Research Article

Advances in Mechanical Engineering

Advances in Mechanical Engineering
2024, Vol. 16(9) 1–9

DOI: 10.1177/16878132241282522
journals.sagepub.com/home/ade
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Frictional power loss in journal bearing considering parabolic shape for the bearing edges under misalignment

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Abstract

In recent years, there has been a rising demand for minimizing any power loss in industrial applications due to the direct relation to economic as well as climate considerations. This work investigates the parameters that affect the frictional power loss in journal bearings as they are widely used in such applications. The frictional power loss is calculated considering misalignment conditions in the vertical and horizontal planes with the use of a modified bearing shape. Wide ranges of misalignment and bearing shape parameters are considered in the numerical investigation to study the combined effects of misalignment and bearing shape. It has been found that the misalignment significantly increases the frictional power loss at high operating speed levels. The use of a modified bush shape can play an important role in reducing the frictional power loss as well as, it added further advantages in elevating the lubricant film thickness and reducing the pressure levels. Results show that using a modified bearing shape despite the presence of severe misalignment levels reduces the maximum pressure from 20.30 to 17.94 MPa, increasing the minimum film thickness from 2.66 to 8.21 μm and reducing the fictional power loss from 1179.74 to 1094.41 W.

Keywords

Frictional power loss, misalignment, bearing shape, numerical solution

Date received: 7 January 2024; accepted: 26 August 2024

Handling Editor: Sharmili Pandian

Introduction

Journal bearings are typically used in a wide range of critical positions in industrial applications such as internal combustion engines, turbines, and compressors.[1] The increase in power demand requires using equipment working at high speed levels which causes an increase in the frictional power loss in journal bearings. Recently, there has been a worldwide effort to minimize CO_2 emissions, which requires minimizing frictional power loss. Researchers have investigated this concept over the last few years as the negative consequences of climate change have started to be more clear on a global scale. Studies were presented in Allmaier et al.[2–4] to identify the power losses due to friction in journal-bearing systems using

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different operating conditions and supported loads. A comparison between a numerical solution and an experimental work for the calculation of frictional power loss in journal bearing was presented by Allmaier et al.[1] where excellent agreement was found. They found that the frictional power loss in the considered system of journal bearing was about 600 W when the operating speed was 2000 rpm compared to 1600 W when the speed was 4500 W which illustrates how the recent demand for using high operating speeds can increase the power losses. Another interesting study about the frictional power loss in journal bearing was performed by Knauder et al.[5] In their study, journal bearings in heavy-duty diesel engines are investigated for different operating conditions. They found that the frictional power loss depended on the operating speed in addition to the high operation load. Furthermore, they concluded that using ultra-low lubricant viscosity reduced the frictional losses by 8% but it showed a possibility of metal-to-metal contact at full load condition. A numerical investigation of a journal bearing of a hermetic reciprocating compressor was shown by Posch et al.[6] to examine the levels of friction power loss of the journal bearings considering different lubricant viscosities. A hydrodynamic lubrication model was developed by Razavykia et al.[7] to determine the characteristics of connecting rod big-end bearings, which also included the evaluation of the frictional power loss.

Maintaining optimal operating conditions for the journal-bearing system faces significant challenges. One of the most important and unavoidable problems in this concept is journal misalignment. Misalignment may result in large deformation, installation and manufacturing errors, and many other causes. It results in a sharp reduction in the lubricant thickness and also increases the pressure levels. Such changes clearly reduce the system's performance. One of the early studies on the misalignment effect on energy loss in journal bearing was conducted by Safar,[8] where it was found that for the same supported load, a misaligned journal bearing consumed more power due to friction in comparison with the aligned journal bearing. The axial movement of misaligned journal bearing on the frictional power loss was investigated by Li et al.[9]

An experimental study was performed by Bouyer and Fillon[10] to assess the journal bearing performance under misalignment, where they found about an 80% reduction in the film thickness level due to misalignment. A test bench was used by Sun et al.[11] to evaluate the performance of journal bearing under misalignment resulting from shaft bending, where they found that misalignment changes the pressure and lubricant thickness values as well as their distribution shapes. A relation among friction, misalignment, and wear levels was presented by Nikolakopoulos and Papadopoulos[12] based on a numerical solution method. The researchers have investigated the possible reduction of the negative effects of misalignment on system performance. One of the attempts was performed by Fillon and Bouyer[13] where pre-designed defects were used on the bearing geometry to increase the film thickness levels under misalignment conditions. A hyperboloidal bearing shape was by Strzelecki[14] for the purpose of increasing bearing load despite the misalignment presence. A variable bearing profile was used by Ren et al.[15] in order to improve the bearing performance.

