were performed in order to compensate for these deformations. The compensation for thermal deformation has already been detailed in [8] for a similar system.

A FEM tool (CosmosWorks) has been used to estimate the deformation of the bush in the loading range tested and its influence on the distance between the tip of the proximity probe and the journal surface. The estimated values were then used to correct the measurements of the proximity probes.

Additionally, some reference tests had to be used, with the locus of the shaft center being estimated in reference to these tests, where the absolute position of the shaft has been obtained from a theoretical model presented by [1] adapted for twin groove bearings.

3. Results and discussion

3.1. Lubricant flow rate

The values for total flow rate and the flow rate through each groove are displayed in Fig. 4 for two different feeding pressures (100 kPa and 250 kPa).

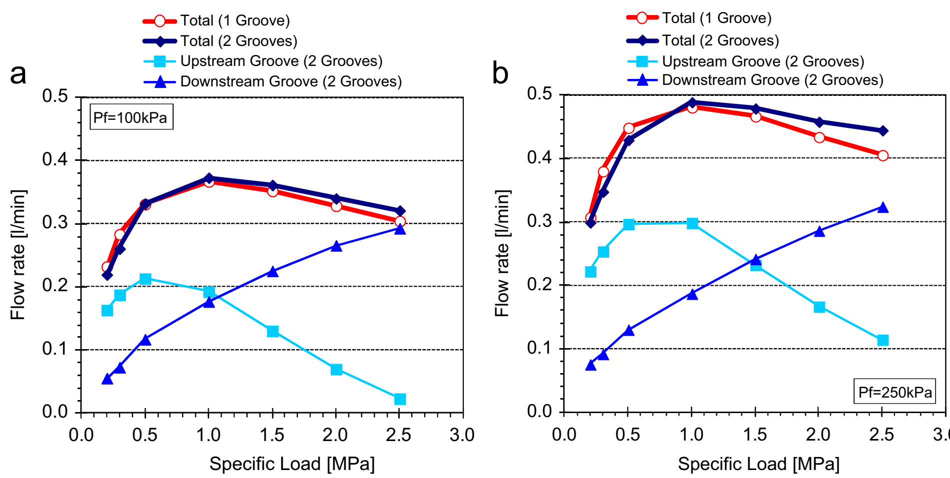

Fig. 4. Influence of the number of grooves on the lubricant flow rates for (a) $P_f = 100$ kPa and (b) $P_f = 250$ kPa.

It is interesting to note that the total flow rate is nearly the same for both the single and the twin groove bearing. Only a slight difference appears as specific load increases, with the single groove bearing displaying slightly lower flow rate values (5% and 9% maximum, respectively for 100 kPa and 250 kPa tests). It is natural that these differences are greater under higher specific loads, because these are the tests where the additional groove (downstream groove) is more active: at high specific loads this groove is located at a very low pressure region, the ruptured film region. Here, the resistance to lubricant entry is lowest, causing a great part of the lubricant to enter the bearing gap through this groove instead of through the groove which feeds the loaded land of the bearing (upstream groove).

It can be concluded that when increasing the number of grooves from 1 to 2 the lubricant supplied directly to the active region of the bearing through the upstream groove decreases, as it is partially deflected to the unloaded land of the bearing through the downstream groove. It can be seen clearly in Fig. 4a that the flow rate behavior of the twin groove bearing can be problematic at high specific loads and low values of P_f: in the highest specific load test displayed in this figure the upstream groove flow rate was nearly zero. This means that the critical region of the bearing (the region of minimum film thickness) is poorly lubricated, even admitting that some of the lubricant coming from the upper land is recirculated. Observing the trend of the upstream groove flow rate curve it may be estimated that negative flow rate would likely occur at specific loads above 3 MPa. These conditions have not been tested, but the aforementioned phenomenon has been observed by the authors for other operating conditions [12–14]. Unlike the twin groove case, the single groove journal bearing still displayed an upstream groove flow rate of 0.3l/min in the 2.5 MPa test, which was only 17% lower than the maximum measured flow rate (1 MPa test).

3.2. Shaft locus

The effect of the number of grooves, specific load and feeding pressure on the shaft eccentricity ratio and attitude angle is displayed in Fig. 5. The increase of feeding pressure has a positive influence in shaft eccentricity ratio, reducing it as much as 0.06 (Fig. 5a). On the other hand, changing the groove configuration from a single to a twin groove arrangement raised the eccentricity for specific loads above 0.5 MPa. This deleterious effect is amplified as specific load increases, with a maximum increase of 0.06, from 0.87 to 0.93 for the highest specific load/lowest P_f tested.

