INVESTIGATION OF EFFECTS OF OIL ADDITIVE ON FRICTION COEFFICIENT FOR STATICALLY LOADED RADIAL JOURNAL BEARINGS

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ABSTRACT

In this study, variation of friction coefficient under static loading on a radial journal bearing (Chrysler connecting rod bearing) is studied both theoretically and experimentally. The theoretical section of the study was conducted with the KissSOFT program while the experimental part was conducted on a TM 290 hydrodynamic radial journal bearing experimental set. In this study we examine the effect of bearing loads and different proportion of poured additive in alkali – based lubrication oil on friction coefficient. Friction coefficient was calculated from the frictional torque. Consequently, theoretically calculated and experimentally tested friction coefficients were compared and the optimum oil additive ratio was determined.

Keywords: statically loaded radial journal bearing, coefficient of friction, oil additive.

AIMS AND BACKGROUND

Friction can be defined as resistance imposed by surfaces of 2 elements in contact and that are in relative motion against movement or a tendency to move. Sliding, rolling or sliding-rolling type of motion may exist between running components in contact. Therefore, the kinematic friction also becomes in terms of sliding, rolling and sliding-rolling manner. Journal bearings operate based on the principle of sliding friction.

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Friction on surfaces in relative motion can be classified as below based on whether the medium is lubricated or dry:

1. Dry friction;
2. Semi fluid friction;
3. Fluid friction.

Stribeck curve for journal bearings is presented in Fig. 1.

In fluid friction regime, the surfaces undergoing relative motion are completely separated by an oil film. Friction occurs between the oil molecules. The force opposing a body movement depends on the sliding stress between the oil layers. When the pressure on the oil layers reaches a point of balancing the external loads, the surfaces separate from each other.

In hydrodynamic liquid friction regime, the surfaces form a pressure field based on the geometrical and kinematic conditions. Circumferential and transverse pressure distribution for a radial journal bearing is as shown in Fig. 2 (Refs 1–4).

In this study, all the experiments were conducted at 50°C oil temperature, at a constant bearing space of 0.095 mm and along the hydrodynamic liquid friction zone, and Chrysler engine connecting rod bearing (Fig. 3) is used. The tests were run at rotational speeds of 300, 500, 700, 900, 1100, 1300 and 1500 rpm, respectively.
The technological and physical properties of 20W 50 base oil was used in these experiments are shown in Table 1. Boric-based oil additive was added to the base oil in amounts of 3, 5 and 10% and the resulting oil was used for the tests.
Three different parameters were studied in this paper theoretically and experimentally, which are the following:

1. Effects of load on coefficient of friction at constant bearing space;
2. Effects of the oil additives added in the alkali oil on the coefficient of friction;
3. Determination of the optimum amount of oil additive to pour the alkali oil.

THEORETICAL AND EXPERIMENTAL DETERMINATION OF FRICTION COEFFICIENT

Theoretical determination of friction coefficient. KissSOFT packet software was used in determining theoretical behaviour of radial journal bearings under static load. The program executes computations according to DIN 31652 (Refs 7–9). This standard is used for calculations of static hydrodynamic journal bearings operating with low and medium speeds.

The dimensionless bearing gap defined as \( \psi = \frac{D}{d} \) is presented with the Sommerfeld number as given below:

\[
So = \frac{F \psi_{\text{eff}}^2}{DB\eta_{\text{eff}} \alpha_{\text{eff}}} = f \left( \varepsilon, \frac{B}{D}, \Omega \right) \tag{1}
\]

\[
So = \left( \frac{B}{D} \right)^2 \frac{\varepsilon}{2(1-\varepsilon^2)} \sqrt{\pi^2 + 16\varepsilon^2 \left[ \frac{\alpha_1(\varepsilon-1)}{\alpha_2 + \varepsilon} \right]} \tag{2}
\]

where So is the Sommerfeld number; \( D \) – diameter of bearing, mm; \( B \) – width of bearing, mm; \( \Omega \) – angular velocity, 1/s; \( \psi \) – dimensionless bearing gap; \( \varepsilon \) – eccentricity ratio; \( \mu \) – friction coefficient; \( \psi_{\text{eff}} \) – effective Dimensionless bearing gap; \( \eta_{\text{eff}} \) – effective dynamic viscosity, mPas;

where \( \alpha_1 \alpha_1 \) and \( \alpha_2 \alpha_2 \) are defined as follows:

\[
\alpha_1 = 1.1642 - 1.9456 \left( \frac{B}{D} \right) + 7.1161 \left( \frac{B}{D} \right)^2 - 10.1073 \left( \frac{B}{D} \right)^3 + 5.0141 \left( \frac{B}{D} \right)^4 \tag{3}
\]