Recently, the problem of frictional power loss in journal bearing has been investigated using different aspects. Tauviqirrahman et al.[16] explained how the relatively high-power loss in journal bearing affects the reliability of this type of bearings. They employed an artificial roughness in bearing in order to enhance performance behavior using computational fluid dynamics analysis. Liu et al.[17] used a multiobjective optimization method of journal bearing design. Double parabolic profiles and groove textures were considered in this method using Taguchi and grey relational analysis methods. Their results showed that the optimal combination of six design factors gave maximum load carrying capacity and minimum friction loss. Yang and Palazzolo[18] showed that the power losses in tilting pad journal bearing increase with machine size and speed. They suggested a novel power loss reduction technique based on a thermo-Elasto-Hydrodynamic simulation model. The results showed a power loss reduction up to 27%. Mandal et al.[19] investigated the frictional torque in journal bearings using Taguchi method where both experimental and analytical approaches were carried out to identify the optimal performance of the bearing. The frictional torque in journal bearing was also investigated by Biswas et al.[20] using response surface methodology. The results of the considered optimization strategy indicated a direct relation between peak pressure and frictional torque at a given pressure angle. The defined the presser ingle as the direction of the journal's elevation relative to the bearing on the oil film.

It is clear that the frictional power loss has negative consequences on the economic and CO_2 emissions considerations. Therefore, this work presents a detailed numerical evaluation for the effect of modifying the bearing geometry on the levels of frictional power loss in journal bearings. The numerical solution takes into consideration the presence of deviations (misalignment) in the journal axis with respect to the bearing longitudinal axis. Wide ranges of misalignment and geometry parameters are considered in this study.

Governing equations

The model used in the current work for the solution of the misalignment problem is illustrated in Figure 1. The governing equations for the solution of the ideal case

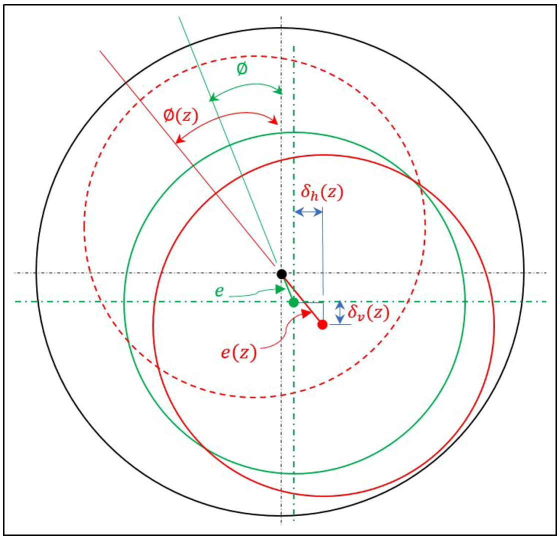

Figure 1. Schematic drawing of the current model (enlarged scale). Green: aligned case, solid red: misaligned shaft (front), and dashed red: misaligned shaft (rear side).

will be represented at first and then the misalignment effect will be explained later.

The Reynolds and the film thickness equations are given by,[21,22]

$$\frac{\partial}{\partial x}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{\rho h^3}{12\eta}\frac{\partial p}{\partial z}\right)=U_m\frac{\partial \rho h}{\partial x}+\frac{\partial \rho h}{\partial t} \quad (1)$$

$$h=c(1+\varepsilon_r\cos(\theta-\emptyset)) \quad (2)$$

where,

p: pressure

h: film thickness.

$U_m=\frac{u_1+u_2}{2}$, u_1, u_2: journal and bearing velocity respectively, $u_1=R\omega$ and $u_2=0$.