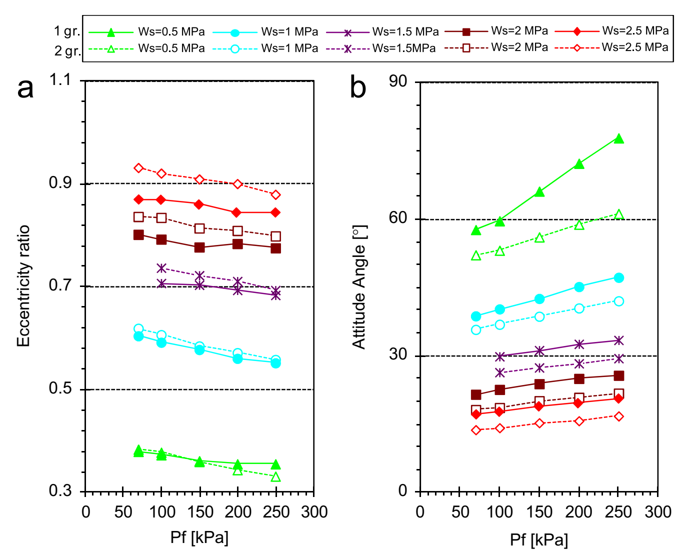

Fig. 5. Influence of the number of grooves on (a) eccentricity ratio and (b) attitude angle, for several specific loads and feeding pressures.

Although the absolute values of eccentricity are to be taken mainly as qualitative since they are prone to a high degree of uncertainty, the relative differences between tests, especially between single and twin groove tests carried out at similar operative conditions, may be analyzed with some confidence as the switch between single and twin groove configuration was made consecutively for each test and under operation.

The eccentricity increase due to the increase in number of grooves and the decrease in P_f was always accompanied by a decrease in attitude angle, as seen in Fig. 5b. The variation in attitude angle is especially strong under light loading. This could be related to the balancing of the hydrostatic effect provided by each groove pocket. Whereas for single groove bearing the feed pressure at the upstream groove will push the journal towards the opposite side (increasing the attitude angle), for a twin groove bearing this effect will be minimized. It can be seen that this hydrostatic effect is increased when P_f increases. This should happen mainly for lightly loaded bearings because the hydrostatic lift provided by the groove pockets is not negligible when compared with the hydrodynamic lift.

3.3. Temperature field

The influence of the number of grooves on the temperature profile at the midplane of the inner bush surface is presented in Figs. 6 and 7 for two different specific loads and feeding pressures. The added cooling effect provided by the increase in P_f (and consequently in flow rate) does not seem to affect significantly the temperature drop across grooves but induces a uniform decrease of the temperature level of the inner bush surface. One main reason for this should be linked with the heat transfer process. In fact, the high conductivity of the bronze bush tends to smooth the temperature differences, so that the cooling effect of the fresh oil entering the bearing gap tends to be quite evenly distributed.

To analyze the influence of the number of grooves, it seems useful to perform a separate analysis for low specific load (Ws=0.2 MPa, Fig. 6) and high specific load (Ws=2.5 MPa, Fig. 7).

Lightly loaded bearing (Ws=0.2 MPa)—Fig. 6:

– Under low loads there is little difference between the temperature trends of the single and twin groove configurations. Only a slight temperature fall across the downstream groove was observed in the twin groove bearing case which did not occur in the single groove case. This is due to the cooling effect of the lubricant entering the bearing through the additional downstream groove. Surprisingly, the temperature fall across the groove, which was of 1.2 °C at a feeding pressure of 100 kPa