Table 1. Technological and physical properties of base oil

<table>
<thead>
<tr>
<th>Test</th>
<th>Unit</th>
<th>Typical value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density at 15°C</td>
<td>g/cm³</td>
<td>0.886</td>
</tr>
<tr>
<td>Viscosity at 40°C</td>
<td>cSt</td>
<td>158.9</td>
</tr>
<tr>
<td>Viscosity at 100°C</td>
<td>cSt</td>
<td>18</td>
</tr>
<tr>
<td>Viscosity at –15°C</td>
<td>cP</td>
<td>9300</td>
</tr>
<tr>
<td>Viscosity index</td>
<td>–</td>
<td>125</td>
</tr>
<tr>
<td>Total alkali number</td>
<td>mg KOH/g</td>
<td>8.1</td>
</tr>
<tr>
<td>Flash point</td>
<td>°C</td>
<td>198</td>
</tr>
<tr>
<td>Yield point</td>
<td>°C</td>
<td>–29</td>
</tr>
</tbody>
</table>
$$\alpha_2 = -1.000026 - 0.023634 \left( \frac{B}{D} \right) - 0.4215 \left( \frac{B}{D} \right)^2 - 0.038817 \left( \frac{B}{D} \right)^3 - 0.090551 \left( \frac{B}{D} \right)^4.$$  \hspace{1cm} (4)

The friction coefficient for a statically loaded radial journal bearing is defined as follows:

$$\frac{\mu}{\Psi_{\text{eff}}} = f \left( \text{So}, \frac{B}{D}, \Omega \right)$$ \hspace{1cm} (5)

$$\frac{\mu}{\Psi_{\text{eff}}} = 10^y$$ \hspace{1cm} (6)

$$Y = C + E(\lg \text{So}) + F(\lg \text{So})^2 + G(\lg \text{So})^3 + H(\lg \text{So})^4$$ \hspace{1cm} (7)

$$C = 1.153423 - 2.69332 \left( \frac{B}{D} \right) + 6.552763 \left( \frac{B}{D} \right)^2 - 7.81938 \left( \frac{B}{D} \right)^3 + 3.405146 \left( \frac{B}{D} \right)^4$$ \hspace{1cm} (8)

$$E = -0.7441784 + 0.104245 \left( \frac{B}{D} \right) - 0.343503 \left( \frac{B}{D} \right)^2 + 0.4677244 \left( \frac{B}{D} \right)^3 - 0.215028 \left( \frac{B}{D} \right)^4$$ \hspace{1cm} (9)

$$F = -0.0105291 + 0.342048 \left( \frac{B}{D} \right) - 0.459955 \left( \frac{B}{D} \right)^2 + 0.381193 \left( \frac{B}{D} \right)^3 - 0.1056112 \left( \frac{B}{D} \right)^4$$ \hspace{1cm} (10)

$$G = -0.000397154 + 0.01669 \left( \frac{B}{D} \right) + 0.00966612 \left( \frac{B}{D} \right)^2 - 0.0191126 \left( \frac{B}{D} \right)^3 - 0.01094135 \left( \frac{B}{D} \right)^4$$ \hspace{1cm} (11)

$$H = -0.002584 + 0.00870384 \left( \frac{B}{D} \right) + 0.00157289 \left( \frac{B}{D} \right)^2 - 0.017599 \left( \frac{B}{D} \right)^3 - 0.00668883 \left( \frac{B}{D} \right)^4$$ \hspace{1cm} (12)

Experimental determination of friction coefficient. The TM 290 hydrodynamic journal bearing test machine was developed by GUNT for the purpose of investigating the behaviours of bearings while working. Figures 4a, b and c present the photograph of the test set up and its parts\textsuperscript{10}.