ρ, η: lubricant density and viscosity respectively.

c: clearance.

ε_r: eccentricity ratio ($\varepsilon_r=e/c$ where e is the eccentricity between the centers)

$\emptyset$: attitude angle

In the steady state case, $\frac{\partial h}{\partial t}=0$.

The solution to the problem is performed using the Reynolds boundary method that uses the following conditions,[23]

$P=0$ when $\theta=0P=\frac{\partial P}{\partial \theta}=0$ when $\theta=\theta_{cav}$

The position of the cavitation angle ($\theta=\theta_{cav}$) is identified using an iterative solution.[23,24]

The load components, total load, and the attitude angle which are required in determining the convergence of the solution are given by[22]

$$W_r=6\eta\omega RL\left(\frac{R}{C}\right)^2\int_0^L\int_0^{\theta_{cav}}p\cos\theta\, d\theta\, dz \quad (3)$$

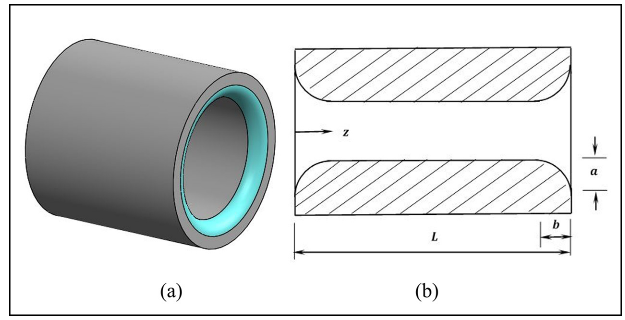

Figure 2. Modifying the bearing edges: (a) whole bush and (b) section illustrates the modification parameters.

$$W_t=6\eta\omega RL\left(\frac{R}{C}\right)^2\int_0^L\int_0^{\theta_{cav}}p\sin\theta\, d\theta\, dz \quad (4)$$

$$W=\sqrt{W_r{}^2+W_t{}^2} \quad (5)$$

$$\emptyset=tan^{-1}(W_t/W_r) \quad (6)$$

In the ideal aligned case the eccentricity ratio (i.e. the eccentricity distance, e) and the attitude angle that appears in equation (2) to calculate the film thickness are constant along the bearing width. The case is different when the misalignment is taken into consideration. The shaft deviation shown in Figure 1 results in variable eccentricity ratio (i.e. the eccentricity distance) and attitude angle in terms of the z. These variations are given by,

$$e(z)=\sqrt{(e_m\cos\emptyset_m-\delta v(z))^2+(e_m\sin\emptyset_m+\delta h(z))^2} \quad (7)$$

$$\emptyset(z)=tan^{-1}\frac{e_m\sin\emptyset_m+\delta h(z)}{e_m\cos\emptyset_m-\delta v(z}\ (\text{for } z\leqslant L/2) \quad (8)$$

$$e(z)=\sqrt{(e_m\cos\emptyset_m+\delta v(z))^2+(e_m\sin\emptyset_m-\delta h(z))^2} \quad (9)$$

$$\emptyset(z)=tan^{-1}\frac{e_m\sin\emptyset_m-\delta h(z)}{e_m\cos\emptyset_m+\delta v(z}\ (\text{for } z>L/2) \quad (10)$$

where,

$\delta v(z)$, $\delta h(z)$: shaft deviation in the vertical and horizontal directions, respectively.

e_m, $\emptyset_m$: eccentricity and attitude angle at the middle plane of the bearing, respectively.

A detailed explanation of the misalignment model can be found in a previous work.[21]

Misalignment results in a reduced clearance at the bearing edges as explained previously. Therefore, modifying the edges of the bush helps in increasing this gap. Figure 2 illustrates this modification in terms of two parameters which are a and b. These parameters represent the height (in the radial direction) and the length (in the longitudinal direction) of modification, respectively.