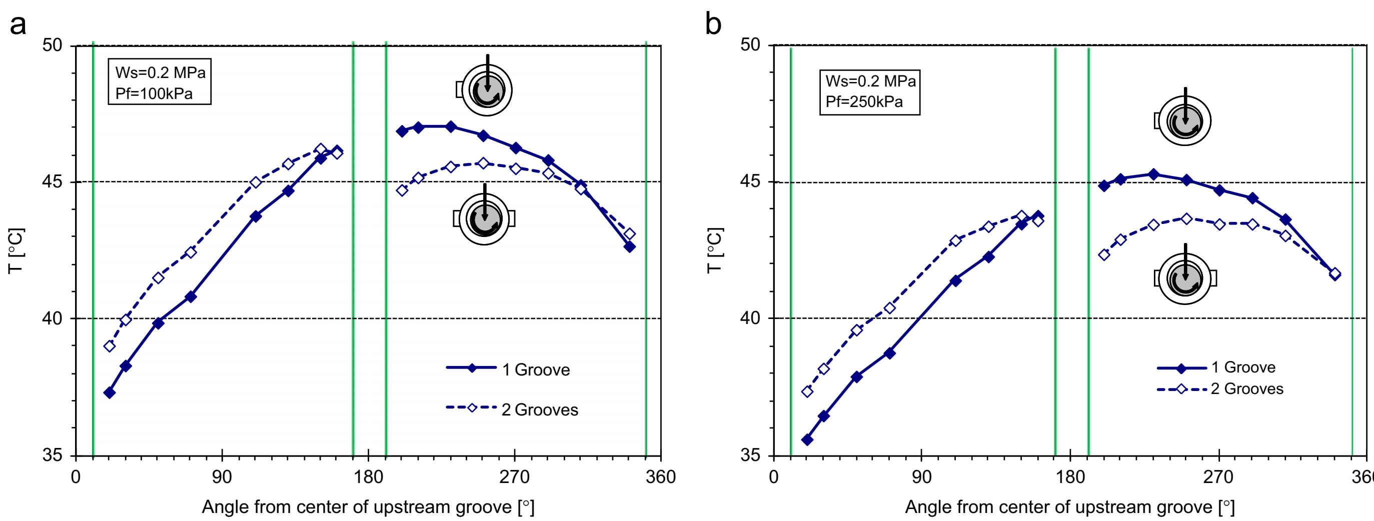

Fig. 6. Influence of the number of grooves on the temperature profile at midplane of the inner bush surface, for Ws=0.2 MPa and (a) P_f=100 kPa and (b) P_f=250 kPa.

