The verification of adiabatic model of a lateral sliding bearing with a floating ring bearing

Aleksander Mazurkow

Rzeszow University of Technology, The Faculty of Mechanical Engineering and Aeronautics, Dept. of Mechanical Engineering, Powstancow Warszawy Avenue 8, 35-959 Rzeszów, Poland
amazurkow@gmail.com

Abstract
This research work describes the adiabatic model of a lateral slide bearing with a floating ring bearing. In order to support recommended assumptions for a designed model and for an evaluation accuracy of achieved results, the author developed a verification method which let comparing results received in a numerical way with that of the ones received in experimental tests. The evaluation criteria of a theoretical model were presented in regard to results from empirical studies. As the basic criteria, the author accepted the position of a bearing journal centre towards a fixed bearing bush, also for these positions pressure and temperature distributions in the outer oil film were chosen. In the conclusion, suggestions were made about the comparison of pressure and temperature distributions, frictional moments on a fixed bush surface, and angular velocities of a floating ring bearing towards a fixed bearing bush. The reasons of differences between measured and calculated values were also discussed.

Keywords: Floating ring bearing, bearing load, oil film, adiabatic model, eccentricity ratio.

Register of symbol: B - bearing width [m], \( B^* = B / D \) - bearing relative width, \( c_p \) - specific heat measured at the constant pressure [J/kg·°C], \( C_R \) - radial clearance [m], \( C_R^* = C_R^2 / C_R^1 \) - quotient of radial clearances, \( D \) - diameter [m], \( e \) - eccentricity [m], \( F \) - bearing load [N], \( F^* \) - dimensionless bearing load, \( F_L \) - oil film load capacity [N], \( h \) - lubricant gap height [m], \( h^*_i = h_i / C_R^i \) - dimensionless lubricant gap height, \( M \) - moment [Nm], \( n \) - rotational speed [rps], \( \psi_i \) - dimensionless pressure of oil film, \( Q_{o,v} \) - volumetric flow of rate \( [m^3/s] \), \( Q_{o,m} \) - mass flow \( [kg/s] \), \( T \) - temperature [°C], \( T_{o,i} \) - oil temperature in feeding gap [°C], \( v \) - speed [m/s], \( x_i \) - Cartesian coordinate in circumferential direction, \( y \) - Cartesian coordinate in radial direction, \( z \) - Cartesian coordinate in axial direction, \( \beta \) - an angle contained between lines going through centres of bearing journals and the beginning of angular coordinates \( \varphi \) – measured to the minimal height of a lubricant gap, \( \varepsilon_i \) - oil film relative eccentricity, \( \varepsilon_0 = e_0 / C_{R_0} \) - resultant relative eccentricity, \( \eta \) - oil dynamic viscosity [Pa·s], \( \eta^*_o \) - oil viscosity in reference temperature [Pa·s], \( \lambda \) - thermal conductivity factor [W/m·°C], \( \rho \) - oil density [kg/m³], \( \psi_i = C_R^i / R_i \) - relative clearance, \( \omega_i \) - angular speed [1/s].

Introduction
In a lateral sliding bearing with a floating ring bearing (Fig.1) in a thermohydrodynamic balance position, the constant load \( F \) in its course and its value which is attached to the bearing journal is balanced by hydrodynamic load capacity of the inner bearing \( F_{L1y} \) as well as the outer bearing \( F_{L2y} \) (Fig. 2). The centre of a journal \( (O_{J1}) \) and a floating ring bearing \( (O_{J2}) \) is moved in an eccentric way and the centre of a fixed bearing bush \( (O_2) \). Values of journals centres displacements define...
eccentricities \((e_1, e_2, e_0)\) whereas, the distance between the straight of a bearing load and the straight of an oil film load capacity define displacements \((e_1, e_2)\).

Frictional moments \(M_2\) \((O_{12})\), \(M_3\) \((O_{12})\), which appear on the surface of the outer and inner floating ring bearing are equal. Frictional moments on the journal and a fixed bush surfaces stand for: \(M_1\) \((O_{11})\) and \(M_4\) \((O_{12})\). Further information concerning the lubricant film geometry is presented in a research work (Mazurkow, 2009).

