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THE DYNAMIC CIRCUMFERENTIAL PRESSURE DISTRIBUTION TO THE SLIDING RADIAL BEARING WORKING UNDER HARD SHOCKS

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Abstract: *This paper presents a few theoretical assessments concerning the function of radial bearings with HD lubrication in the case of huge challenging working. We consider only the approaching motion between spindle/axle and bushing on the direction of the centre 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. We present the determining relationship of circumferential pressure distribution, squeeze force coefficient γ 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.*

Key words: *squeeze film, radial hydrodynamic bearing, impulse loading, circumferential pressure distribution.*

1. Introduction

The modelling of the lubricant expulsion effect (squeeze) starts from Reynolds's equation, in which we have to consider the terms that contain the closing speed of the two surfaces ($V = -\frac{\partial h}{\partial t}$). Analytically expressed, the Reynolds equation corresponding to this study, within an isothermal approach is [5]:

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

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

The simplified modelling 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

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minimum thickness of the lubricant film under the condition of static loading, on the basis of the rectangular model of infinite length.;

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 [1], [3].

We can write:

$$h = \frac{J}{2}\left(1 - \varepsilon\cos\theta\right), \tag{2}$$

and the medium circumferential pressure distribution to the narrow radial bearing is:

$$p_m(\theta) = \frac{8\eta V B^2}{J^3\left(1-\varepsilon\cos\theta\right)^3}, \tag{3}$$

So, the circumferential pressure distribution is:

$$p(\theta) = \frac{12\eta V B^2}{J^3\left(1-\varepsilon\cos\theta\right)^3}, \tag{4}$$

where ɗ is the angular coordinate, V is the bushing surfaces velocity immediately before impact and V0 is the velocity immediately after impact:

$$\begin{aligned} V &= -\frac{dh}{dt} = \frac{J}{2}\dot{\varepsilon} = \\ &= V_0 - \frac{\eta\pi\,\mathrm{DL}^3\mathrm{g}}{8F}\left(\frac{1}{h_m^{\ 2}} - \frac{1}{h_{m0}^2}\right). \end{aligned} \tag{5}$$

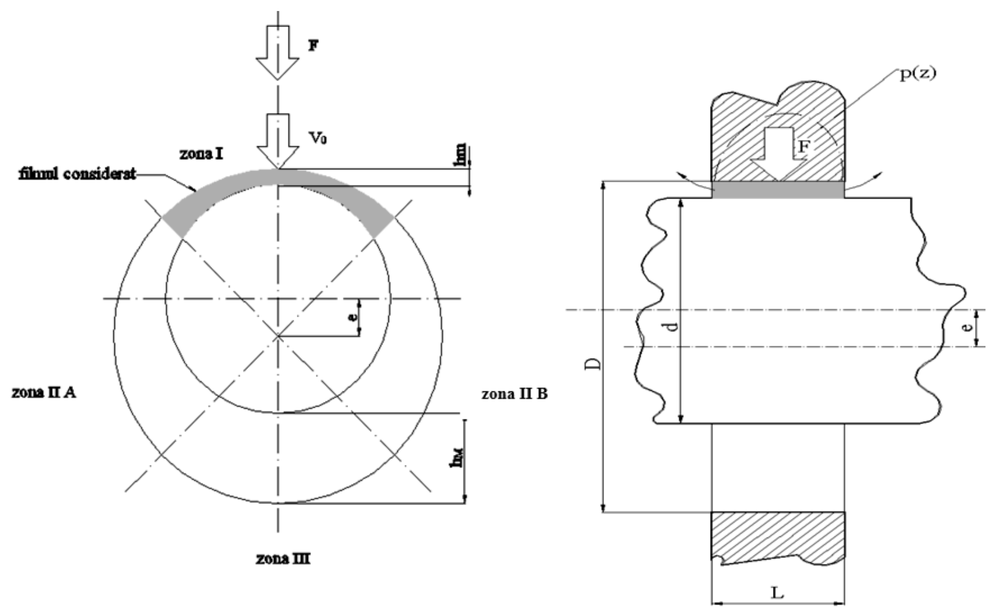

Fig. 1. *The effect of lubricant expulsion under shock for narrow radial bearing*

L- length of bearing [m]; η- viscosity of lubricant [Ns/m²]; G- static loading [N]; p- pressure [Pa]; F- dynamically loading [N]; h- fluid film thickness [m]; D- journal diameter [m].