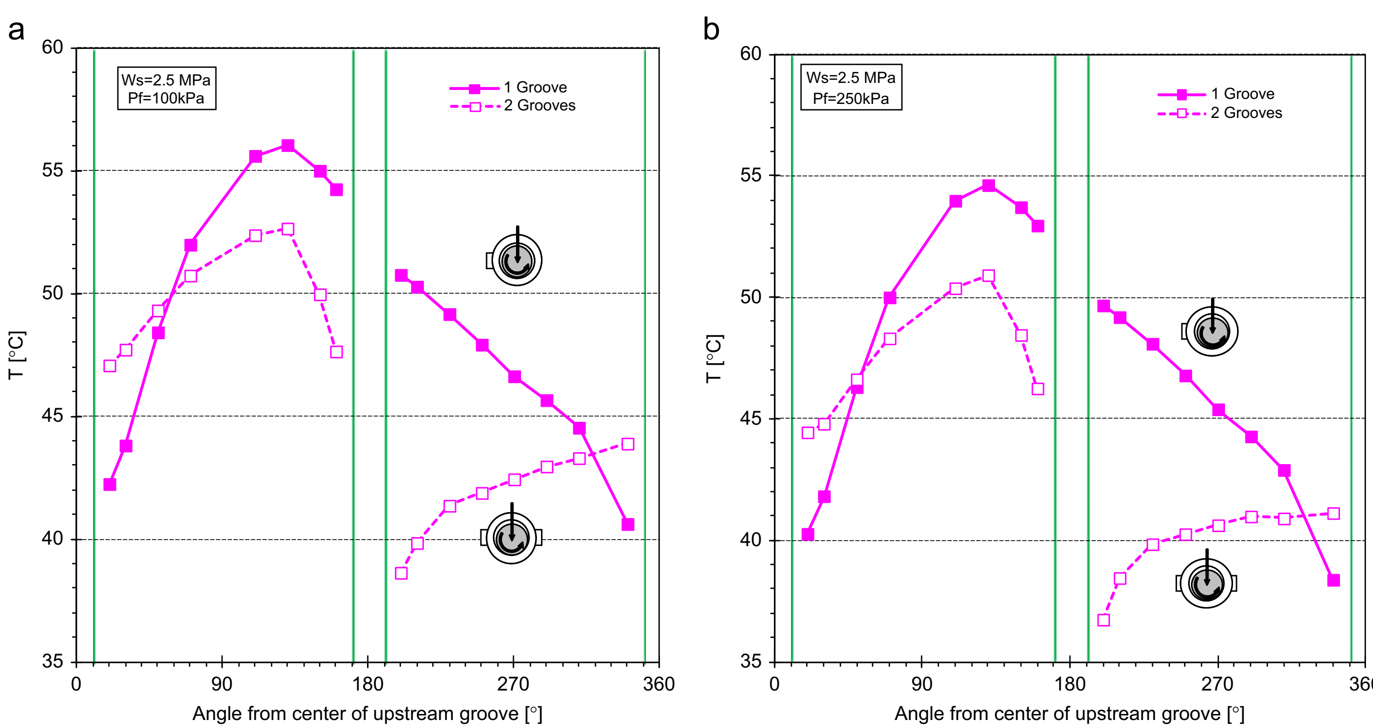

Fig. 7. Influence of the number of grooves on the temperature profile at midplane of the inner bush surface, for Ws=2.5 MPa and (a) P_f=100 kPa and (b) P_f=250 kPa.

(Fig. 6a), was roughly the same for a feeding pressure 2.5 times higher (1.4 °C – Fig. 6b). This is in agreement with the fact that the downstream groove flow rate at low specific loads was small both for low and high feeding pressures (0.05 and 0.07 l/min respectively for P_f = 100 kPa and P_f = 250 kPa —see Fig. 4).

– Despite being moderate, the effect of the activation of the downstream groove was sufficient to move the maximum temperature location from the unloaded land, as in the case of the single groove bearing, to the loaded land of the bearing. Increasing the number of grooves yielded a decrease of 1.1 °C in maximum temperature for 100 kPa and of 1.5 °C for 250 kPa.

Heavily loaded bearing (Ws = 2.5 MPa)—Fig. 7:

Under high specific loads the differences between the single and the twin groove bearings are noticeable both at low (100 kPa—Fig. 7a) and high (250 kPa———Fig. 7b) feeding pressure.

– The cooling effect of the downstream groove is now clear. When this groove is active (twin groove case) not only the temperature falls steeply across it, with a temperature decrease above 9 °C, but also the maximum bush temperature is lowered by about 3.5 °C. Such cooling effect is explained by the high flow rate at this groove (see Fig. 4).
– A rather dissimilar cooling effect of the upstream groove is apparent when comparing the two groove configurations. The temperature at the vicinity of the groove region is lower in the single groove case than in the twin groove case. In the latter case there is even a significant increase of the temperature between the thermocouples located immediately upstream and downstream of this groove. Nevertheless when comparing Fig. 7a and b, some cooling of the bush body can still be perceived because the increasing trend of the temperature curve at the unloaded land (between 180° and 360°) is less steep for the higher P_f test. This indicates that some cooling still exists, nonetheless. Again these differences are only explained with the knowledge of the flow rates at each groove displayed in Fig. 4.

The evolution of the bush maximum temperature (T_{max}) and the lubricant outlet temperature (T_{out}) with specific load is presented in Fig. 8, for both groove configurations (single and twin) and for two different values of P_f (100 kPa and 250 kPa).

As seen in Fig. 8a, T_{max} suffers an initial decrease with increasing specific load but above Ws = 0.3 MPa T_{max} begins increasing consistently. T_{out} also displays an initial decrease with increasing specific load, showing a minimum for a specific load somewhere between 0.3 and 1 MPa, and an ascending trend for higher specific loads. This happens for both groove configurations (single and twin). The trend of T_{out} with specific load (Fig. 8b) displays some correlation with the inverse of the trend of the upstream groove flow rate (recall Fig. 4). When this flow rate increases with increasing specific load, T_{out} tends to decrease, and vice versa. Therefore, T_{out} seems to be intimately connected with the cooling efficiency of the upstream groove flow rate.

It is interesting to note that although T_{max} was always higher for the single groove configuration, T_{out} was almost always higher for the twin groove configuration, except in some lightly loaded tests. This may be explained again by the interconnection found between T_{out} and the upstream groove flow rate that was mentioned in the previous paragraph. In fact, this flow rate is stronger in the case of the single groove bearing.

3.4. Motor consumption

The total motor electrical consumption was recorded in some test sessions with a portable multimeter. The results are displayed in Fig. 9. Although the motor consumption depends on more factors than just the effect of the bearing system, it can provide some insight on the relative influence of specific load and feeding pressure and, particularly, on the impact of the number of grooves, in bearing performance.