To design a theoretical model the author took into account assumptions concerning oil flow and heat in a bearing, bearing elements structure as well as a lubricant oil. These presumptions were described in research works (Kaniewski, 1977; Mazurkow, 1993). In order to support recommended assumptions for a designed model and for an evaluation accuracy of achieved results we developed a verification method which let us compare results received in a numerical way with ones received in experimental tests. The author of the discussed above method benefited from experiments conducted by (Buluschek, 1980).

**Equations of a bearing mathematical model**

Model of bearing is described by equations in a dimensionless way:

**Equations of pressures distribution in lubricant gaps \((i=1, 2)\):**

\[
\begin{align*}
&\left(\frac{\partial^2 p_i^*}{\partial \varphi_i^2}\right) + \left(\frac{D_i}{B}\right)^2 \left(\frac{\partial^2 p_i^*}{\partial z^2}\right) + \frac{3}{h_i^*} \left(\frac{\partial p_i^*}{\partial \varphi_i^*}\right) \left(\frac{\partial h_i^*}{\partial \varphi_i^*}\right) = \frac{6}{h_i^*} \omega_i R_i + \omega_{i+1} R_{i+1}\left(\frac{\partial h_i^*}{\partial \varphi_i^*}\right)
\end{align*}
\]

(1)

**Equations of temperature distribution in lubricant gaps \((i=1, 2)\):**

\[
\begin{align*}
&\left(\frac{\partial T_i^*}{\partial \varphi_i^*}\right) = \left(\frac{D_i}{B}\right)^2 \left(\frac{\partial T_i^*}{\partial z}\right) + \frac{12}{h_i^*} \left(\frac{\partial T_i^*}{\partial \varphi_i^*}\right)
\end{align*}
\]

(2)

**Equations of balance of powers and frictional moments:**

\[
F_{L1x}^* = 0, \quad F_{L2x}^* = 0, \quad F_{L1y}^* = F_1^*, \quad F_{L2y}^* = F_2^*,
\]

(3)

\[
M_{2}^* (O_{12}) = M_{3}^* (O_{12}).
\]

(4)

The condition of balance of frictional moments in a bearing can be written as:

\[
\alpha_i = \frac{R_{i1} \cdot \eta_0 \cdot \omega_i \cdot B}{\psi_i \cdot F_{Li} y}.
\]

**Equations of a lubricant factor condition:**

\[
\eta_i^* = \frac{\eta}{\eta_0} = a \cdot T_i^* + b \cdot T_i^*^2,
\]

where: \(a = \eta \cdot \omega_i \cdot \rho \cdot c \cdot p \cdot \psi_i^2\), \(b = b \left(\frac{\eta_0 \cdot \omega_i}{\rho \cdot c \cdot p \cdot \psi_i^2}\right)^2\)

Having established boundary conditions for pressure and temperature areas the authors assumed that circumferential lubricant groove in the inner and outer bearings are completely filled up with oil.

**Boundary conditions of pressure area for these accepted assumptions are as follow:**

- On the boundary separating a circumferential lubricant grooves and a bearing bush:
  \[p(\varphi_{ni} - B/2) = p_{ni}^2\]
- On the boundary separating a bearing bush and environment:
  \[p(\varphi_{ni} + B/2) = 0\]
- On the boundary separating working and non-working zones:
  \[p(<0.2\pi, z) \geq 0,\]

- A zone containing a boundary that joins a bearing bush and a feed tubule, in which temperature has a constant value,
A zone containing a boundary that joins a bearing bush and a circumferential lubricant grooves or a bearing bush and an environment, in which temperature has a variable value.

A zone that occurs on a bearing bush surface, in which pressure has a value equal to zero, it contains a boundary where oil is disappearing.

Essential conditions on boundaries of a bearing bush for solving equations of a temperature area are presented in Fig. 3.

Verification of a theoretical bearing model

Having examined the structure of oil supply system of a bearing in empirical studies it was assumed that it suited feeding method in a theoretical model. Network diagrams of parameters correlations:

- Input and resultant measurements
- Input values for calculations and examined resultant values are described in the Fig. 4
- The analysis of network diagrams suggests that for a verification of author’s own model in numerical investigations as input values he should accept the following parameters from empirical studies:
  - Pressure of oil feeder: $p_z$, feeding temperature: $T_z$,
  - Oil thermophysical qualities: $\eta(T)$, $\rho(T)$, $c_p(T)$, $\lambda$,
  - Geometry of a bearing $R_{ji}$, $R_i$, $B$,
  - Angular velocity of a floating ring $\omega_2$.