The relative eccentricity given by:

$$\varepsilon(t)=1-\frac{2h_m(t)}{J}, \tag{6}$$

and,

$$h_m=\frac{1}{\sqrt{\frac{1}{h_{m0}^2}+\frac{8F\sqrt{2\mathrm{gH}}}{\eta\pi\,\mathrm{DL}^3\mathrm{g}}}}, \tag{7}$$

where h_{m0} represents the minimum thickness of lubricant under static regime, and h_m represents the minimum lubricant thickness in the dynamic regime [5].

The instantaneous squeeze force has the following expression:

$$F^*=2\int_0^{\pi/2}\frac{4\eta V\,\mathrm{DL}^3\cos^2\theta\cdot d\theta}{J^3(1-\varepsilon\cos\theta)^3}, \tag{8}$$

We can write:

$$F^*=\frac{8\eta V\,\mathrm{DL}^3}{J^3}\int_0^{\pi/2}\frac{\cos^2\theta\cdot d\theta}{(1-\varepsilon\cos\theta)^3}= =\frac{8\eta VDL^3}{J^3}I_3^{02}\Big|_0^{\pi/2}, \tag{9}$$

where $I_3^{02}\Big|_0^{\pi/2}$ represent the ,,Booker integralò, and have the expression [2]:

$$I_3^{02}\Big|_0^{\pi/2}=\frac{1}{2(1-\varepsilon^2)^2}\cdot \cdot\left[\frac{(2\varepsilon^2+1)}{(1-\varepsilon^2)^{1/2}}\cos^{-1}(\varepsilon)-3\varepsilon\right], \tag{10}$$

We can write:

$$\frac{F^*}{\mathrm{LD}}=p_m=\frac{4\eta\dot{\varepsilon}}{\psi^2}\left(\frac{\mathrm{L}}{\mathrm{D}}\right)^2 I_3^{02}\Big|_0^{\pi/2}, \tag{11}$$

results the following expression for the squeeze force coefficient ɔ:

$$\gamma=\frac{2p_m\psi^2}{\eta\dot{\varepsilon}}=\frac{2}{\delta^2(2-\delta)^2}\cdot\left(\frac{L}{D}\right)^2 \left[\frac{2(1-\delta)^2+1}{\sqrt{\delta(1-\delta)}}\cos^{-1}(1-\delta)-2(1-\delta)\right]. \tag{12}$$

2. Theoretical Results

The dynamic loading of the bearing is made through the launching 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 F1= 1665 N, for H= 20 cm we have F2= 2356 N, and for H= 40 cm we have F3= 3332 N. The static working conditions is presented for the following value H= 0 cm [4]. The research was made using a HD radial bearing with L/D= 0,5 and the spindleȏs diameter de = 59.86 mm, and the bushing diameter De = 59.93 mm, spindleȏs asperity 58-62 HRC, made of 18MoCr10, bronze bushing made of 88% Sn, 8%Sb, 4%Cu.

The dynamic circumferential pressure distribution depending on the static and dynamic charging conditions of the bearing in a dynamic running regime, for three rotations of spindle n= 370 rot/min, n= 600 rot/min and n= 960 rot/min, loading pressures ranging from 0.5 bar to 8 bar, and two static loadings, G= 2250 N respectively G= 4500 N, are presented in Figure 2 to Figure 4.