The gap due to modification, $d(z)$ is a function of z position which can be easily determined for a parabolic shape for the bearing edges as given by the following equations,

$$d(z) = a\left(\frac{1}{b^2}z^2 - \frac{2}{b}z + 1\right) \text{ (for} < b) \quad (11)$$

$$d(z) = a/b^2(z^2 - 2(L-b)z + (L-b)^2) \text{ (for } z > (L-b)) \quad (12)$$

$$d(z) = 0 \text{ (for } b \leqslant z \leqslant (L-b)) \quad (13)$$

The coupling of equations (2), (7–10), and (11–13) results in the total gap between the shaft and the bush surfaces, considering the misalignment and the effect of edge modifications.

Frictional power loss

The frictional power loss (FPL) is determined based on the model suggested by Lund and Thomsen[25]:

$$FPL = \omega \sum \left[\eta R^3 \omega \int_0^\theta l \frac{d\theta}{h} + \frac{1}{2}\varepsilon_r (F_X sin\emptyset - F_Y cos\emptyset) \right] \quad (14)$$

The bearing forces (F_X and F_Y) are determined by integrating the pressure over the solution domain.

Numerical solution

The finite difference method is used in discretizing the related equations. Then, the Gausses- Seidel method is adopted in solving the resulting equations where an overrelaxation scheme is considered to accelerate the convergence of the solution. More details about the discretizing steps can be found in the author's previous work.[21] The convergence of the solution is based on pressure and load criteria. The pressure convergence criterion is given by,

$$\frac{\sum \left| p_{(i,j)new} - p_{(i,j)old} \right|}{\sum p_{(i,j)old}} < 10^{-7}$$

After obtaining the pressure convergence, the bearing forces are calculated using numerical integration for the pressure distribution to determine the total load. If the resulting load is within a tolerance of $\mp 10^{-5}$ of the actual load, the load convergence is satisfied. Otherwise, ε_r is varied and the previous steps are repeated until both criteria are satisfied. Figure 3 shows the main steps of the solution.

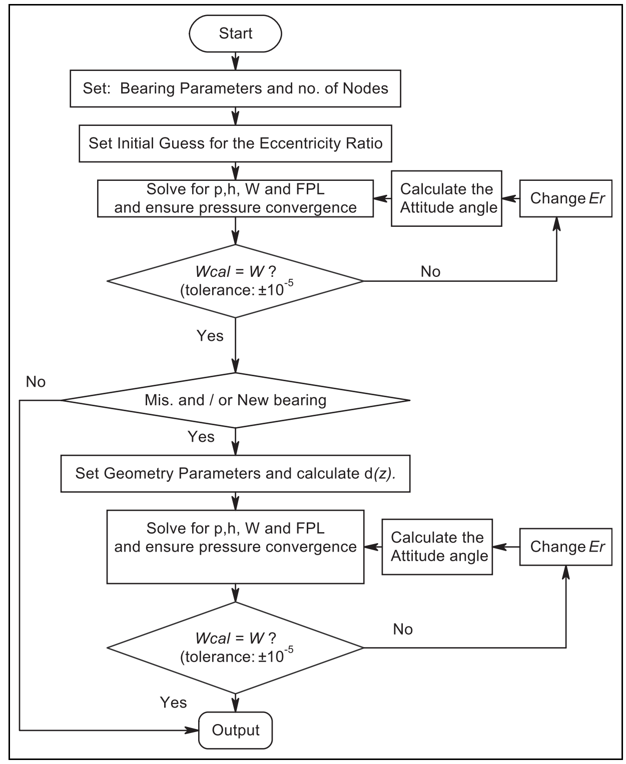

Figure 3. Solution steps.

Table 1. Validation of the current model ($n = 3000$ rpm, $\varepsilon_r = 0.8$, $R = 30$ mm, $L = 66$ mm, $\eta = 0.009$ Pa·s, and $c = 30\mu$m).

Results	Jamali and Al-Hamood[21]	Current work	% Difference
p_{max} (MPa)	33.060	33.183	0.372
Load (kN)	42.889	42.973	0.196

Validation of the current model

The numerical solution is started with the mesh independency test to ensure the use of an adequate number of nodes in the solution space. After a series of tests, it has been found that using 65,341 nodes is sufficient enough to minimize any error in this direction. The results of the current model are compared with the corresponding results in Sun and Changlin[26] for system parameters of $n = 3000$ rpm, $\varepsilon_r = 0.8$, $R = 30$ mm, $L = 66$ mm, $\eta = 0.009$ Pa·s, and $C = 30\mu$m. Table 1 shows this comparison where excellent agreement has been found as the maximum difference is less than 1% in the maximum pressure and the load carrying capacity.