Moreover, for calculations we should choose such a relative eccentricity value of the outer oil film $\varepsilon_2$ that a determined load capacity $F_L = F_{L1y} = F_{L2y}$ or an angular velocity of a bearing journal $\omega_1$ can be precisely consistent with measured values. The parameter value $\varepsilon_2$ can be found through an iterative method. The choice of one from above parameters depends on a gradient value of functions:

$$\frac{\partial F_L}{\partial \varepsilon_2}, \frac{\partial \omega_1}{\partial \varepsilon_2}.$$
**Fig. 2.** System of forces and moments in a bearing

**Fig. 3.** Boundary conditions of temperature area

**Fig. 4.** The network diagram of parameters correlations:

<table>
<thead>
<tr>
<th>Input parameters</th>
<th>Resultant parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>( p ), ( T_n ), ( \eta ), ( B_0 ), ( B )</td>
<td>( Q_m ), ( M_4 ), ( e_0 ), ( M_1 ), ( e_1 )</td>
</tr>
</tbody>
</table>

- a) input and resultant measurements
- b) input values for calculations and also calculated resultant values.

\( Q_{010} \) or \( Q_{01} + Q_{02} \)

\( \frac{\partial T_1}{\partial z} = \beta_x \)

\( \frac{\partial T_1}{\partial z} = 0 \)

\( \phi, \beta' = \beta_{1} - 180^\circ \)

\( \beta' = \beta_{1} - 180^\circ \)

\( \omega_1, \omega_2 \)

\( M(O_2) \)

\( F_{oz} \)

\( M_1 \)

Fixed bearing bush

Floating ring

Journal

Feed tube
Fig. 5 a). The influence of a relative eccentricity of an outer bearing on the relative growth of a journal angular velocity and a bearing load capacity

\[ \Delta F/F_c = (\Delta \omega / \omega)_c \times 100 \]

\[ \Delta F/F_c \]

\[ (\Delta \omega / \omega)_c \times 100 \]

Fig. 5b). The comparison of frictional moments on the surface of a fixed bearing bush in the function of a bearing journal angular velocity

\[ M_i [N m] \]

- Theoretical model, \( F = 11852 \) [N]
- Empirical studies, \( F = 11850 \) [N]

Fig. 6. a) The comparison of pressure distributions in an outer lubricant gap for \( z = 0 \), in the section consistent with a circumferential coordinate direction, b) The comparison of temperature distributions in an outer lubricant gap for \( z = B/6 \) in the section consistent with a circumferential coordinate direction, c, d) The comparison of temperature distributions in an outer lubricant gap for \( \phi = 0^\circ, 180^\circ \), and \( 135^\circ, 315^\circ \), in sections consistent with an axial coordinate direction.
Comparative studies of pressure distribution, temperature, position of a journal centre towards a fixed bearing bush

For numerical counts we accepted input values from our experiments which are presented in the Table 1.

\[
\eta(T) = \eta_o \cdot e^{a_o \cdot (T-T_o) + b_o \cdot (T-T_o)^2},
\]

\[
\rho(T) = a_\rho + b_\rho \cdot T + d_\rho \cdot T^2,
\]

\[
c_\rho(T) = a_c + b_c \cdot T + d_c \cdot T^2, \quad \lambda = 0.145,
\]

where:

\[
\eta_o = 0.1084, \quad a_o = -0.55291 \times 10^{-1}, \quad b_o = -0.239 \times 10^{-3},
\]

\[
a_\rho = 896.25, \quad b_\rho = -1.437, \quad d_\rho = 0.87 \times 10^{-2},
\]

\[
a_c = 1802.1, \quad b_c = 2.878, \quad d_c = 0.87 \times 10^{-2}.
\]

As a result of numerical counts we received pressure and temperature distributions in the outer lubricant gap. Distributions in sections consistent with a circumferential coordinate and equivalent to their measured points are shown in the Fig. 6. The comparison of frictional moments on the surface of a fixed bush is described in Fig. 5b. The comparison of a journal positions in relation to a fixed bush is presented in Fig. 7a. The comparison of an angular velocity of a floating ring bearing in the function of a bearing journal angular velocity is described in Fig. 7b. The comparison of other parameters is given in Table 1.