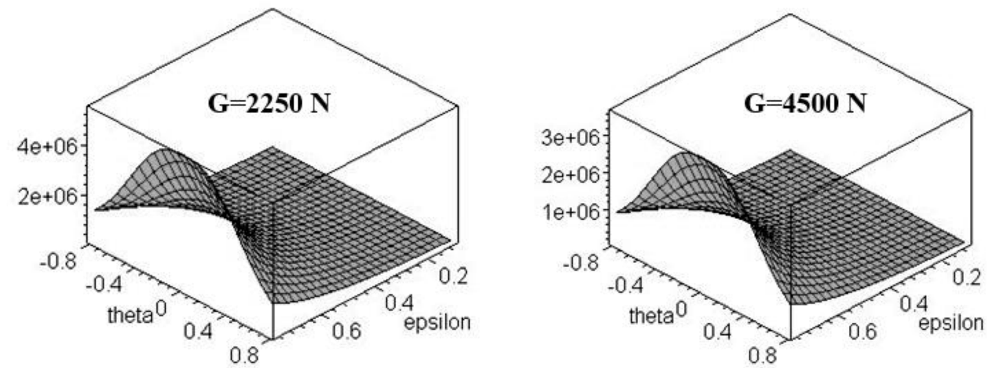

Fig. 2. *The dynamic circumferential pressure distribution depending on the static and dynamic charging conditions of the bearing (F_3=3332.5 N, n=370 rot/min, pin =0.5 bar)*

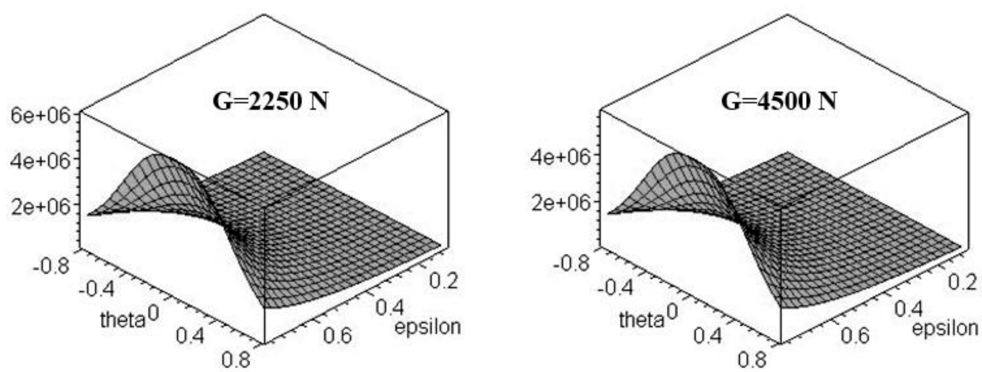

Fig. 3. *The dynamic circumferential pressure distribution depending on the static and dynamic charging conditions of the bearing (F_3=3332.5 N, n=600 rot/min, pin =1.5 bar)*

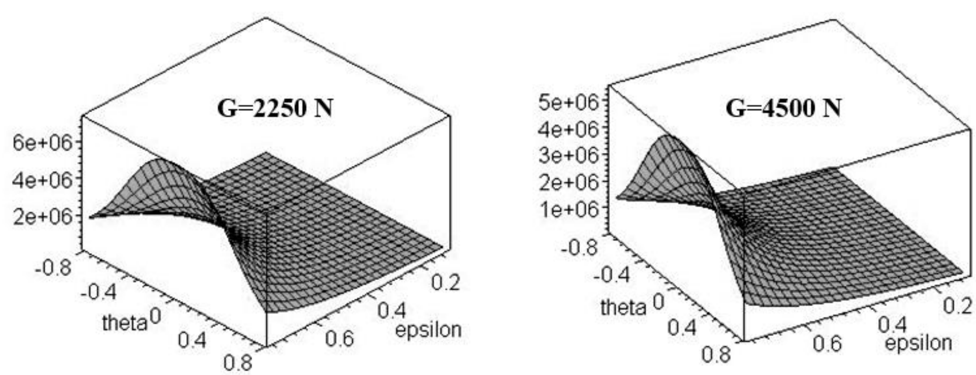

Fig. 4. *The dynamic circumferential pressure distribution depending on the static and dynamic charging conditions of the bearing (F_3=3332.5 N, n=960 rot/min, pin =8 bar)*

The squeeze force coefficient ɔ depending on the relative fluid film thickness ŭ are presented in figures 5 and 6.

