CORRELATIONS REGARDING THE INSTANTANEOUS CARRYING FORCE AND PRESSURE DISTRIBUTION TO THE NARROW SLIDING RADIAL BEARING (L/D < 0.6) WORKING UNDER HARD SHOCKS

Marius Alexandrescu¹, Radu Coteţiu², Adriana Coteţiu³, Nicolae Ungureanu⁴
¹,²,³,⁴North University of Baia Mare, Romania Universitatea de Nord Baia Mare
RO – 420083 Baia Mare, Str. Dr. V. Babeş 62A, ROMANIA

Abstract: We present the determining relationship of carriage in non-dimensional form for narrow radial bearings exposed to shocks and vibrations, as well as the determining relationships of the lubricant minimum thickness in relation to the dynamic loading, and a few experimental assessments concerning the function of radial bearings with HD lubrication in the case of huge challenging working. Due to the very short time of loading radial bearings exposed to shocks and vibrations, of about 0.5-1 ms, we consider only the approaching motion between spindle/axle and bushing on the direction of the center line, without the rotation of the spindle/axle (the case of the non-rotating bearing), so that the effect of the lubricant expulsion be prevalent in the achieving of the self-carrying film.

Key words: impulse loading, squeeze film, radial hydrodynamic bearing.

1. INTRODUCTION

The behaviors study of radial bearings with hydrodynamic lubrication, functioning under conditions of shocks and vibrations, is carried out from a tribological point of view, observing under the aspect of friction and lubrication, the lubricant film, by which the shock is damped. In the case of bearings exposed to heavy loading (shocks) the difficulty occurring stays in the solution to Reynolds’ equation, the equation of energy, the equation of elastic deformations of the axle and bushing surfaces, and the equation of lubricant viscosity and density variation with pressure, and all these together form a non-linear integral and differential system [6].

That is why we consider useful a systemic approach to these problems, with the conviction that the results obtained will contribute to the finding of new solutions, in the qualitative understanding of the phenomena that occur in the functioning of sliding bearings.

We consider the closing motion between spindle and bushing on the direction of the center line, without the rotation of the spindle (the case of the non-rotating bearing), so that the lubricant expulsion effect be prevalent in the achieving of the squeeze film [2].

The modeling of the lubricant expulsion effect (squeeze) starts from Reynolds’s equation. Analytically expressed, the Reynolds equation corresponding to this study, within an isothermal approach is [5]

\[
\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial p}{\partial z} \right) = 12\eta \frac{\partial h}{\partial t},
\]

where \( \eta \) - viscosity of lubricant (Ns/m²); \( p \) - pressure (Pa); \( h \) - fluid film thickness (m).
NOMENCLATURE
L - length of bearing (m); \( \eta \) - viscosity of lubricant (Ns/m\(^2\)); \( G \) - static loading (N); \( p \) - pressure (Pa); \( F \) - dynamically loading (N); \( h \) - fluid film thickness (m); \( D \) - journal diameter (m); \( A_i, B_i, C_i \) – instantaneous squeeze force in dimensional form (N); \( H \) - weight launching height (m); \( c_i \) – time of shock (sec.).

The scheme of a narrow hydrodynamic radial bearing with circular bushing exposed to shocks, modeled in 4 areas, is presented in fig. 1 [2].

![Fig. 1](image)

**Fig. 1 The effect of lubricant expulsion under shock for narrow radial bearing [2]**

The simplified modeling of the lubricant film thickness and carriage under the conditions of a closing motion of the spindle/axle and bushing surfaces for the narrow radial bearing exposed to shocks (figure 1) has as starting point the following hypotheses:

- in area III the motion is of separating surfaces, pressure decreases, it can be practically considered constant under the conditions of cavity occurrence;
- in area II A and II B the section remains “approximately” constant and thus the pressure remains constant;
- area I represents the only area that really opposes the closing motion: the geometry of the lubricant film will be approximated with a constant thickness surface, equal to the minimum thickness of the lubricant film under the condition of static loading, on the basis of the rectangular model of infinite length.