Results

The numerical solution is started with the mesh independence test to ensure the use of an adequate number

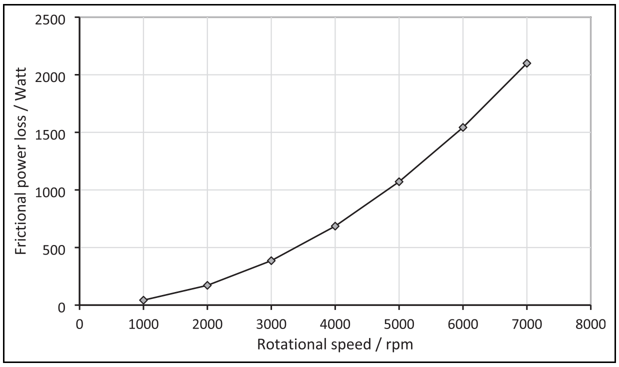

Figure 4. Variation of frictional power loss with the operating speed for the aligned case. $\varepsilon_r = 0.6$, $R = 30$ mm, $L = 66$ mm, $\eta = 0.009$ Pa·s, and $c = 30\mu$m.

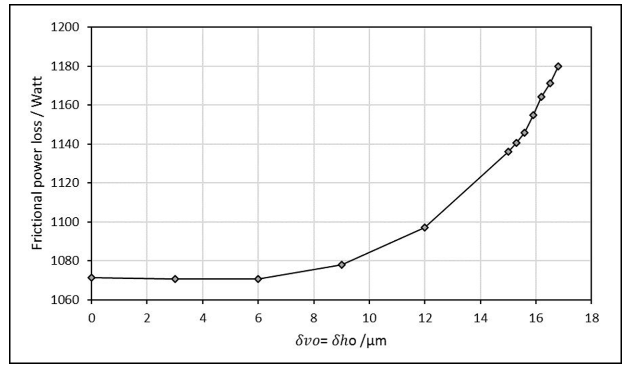

Figure 5. Effect of misalignment on the frictional power loss (at 5000 rpm).

of nodes in the solution space. The case of highest level of misalignment is examined in this test using wide range of nodes ($N \times M$) in the circumferential (N nodes) and longitudinal (M nodes) directions. The maximum pressure, minimum film thickness, and frictional power loss are determined for each $N \times M$ case. After a series of tests, it has been found that using 65,341 nodes ($N = 361$ nodes and $M = 181$ nodes) is sufficient enough to minimize any error in this direction. The bearing parameters of $R = 30$ mm, $L = 66$ mm, and $c = 30\mu$m with a lubricant viscosity of $\eta = 0.009$ Pa·s are used in obtaining the following results. The eccentricity ratio used in the current work is $\varepsilon_r = 0.6$, which is a typical value in industrial applications. Figure 4 shows the variation of frictional power loss with the operating speed for the aligned case. A wide range of operating speeds is used to obtain the results presented in this figure. The range of speed starts from 1000 up to 7000 rpm, which is just below half of the critical speed of the journal. It can be seen that the frictional power loss is significantly related to the operating speed. The frictional power loss, for example, is 42.9 and 2099.8 W when the operating speed is 1000 and 7000 rpm, respectively. It is worth mentioning that the relation between the frictional power loss and the operating speed is not linear, as the operating speed affects the values of the generated pressure in a non-leaner relation.