3. Conclusions

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

- 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 ratio of film thickness sensitively influences carriage: once the area of maximum is outrun, the carriage rapidly decreases;

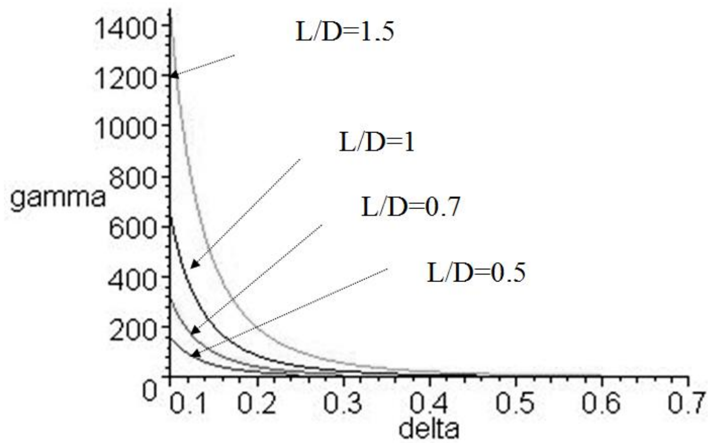

Fig. 5. *The squeeze force coefficient γ (gamma) depending on the relative fluid film thickness δ (delta), for L/D between 0.5 and 1.5*

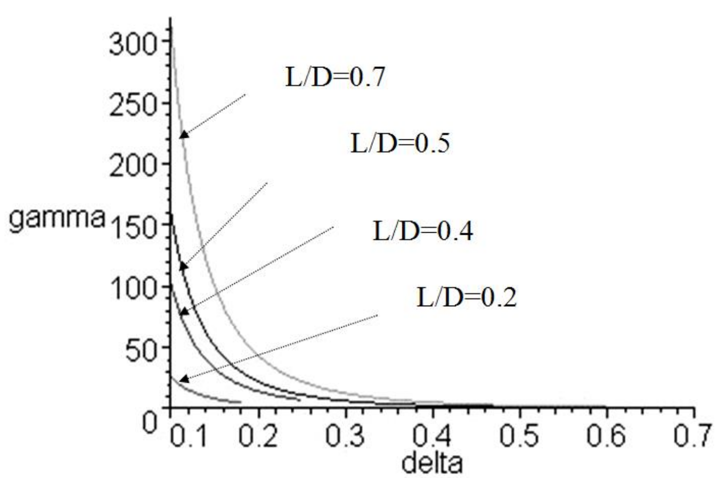

Fig. 6. *The squeeze force coefficient γ (gamma) depending on the relative fluid film thickness δ (delta), for L/D between 0.2 and 0.7*

- the decrease, for high dynamic loading (over 2350 N) of the lubricant film thickness under the admissible acceptable value on the basis of rugosity of spindle surfaces, of the bushing respectively ($h_{min,a} \geq 5$ μm);
- the static charging conditions of the bearing does not have an important influence regarding the changing in the pressureôs values, as the static charging conditions gets bigger, so as the dynamic pressure is bigger;
- the draughtôs pressure in dynamic conditions has a slightly shifting to the entrance zone of the lubricant when static charging conditions are increasing;
- the instantaneous squeeze force increase with the increase of relative eccentricity Ů (decrease of ŭ); the instantaneous squeeze force increase with L/D, for the same value of the relative fluid film thickness ŭ;
- the maximum circumferential pressure distribution is for the angular coordinate ď=0 ę (to mark with centre line);
- the circumferential pressure distribution decrease with the decrease of relative eccentricity Ů;
- the maximum circumferential pressure distribution, for the same dynamic charging conditions, decrease with the static charging conditions gets bigger.

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