We can write [3]

\[
h_m = \frac{1}{\sqrt{\frac{1}{h_{m0}^2} + \frac{8F}{\eta \pi DL^3 g}}},
\]

where \( h_{m0} \) represents the minimum thickness of lubricant under static regime, and \( h_m \) represents the minimum lubricant thickness in the dynamic regime.
The instantaneous squeeze force has the following expression [3]

$$F_s = \frac{1}{A} \left[ \overline{H}_s \left( 1 + A \right) - \overline{H}_s^2 \right],$$

(3)

where $A = 4F \Pi$, $\overline{H}_s = \frac{h_{m0}}{h_m} = H_{ad}$ and the parameters of lubricant film expulsion $\Pi$ have the expression $\Pi = \frac{H}{h_{m0}}$ (H being the height from which the weight dynamically loading the bearing is launched).

2. EXPERIMENTAL DEVICES AND ACQUISITION CHAINS

The pressure distribution was determined in the lubricated film in those 5 points on the bearing’s body with the help of pressure measuring with tensometric translators. Figure 2 presents the bushing diagram in experimental assessments having $L/D=0.5$ [3].

![Fig.2. The bushing of the experimental radial bearing [3]](image)

The research was made using a HD radial bearing with $L/D=0.5$ and the spindle’s diameter $d_e = 59.86$ mm, and the bushing diameter $D_e = 59.93$ mm, spindle’s asperity 58-62 HRC, made of 18MoCr10, bronze bushing made of 88%Sn, 8%Sb, 4%Cu [1]. The dynamic loading of the bearing is made through the lancing of a weight which hits the bearing at different heights. They were made assessments for heights between 5 and 40 cm, using a weight with $m=5$ kg, so as for $H=5$ cm we have $F_1=1665$ N, for $H=20$ cm we have $F_2=2356$ N, and for $H=40$ cm we have $F_3=3332$ N. Figure 3 presents the pressure measuring chain in the lubricant film [3].
The pressure increase was made bar by bar, the dose distortion being linear with the pressure. It was established the dependency relation between the pressure and the tension in the exit point in mV (2.3 mV = 1 bar $\Delta p$) [3].

3. THEORETICAL RESULTS

The variations of the instantaneous carrying force, in relation to the dimensional thickness of the lubricant film and in relation to the time of shock, for the three weight launching heights $H$ are presented in figures 4 and 5 [4].

**Fig. 4** The instantaneous carrying force in relation to the dimensional thickness of the lubricant film ($n=370$ rot/min, $p_{in}=0.5$ bar, $G=2250$ N, $h_{m0}=10.175 \mu m$) [4]

**Fig. 5** The instantaneous carrying force in relation to the dimensional thickness of the lubricant film ($n=370$ rot/min, $p_{in}=0.5$ bar, $G=4500$ N, $h_{m0}=6.723 \mu m$) [4]
4. EXPERIMENTAL RESULTS

The pressure distribution in places P1 – P5 of the bearing’s body, depending on the static G and dynamic F charging conditions are presented in figure 6 for spindle’s rotations n=370 rot/min, and available supply pressure $p_{in}=0.5$ bar [3].

Fig.6. The dynamic pressure distribution depending on the static and dynamic charging conditions of the bearing at $n=370$ rot/min, $p_{in}=0.5$ bar [3]
5. CONCLUSIONS

From the analysis of the theoretical and experimental results, the following observations can be stated:

- the drastic decrease of the lubricant film minimum thickness along with the increase of dynamic loading (decrease ranging between 50% for the rotation of 370 rot/min);
- the decrease of the lubricant film minimum thickness along with the increase of static loading;
- the insignificant influence of the feeding pressure on the minimum thickness of the lubricant for the same rotation of the spindle;
- the ratio of film thickness $H_{s\_ad}$ sensitively influences carriage: once the area of maximum is outrun, the carriage rapidly decreases;
- the existence of an optimum point from the viewpoint of carriage: any change in the functional parameters of the bearing leads to straying from the optimum value from the viewpoint of carriage;
- the dynamic pressure from the moment of shock is increased when increasing the dynamic charging conditions; this increasing process refers to the all active zone, the dynamic pressure rise at the same time with the rise of dynamic charging, depending on the studied position of the periphery zone of the bushing the pressure leap being between 5,95 and 7,45 multiplied with static pressure;
- in all these situations the following fact is to bare in mind: the short time for pressure variation in dynamic charging (under 0,5 ms).

6. BIBLIOGRAPHY