The hydrodynamic journal bearing experimental set has 5 different bearing shafts. These shafts are driven by an electric motor, wich provides to analyse in different bearing gaps (0.035, 0.055, 0.075, 0.095 and 0.2 mm). Radial load (0–500 N) can be adjusted by hand wheel (I7) which has been placed on the radial journal bearing. The shaft rotational speed (0–1500 rpm) can be adjusted by a shaft speed adjusting button easily (30). Frictional torque was measured with a help of a bending beam which has a length of 0.10 m with a strain gauge stickered on it (31). Shaft position can be determined with position sensors which are shown in Fig. 4c. Friction torque (21), shaft rotational speed (I9), oil temperature (25), radial load (I8), oil pressure (20, 24) and shaft position (22, 23) can be read from digital indicator on the test device.
Friction torque which is read on device is utilised for calculation of friction coefficient with operated hydrodynamic journal bearing test device. The friction force is calculated by dividing the frictional torque with the shaft radius. The fric-
tation force on the shaft is formed as shown in Fig. 5 (Ref. 11). As the applied radial load is known, so the frictional force is calculated from equation (15).

\[ M_S = F_S R \]  \hspace{1cm} (13)

\[ M_S = \mu F_N R \]  \hspace{1cm} (14)

where \( M_S \) is friction torque, Nmm; \( F_S \) – friction force, N; \( \mu \) – friction coefficient; \( R \) – radius of bearing, mm; \( F_N \) – radial force, N.

**RESULTS AND DISCUSSION**

*Effects of load on coefficient of friction under constant bearing clearance.* In order to determine the effects of loads on the friction coefficient under constant bearing space, 100, 200 and 300 N loads were applied on the bearing running with the 20W 50 base oil. Values of frictional torque were read from the experimental set and the friction coefficient was determined experimentally using equation (15). Under the same operating conditions KissSOFT packet program was used for the calculation of the theoretical friction coefficients. In Fig. 6 are presented the experimental and theoretical results of variations of friction coefficients with the rotational speeds under constant bearing space.

As it is seen from Fig. 6, the increase in the applied force leads to decrease in the values of friction coefficients on the hydrodynamic liquid friction field. For example, at a rotational speed of 700 rpm, the values of friction coefficients for the experimental work were determined, respectively, 0.078 for 100 N, 0.045 for 200 N and 0.0327 for 300 N. Under each load, the values of theoretically and experimentally determined coefficients of friction at the hydrodynamic liquid friction zone are clearly seen to be very close to each other as seen in Fig. 6.
Effects of oil additive to the base oil on the coefficient of friction. In order to determine the effects of oil additive on the friction coefficient 4 experiments were conducted with different proportions of oil additive into the base oil at a rate of 3, 5 and 10% including base oil. During the experiments 300 N load of stable forces was applied onto the bearing. The values of friction coefficients with respect to the oil additive were determined. Figure 7 shows the graph of variations of friction coefficients with the rotation speed under the condition that 3, 5 and 10% of oil additive has been poured into the base oil.

When Fig. 7 is analysed, it is found that the oil additive has reduced the value of friction coefficient. For example, the values of friction coefficients at the operating speed of 700 rpm, were found to be 0.032 for 20W 50, 0.029 when a 3% of additive oil was poured, 0.027 for 5% of oil was poured and 0.027 for 10% of additive oil was poured.
Determination of optimum amount of oil additive in the base oil. For the purpose of determining the optimum amounts of oil additive for the base oil, loads of 100, 200 and 300 N were applied, respectively 20W 50, containing an additive oil of 3, 5, and 10%. The graphs in Figs 8–10 show that variations of the friction coefficients for each load depending on operating rotation speed.

It is clearly seen from these graphs that for each load value, as the oil additive increases the values of friction coefficients decrease. For instance, by applying a load of 300 N at an operating speed of 1100 rpm, the values of friction coefficients were found to be 0.0411, 0.0403, 0.0387 and 0.0384 for 20W 50 oil, containing an additive of 3, 5 and 10% of oil, respectively.

For every single load in the conducted tests, comparison of the changes in friction coefficients for the oil additives with those of the pure base oil was made and the percentage average variations were calculated and presented in Table 2.

Fig. 8. Variation of friction coefficient with pure oil and with oil additive for 100 N load

Fig. 9. Variation of friction coefficient with pure oil and with oil additive for 200 N load
The data in the Table shows that when more than 5% of oil additive is poured, a relatively slight decrease in the friction coefficients is observed. Thus the optimum oil additive to the base oil is around 5%.

**CONCLUSIONS**

The results of the study show to make the following conclusions:

1. An increase in applied force on a journal bearing running at constant bearing space causes a reduction in the friction coefficient around the hydrodynamic liquid friction zone.

2. It was found that under a constant applied load, addition of additive oil into the alkali-based oil used in a bearing leads to a reduction in the values of friction coefficient on the hydrodynamic liquid friction zone.
3. The optimum amount of oil additive to be added to the alkali based oil is around 5%.

4. For all the experiments conducted, it was found that the values of friction coefficients obtained theoretically on the hydrodynamic liquid zone are very close to those obtained experimentally.

REFERENCES


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