**Conclusion**

On the basis of a developed method the author carried out the verification of a theoretical model of a slide bearing with a floating ring bearing. The evaluation criteria of a theoretical model were presented in regard to results from empirical studies. These criteria were divided into three groups. As the basic criteria the author accepted the position of a bearing journal centre towards a fixed bearing bush, also for these positions he chose pressure and temperature distributions in the outer oil film. These studies constitute the part of a verification method of theoretical models of sliding bearings developed by the author.

These investigations lead to the following conclusions:

- **a)** Comparison tests of pressure distributions (Fig. 6a):

<table>
<thead>
<tr>
<th>Task 1</th>
<th>Input values</th>
<th>Resultant values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
<td>Empirical studies</td>
<td>Results of numerical counts</td>
</tr>
<tr>
<td>(p_2(\varphi_2, z=0))</td>
<td>rys.6a</td>
<td>rys.6a</td>
</tr>
<tr>
<td>(F, F_L [N])</td>
<td>11850</td>
<td>11852</td>
</tr>
<tr>
<td>(\omega_1 [1/s])</td>
<td>366.5</td>
<td>336.06</td>
</tr>
<tr>
<td>(\epsilon_0)</td>
<td>0.65-0.85</td>
<td>0.58</td>
</tr>
<tr>
<td>(\gamma_0)</td>
<td>(300^\circ-303^\circ)</td>
<td>(316^\circ)</td>
</tr>
<tr>
<td>(M_4 [Nm])</td>
<td>3.9</td>
<td>5.6</td>
</tr>
<tr>
<td>(Q_b [kg/s])</td>
<td>0.0065</td>
<td>0.026</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Task 2</th>
<th>Input values</th>
<th>Resultant values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
<td>Empirical studies</td>
<td>Results of numerical counts</td>
</tr>
<tr>
<td>(T_2(\varphi_2, z=B/6))</td>
<td>rys.6b</td>
<td>rys.6b</td>
</tr>
<tr>
<td>(T_2(\varphi_2=135^\circ, 315^\circ, z))</td>
<td>rys.6c</td>
<td>rys.6c</td>
</tr>
<tr>
<td>(T_2(\varphi_2=0^\circ, 180^\circ, z))</td>
<td>rys.6d</td>
<td>rys.6d</td>
</tr>
<tr>
<td>(F, F_L [N])</td>
<td>7830</td>
<td>7825</td>
</tr>
<tr>
<td>(\omega_1 [1/s])</td>
<td>314.16</td>
<td>261.0</td>
</tr>
<tr>
<td>(M_4 [Nm])</td>
<td>2.1</td>
<td>3.0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Task 3</th>
<th>Input values</th>
<th>Resultant values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameter</td>
<td>Empirical studies</td>
<td>Results of numerical counts</td>
</tr>
<tr>
<td>(F, F_L [N])</td>
<td>11850</td>
<td>11850-11894</td>
</tr>
<tr>
<td>(\epsilon_0=\epsilon_0(\gamma_0) [1/s])</td>
<td>Rys.7a</td>
<td>Rys.7a</td>
</tr>
<tr>
<td>(\omega_2=\omega_2(\omega_1) [1/s])</td>
<td>Rys.7b</td>
<td>Rys.7b</td>
</tr>
</tbody>
</table>

- **b)** The comparison between a floating ring bearing angular speed and a bearing journal angular speed shown in the Fig. 6. The comparison of frictional moments on the surface of a fixed bush is described in Fig. 5b. The comparison of a journal positions in relation to a fixed bush is presented in Fig. 7a. The comparison of an angular velocity of a floating ring bearing in the function of a bearing journal angular velocity is described in Fig. 7b. The comparison of other parameters is given in Table 1.

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These investigations lead to the following conclusions:
• Character of a pressure function course for a section compatible with a circumferential direction in the coordinate system, the function obtained through theoretical calculations comes to a big compatibility with pressure distribution calculated in an empirical way.