Figure 5 illustrates the misalignment effect on the frictional power loss when the operating speed is 5000 rpm. Figure 1(b) illustrated previously that the maximum deviations in the two planes are given by δv_o and δh_o. These two terms are considered here as misalignment parameters. A range of 0–16.8 μm misalignment parameters is considered in Figure 5 where the zero value represents the ideal aligned case and the upper limit, 16.8 μm is considered as it produces an extremely thin layer of lubricant. The misalignment effect starts to be significant when the misalignment parameter is greater than 10 μm. This means, in other words, that when the misalignment parameter is less than one-third of the radial clearance, it will not have a significant effect on the frictional power loss. However, journal bearing suffers from misalignment due to installation errors, shaft deformation, and errors related to the tolerances of manufacturing, which may result in higher deviation in the shaft axis. The fictional power losses increase as the misalignment parameters increase and the change is about 10.12% (1071.32–1179.74 W) when the misalignment parameter is 16.8 μm in comparison with the aligned situation. This increase in frictional power loss will certainly have a negative impact on the economic considerations of journal-bearing operations.

The misalignment effect on the frictional power loss is essentially related to its consequences on the maximum pressure (p_{max}) and minimum film thickness (h_{min}) levels. Figure 6 shows the misalignment effect on these bearing characteristics. Figure 6(a) illustrates the misalignment effect on h_{min}. The film thickness reduces from 12.0 μm when the misalignment parameter is zero (ideal case) to 2.661 μm when it is 16.8 μm. This change is a reduction of 77.83% in h_{min}. The corresponding misalignment effect on p_{max} is shown in Figure 6(b). The maximum pressure value increased from 16.35 MPa in the ideal case to 20.30 MPa when the misalignment parameter is 16.8 μm which represents an increase of 24.16%. The misalignment effect on p_{max} in a considerable value starts when the misalignment parameters are greater than 10 μm while it affects h_{min} for the whole range of the misalignment parameter.

The misalignment effect on p_{max}, h_{min} and frictional power loss can be reduced by changing the bearing shape at the positions where the shaft deviation causes the journal and bearing surfaces to be very close to each other which are the bearing edges. Figure 7 shows the effect of using different values of a on the p_{max}, h_{min}, and frictional power loss. The range of a is 0–30 μm where the zero value represents the classical unchanged bearing shape, and the 30μm represents a radial change in the bearing shape equal to the radial clearance. The

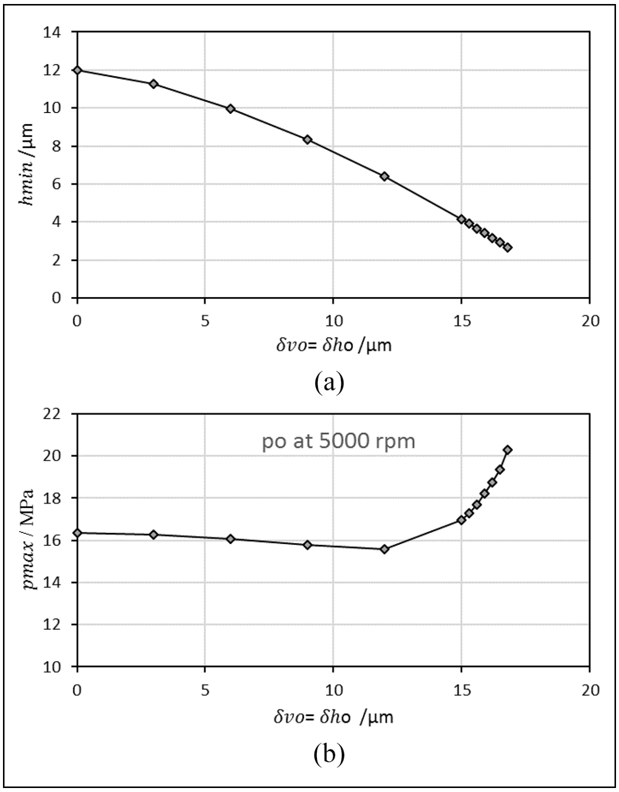

Figure 6. Misalignment effect on (a) h_{min} and (b) p_{max} (at 5000 rpm).