In Fig. 9a, the total motor consumption is displayed as a function of feeding pressure, for several specific loads and for

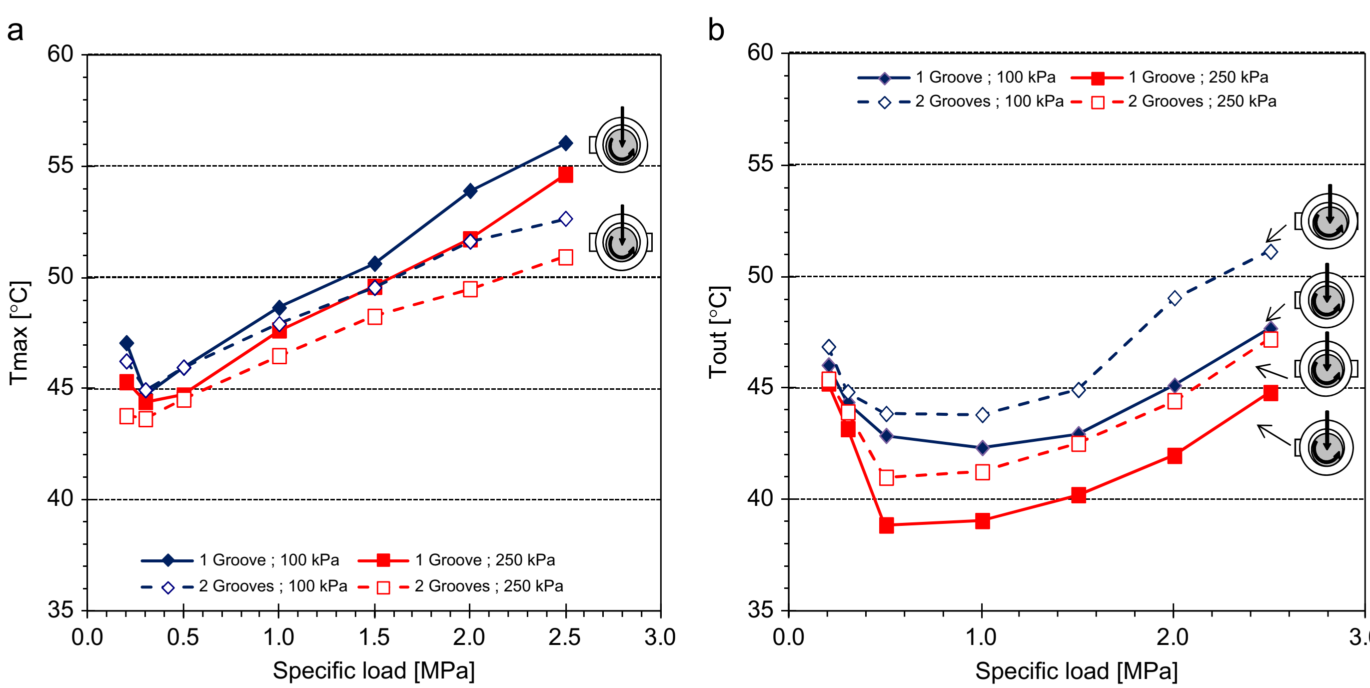

Fig. 8. Influence of the number of grooves (a) on the maximum bush temperature and (b) on the lubricant outlet temperature as a function of specific load and for two different values of feed pressure (P_f= 100 kPa; P_f= 250 kPa).

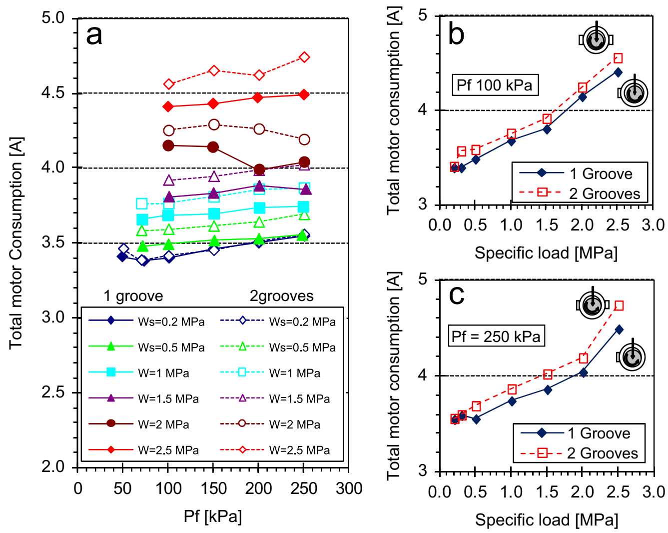

Fig. 9. Influence of the number of grooves on the total motor consumption (a) as function of feeding pressure, (b) as function of specific load for P_f=100 kPa and (c) as function of specific load for P_f=250 kPa.