• Value of a maximum pressure obtained during measurements is smaller than a value obtained in author's own experiments and differs about a relative error equals \( \delta p_{\text{max}} = 0.1 \).

In comparison to experimental researches, in the author's own examinations in circumferential coordinate direction, boundaries position established for a constant oil film lengthens a zone in which pressure is higher than zero.

Operation parameters of a bearing calculated in these studies in comparison with measured values show differences which described by a relative error are equal to:

\[
\delta F_1 = 17 \cdot 10^{-3}, \delta \omega_2 = 9 \cdot 10^{-5} \text{, } \delta M_4 = 3 \cdot 10^{-2} \text{, } \delta \omega_3 = 75 \cdot 10^{-2}.
\]

b) Comparison tests of temperature distribution (Fig. 6b, c, d):

• Character of a temperature distribution course in circumferential coordinate direction received through measurement reaches compatibility with a pressure function course calculated in a numerical way.

• There is a divergence concerning temperature courses determined in sections which are led towards an axial coordinate direction. In the author's own examinations a temperature distribution in the area where \( p > 0 \) has a character of a linear function with increasing values in an oil flow direction from a bearing bush. Whereas, measured temperature values reveal the character of a parabolic curve with the maximum which appears near a bearing bush axis. On the one hand, in the area where \( p > 0 \) measured temperature distributions show the character of parabolic curves, the maximum value of temperature appears in the zone where oil flows away from a bush. On the other hand, a distribution numerically calculated behaves like a curve whose maximum occurs in the oil outflow zone in a bearing bush. The boundary that joins a circumferential lubricant groove and a bush in the area where oil flows away from a circumferential lubricant groove towards a bush, an oil temperature is equal to \( T_{\text{out}} = 55 \, ^\circ \text{C} \). The temperature in a circumferential lubricant groove characterizes itself with its fall near the groove.

c) Comparison tests of frictional moments on the surface of a fixed bearing bush (Fig. 5b), the position of a journal centre in relation to a fixed bearing bush (Fig. 7a), angular velocities of a floating ring bearing in relation to a bearing journal (Fig. 7b):

• Frictional moments on the surface of a fixed bearing bush in the whole analyzed process of journal angular velocities in comparison to measurements take higher values, divergences increase together with the growth of a bearing journal angular velocity.

• For accepted load \( F = 11850 \, [\text{N}] \), in speed range studies \( \omega_1 = 73.3 - 733 \, [1/\text{s}] \) values calculated from the theoretical model and concerning the position of a bearing journal centre relative to a fixed bearing bush \( (\omega_0) \) are smaller than measured values. Whereas, calculated angle values \( (\omega_0) \) corresponding with relative eccentricities \( (\epsilon_0) \) are bigger than measured values.

• Character of a function course \( \omega_2 = \omega_2(\omega_1) \) received by measuring is consistent with a function course achieved during numerical calculations. We can see divergences in higher angular velocities of a bearing journal. When speed equals \( \omega_1 \approx 500 \, [1/\text{s}] \) a bearing journal behaves unsteadily. In numerical calculations there aren't any instabilities in a journal operation, a function course is continuous. The reasons of the instability in operations of slide bearings with a floating ring are described in the following research works (Buluschek, 1980; Mazurkow, 1993; Kicinski, 2005; Mazurkow, 2009).

d) Causes of differences between measured or calculated values

The reasons of above mentioned differences should be looked for either in a developed theoretical model of a bearing or in errors of a measuring method, i.e:

• Defects in a theoretical model which describes the position of constant oil film boundaries result from the assumption of endlessly big lubricant strength on stretching.

• In an accepted thermal model which describes a heat flow in a bearing we do not take into account a heat transfer through bushes and journal surfaces.

• Measurements reveal that the position of a journal axis, a fixed bush or a floating ring bearing is not parallel- where a theoretical model of a bearing was designed for a parallel position of above axis’s (Buluschek, 1980).

• Analysis of temperature distributions achieved in experiments let that misalignment of both a shaft axis and bearing bushes on operation parameters of a bearing is big (Kameron, 1962; Kotynia, 1982). Now, the author is working on the structure of the theoretical model of a slide bearing with a floating ring bearing to deal with the effect of the misalignment of a shaft axis and a floating ring bearing.

References