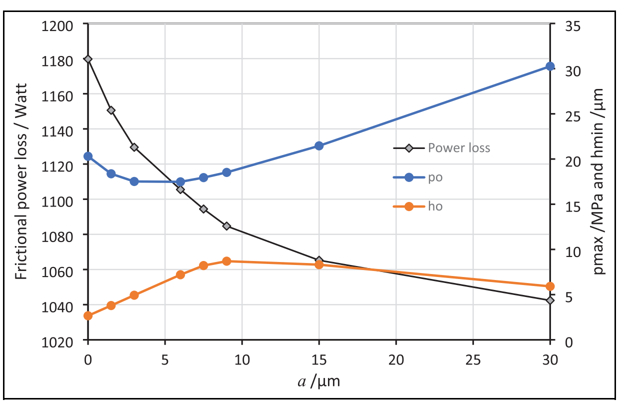

Figure 7. Effect of changing bearing shape on p_{max}, h_{min}, and frictional power loss.

start position of shape-changing is b = 23.1mm which is selected after a series of tests to obtain the optimal value. It is important to note that changing the bearing shape parameter has different consequences on the considered characteristics. Starting with h_{min}, using $a = 9\mu m$ (30% of C) gives the highest value. The minimum film thickness increased from 2.661 μm when $a = 0$ to 8.697 μm when $a = 9\mu m$, which represents a significant improvement in the lubricant layer thickness (more than double). Using higher values of a decreases h_{min}, but its values remain above the corresponding value of the classical unchanged shape. The effect of a on p_{max} is different as p_{max} reduces when $a < 10\mu m$ and increases when $a > 10\mu m$ in comparison with the classical unchanged bearing shape. On the other hand, the frictional power loss decreases in the whole range of a. Comparing all three result trends leads to the conclusion that using $a = 7.5\,\mu m$ improves the levels of p_{max}, h_{min}, and the frictional power loss as it in the same time increases h_{min} by 208.72%, reduces p_{max} by 11.62%, and reduces the frictional power loss by 7.23%.

More details about the effects of changing the bearing shape on the pressure and film thickness distributions are shown in Figure 8. This figure compares the results of three cases, which are the ideal aligned case, the misaligned case, and the misaligned case with the change of bearing shape. It can be seen how the misalignment changes both distributions and causes pressure spikes close to the bearing edges. On the other hand, changing the bearing shape reduces the misalignment influence by elevating lubricant thickness levels and reduces the values of the pressure spikes as the red color (max. pressure and min. film) no longer appears in both distributions.

Conclusions

This work presents a novel solution for the problem of misalignment in journal bearing in terms of using a modified bearing shape considering sever misalignment levels. The frictional power loss, minimum film thickness, and maximum pressure levels are all improved as a result of adopting such bearing shape. A numerical solution to evaluate the effectiveness of minimizing the frictional power loss in journal bearing associated with the shaft misalignment. The solution is performed using the finite difference method. The current solution is validated against the available solution in the literature, where excellent agreement is obtained. At first, the frictional power losses are evaluated for an ideal aligned case and then the misalignment effect on the power loss is determined for a wide range of misalignment parameters. It has been found that misalignment increases the frictional power loss by 10.12% at a misalignment parameter of 16.8 μm. This increase in the frictional power loss is accompanied by an increase in p_{max} by 24.16% and a decrease in h_{min} by 77.83%. Modifying the bearing profile is considered in this work to reduce the misalignment effect on the bearing performance in terms of the frictional power loss, maximum pressure, and minimum film thickness. Considering a variable bearing profile reduces the frictional power loss by 7.23%, reduces p_{max} by 11.62%, and increases h_{min} by 208.72%. These results represent a significant enhancement for the system characteristics, which allow the journal bearing to operate safely (increased