the two groove configurations (single and twin). Fig. 9b and c also show the motor consumption as a function of specific load for two values of feeding pressure (P_f=100 kPa; P_f=250 kPa). By analyzing Fig. 9a the following can be observed:

- Under low specific loads there is a general ascending tendency of the consumption with increasing feeding pressure. This smooth increase of the electrical consumption, typically around 50 mA for each 100 kPa, happens because as P_f increases, more lubricant is present in the bearing gap, increasing the viscous dissipation. This happens both with single and twin groove bearings.
- Under high specific loads (above 1.5 MPa) the results are more scattered and this poses difficulties for the data analysis. In fact, under high specific loads two opposing effects appear when increasing P_f. Just as in the low specific load case, there is an increase of the extension of the full film region that would favor a higher drag by itself. However, at the same time there is a reduction in lubricant starvation and eccentricity (see Fig. 5a) that reduces drag and in some cases might counterbalance the former effect. In the limit, an excessively low feeding pressure combined with the small groove length ratio of the present case (a/b=0.5, see Table 1) could inclusively induce the appearance of local contact (mixed lubrication regime inception), which would clearly affect negatively not only power loss but also operation safety and bearing life.

The consumption was almost always higher with the twin groove bearing than with the single groove bearing, as can be observed in all three figures. Fig. 9b and c represent the same information as Fig. 9a but consumption is represented as a function of specific load for two different feed pressures. It can be seen that the increase in electric consumption when switching from a single to a twin groove configuration is fairly consistent for most cases. The twin groove bearing displayed consumptions which were normally 2% to 5% higher than those obtained with the single groove bearing. Exceptions to this trend were the very low specific load tests, with negligible differences between the two groove configurations.

Several causes can be pointed out to explain why twin groove journal bearings generally display a higher consumption than single groove bearings. While the unloaded region of the twin groove bearing is filled with lubricant that was supplied through the downstream groove, this is not the case with single groove bearings. On the contrary, the lubricant which leaked out of the bearing at its active region is not compensated until the passing through the upstream groove. Therefore, little lubricant is present within the ruptured film region of single groove bearings, thus reducing the viscous dissipation within this region. Also twin groove bearings display a slightly higher working eccentricity than single groove bearings (recall Fig. 5a).

4. Conclusions

An experimental comparison of the performance of a journal bearing with a single and a twin axial groove configuration has been carried out. This work tried to evaluate the commonsense assumption that a journal bearing with a single groove located at 90 °C to the load line upstream of the position of minimum film thickness would improve its performance if an additional groove would be included at the opposite side. Results concerning total flow rate, flow rate through each groove, shaft locus, inner bush surface temperature, lubricant outlet temperature, and motor consumption have been presented and discussed for a range of specific loads, lubricant feed pressures and groove configurations (single/twin). Some particularities of this work include the individualized measurement of the flow rate through each groove and, as far as the authors know, the first experimental comparison of the performance of single and twin groove journal bearings with virtually the same geometry.

For the conditions tested the following conclusions can be drawn:

i) In the range of specific load studied, the measured total flow rate in the bearing was approximately the same for single and twin groove journal bearings. For the twin groove arrangement, the partial flow rates through each groove varied markedly with increasing specific load.
ii) For twin groove bearings in the very low eccentricity range the flow rate in the upstream groove tends to increase up to a maximum, after which it starts decreasing consistently with increasing specific load.
iii) At high eccentricities the flow entering though the upstream groove is significantly lower than that on the downstream groove, providing poor cooling. For the tests with high eccentricity and low feeding pressure the measured flow rate at the upstream groove was nearly zero. This does not happen with a single groove arrangement.
iv) The flow rate in the downstream groove of twin groove bearings is low at low eccentricities providing, therefore, a poor contribution to the bearing cooling. However, the flow rate at this groove increases consistently with increasing eccentricity, becoming the main contributor to the total flow rate at high eccentricities.