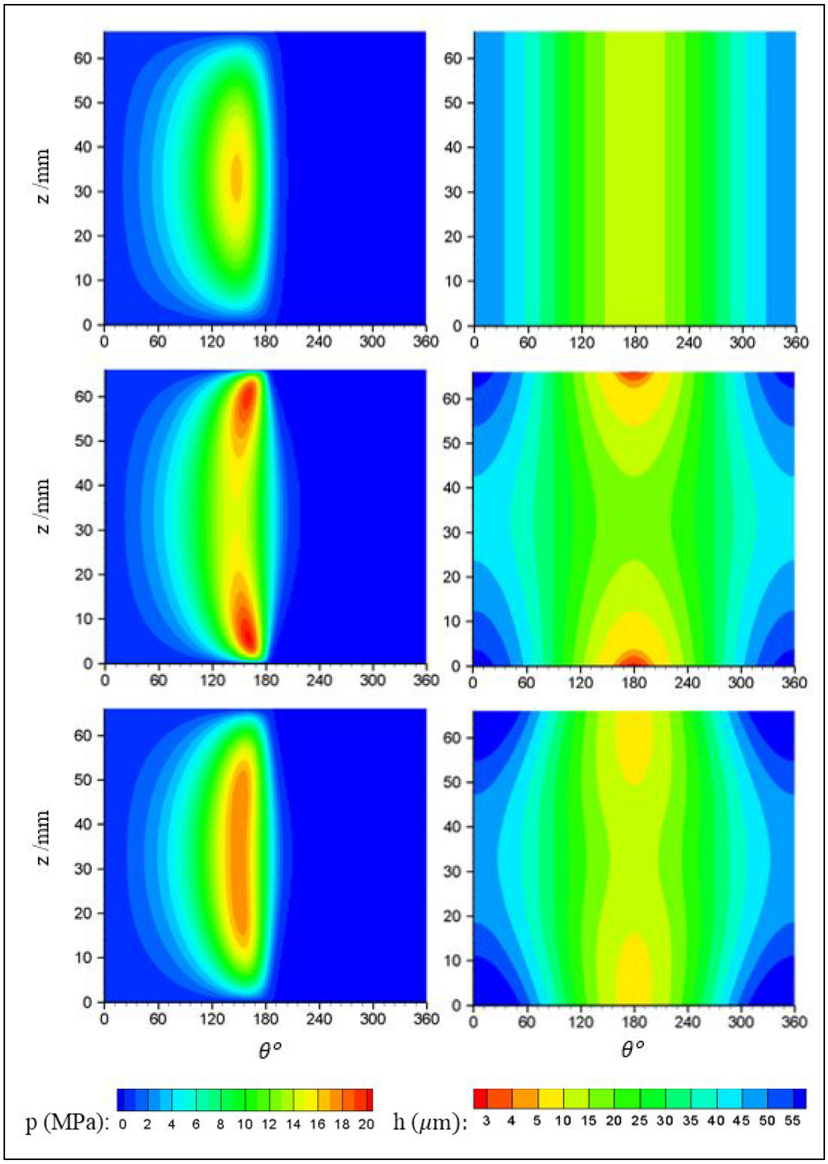

Figure 8. Pressure (left) and film thickness (right distributions for aligned (upper), misaligned (middle), and new bearing design (lower)).

lubricant thickness) at high speed under severe misalignment conditions with a reduced level of frictional power loss. The increase in the minimum film thickness level due to the bearing modification helps in extending the levels of misalignment where journal bearings can operate safely.

Declaration of conflicting interests

The author(s) declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

Funding

The author(s) received no financial support for the research, authorship, and/or publication of this article.

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Data availability statement

The study did not report any data.

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Appendix

Notation

Symbol	Description	Units
a, b	Design parameters	m
c	Clearance	m
$d(z)$	Gap due to modification	m
e	Eccentricity of journal	m
e_m	Eccentricity at the middle plane	m
FPL	Frictional power loss	Watt
h	Oil film thickness	m
$h_{\min}$	Minimum oil film thickness	m
L	Bearing length	m
p	oil film pressure	N/m^2
$p_{\min}$	Maximum oil film pressure	N/m^2
R	Bearing radius	m
U	Velocity	m/s
U_m	Mean velocity	m/s
W_r, W_t	Load components	N
z	Axial coordinate, $0 \leqslant z \leqslant L$	m
β	$\beta = \theta - \emptyset$	degree
$\emptyset$	Attitude angle	degree
$\emptyset_m$	Attitude angle at the middle plane	degree
δh	Horizontal misalignment	m
δv	Vertical misalignment	m
ε_r	Eccentricity ratio, $\varepsilon_r = \frac{e}{c}$	–
η	Lubrication viscosity	Pa·s
ρ	Mass density of oil	kg/m^3
θ	Angle in the circumferential direction	degree
ω	Journal angular velocity, $\omega = 2\pi N$	rad/s