It is not straightforward to conclude which of the groove configurations, single or twin, should be used preferentially. However, based on the present investigation, it seems reasonable to assume with some confidence that the twin groove journal bearing might be problematic under combinations of high eccentricity and low feeding pressure. Twin groove bearings tend to cause slightly more power loss and a higher lubricant outlet temperature than bearings with a single upstream groove at 90° to the load line. Nevertheless, the twin groove bearing tends to display a lower maximum bush temperature and allows the rotation of the shaft in both directions, which might be a design requisite.

As a final remark for bearing design, it seems reasonable to recommend the monitoring of flow rate at each groove, since the information on total flow rate is clearly not sufficient to characterize lubrication effectiveness. With this option available it would be possible to consider groove flow balancing strategies or downstream groove deactivation under selected regimes, for improved lubricant feed to the active region of the bearing and reduced power loss.

Acknowledgment

The present work was supported by the PhD grant SFRH/BD/22278/2005 (financed by FCT POPH—QREN, the European Social Fund and MCTES) and by the project POCTI/39202/EME/2001 funded by FCT—Fundação para a Ciência e a Tecnologia (Portugal) and the European Union fund FEDER. The work has been carried out at the Tribology Laboratory of Minho University. The authors greatly acknowledge the support of these institutions.

References

[1] Costa L, Miranda AS, Fillon M, Claro JCP. An analysis of the influence of oil supply conditions on the thermohydrodynamic performance of a single groove journal bearing. Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology 2003;217:133–44.
[2] Tonnesen J, Hansen PK. Some experiments on the steady state characteristics of a cylindrical fluid film bearing considering thermal effects. ASME, Journal of Lubrication Technology 1981;103:107–14.
[3] Lund JW, Tonnesen J. An approximate analysis of the temperature conditions in a journal bearing. part ii: application. ASME, Journal of Tribology 1984;106:237–45.
[4] Fitzgerald MK, Neal PB. Temperature distributions and heat transfer in journal bearings. ASME, Journal of Tribology 1992;114:122–30.
[5] Ma MT, Taylor CM. A theoretical and experimental study of thermal effects in a plain circular steadily loaded journal bearing. IMechE Seminar: Plain Bearings—Energy Efficiency and Design. London: Mechanical Engineering Publications Ltd.; 1992.
[6] Ma MT, Taylor CM. An experimental investigation of thermal effects in circular and elliptical plain journal bearings. Tribology International 1996;29(1):19–26.
[7] Brito FP, Bouyer J, Fillon M, Miranda AS. Thermal behaviour and performance characteristics of a twin axial groove journal bearing as a function of applied load and oil supply temperature. Tribologia—Finnish Journal of Tribology, The Finnish Society for Tribology 2006;25(3):24–33.
[8] Brito FP, Bouyer J, Fillon M, Miranda AS. Experimental Investigation of the Influence of Supply Temperature and Supply Pressure on the Performance of a Two Axial Groove Hydrodynamic Journal Bearing. ASME, Journal of Tribology 2007;129:98–105.
[9] Basri, H, Neal, PB. Oil flow in axial groove journal bearings. In: Proceedings of the Seminar on Developments in Plain Bearings for the '90s, IMechE Tribology Group. London: Mechanical Engineering Publications; 1990, p. 11–7.
[10] Claro JCP, Miranda AS. Analysis of hydrodynamic journal bearings considering lubricant supply conditions. Proceedings of the Institution of Mechanical Engineers, Part C 1993;207:93–101.
[11] Costa L, Miranda AS, Claro JCP, Fillon M. Temperature, Flow, and eccentricity measurements in a journal bearing with a single axial groove at 90° to the load line. Lubrication Science 2003;15(2):147–61.
[12] Brito, FP, Miranda, AS, Pimenta Claro, JC, Fillon, M. Experimental study of the influence of groove flow rate on the performance of a single and a twin axial groove journal bearing. In: Proceedings of the STLE Annual Meeting. Extended Abstract, 2008.
[13] Brito FP, Miranda AS, Pimenta Claro JC, Fillon M. The role of lubricant supply temperature on the performance of twin groove journal bearings: an experimental study. International Journal of Surface Science and Engineering 2011;5(4):286–99.
[14] Arab Solghar A, Brito FP, Claro JCP, Nassab SAGandjalikhan. . An experimental study of the influence of loading direction on the thermohydrodynamic behavior of twin axial groove journal bearing. Proceedings of the Institution of Mechanical Engineers Part J: Journal of Engineering Tribology 2011;225(5):245–54.