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SOME ASPECTS REGARDING INSTANTANEOUS SQUEEZE FORCE TO THE NARROW RADIAL BEARING WORKING UNDER HARD SHOCKS

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***Abstract**: Due to the very short time of loading radial bearings exposed to shocks and vibrations, of about 0.5-1 ms, the effect of the lubricant expulsion be prevalent in the achieving of the self-carrying film. 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). 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.*
***Key words** : impulse loading, squeeze film, radial hydrodynamic bearing.*

1. INTRODUCTION

The modelling of the lubricant expulsion effect (squeeze) starts from Reynold's equation, in which we have to consider the terms that contain the closing speed of the two surfaces ($V = -\partial h / \partial t$). Analytically expressed, the Reynolds equation corresponding to this study, within an isothermal approach is [6],[7]

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3 \frac{\partial p}{\partial z}\right) = 12\eta \frac{\partial h}{\partial t}. \tag{1.1}$$

NOMENCLATURE

L- length of bearing (m); η- 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); Fs_ad - instantaneous squeeze force; 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 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 1.1) 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.

The scheme of a narrow hydrodynamic radial bearing with circular bushing exposed to shocks, modelled in 4 areas, is presented in fig. 1.1 [2],[3].

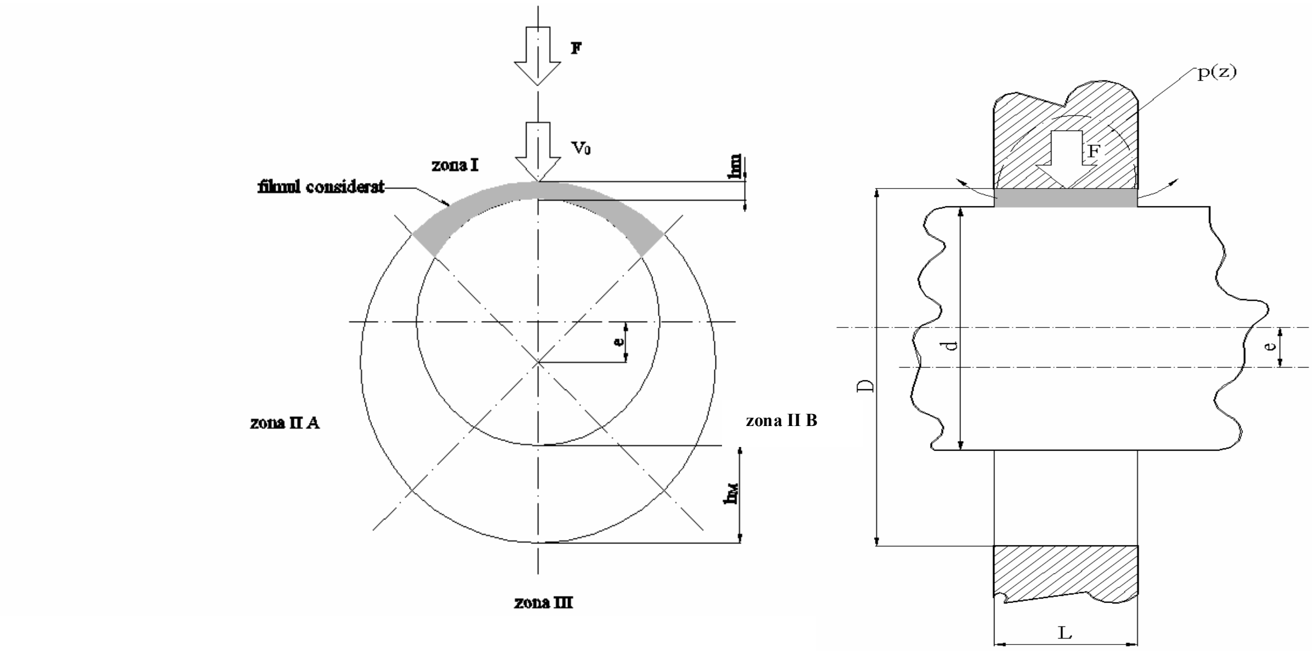

Fig. 1.1 *The effect of lubricant expulsion under shock for narrow radial bearing*

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 [3], [5].

The medium circumferential pressure distribution to the narrow radial bearing is

$$p_m(\theta) = \frac{8\eta V B^2}{J^3(1-\varepsilon\cos\theta)^3}. \tag{1.2}$$

where θ is the angular coordinate, V is the bushing surfaces velocity immediately before impact and V_0 is the velocity immediately after impact:

$$V = -\frac{dh}{dt} = \frac{J}{2}\dot{\varepsilon} = V_0 - \frac{\eta\pi D L^3 g}{8F}\left(\frac{1}{h_m^{\ 2}} - \frac{1}{h_{m0}^2}\right). \tag{1.3}$$

The relative eccentricity given by

$$\varepsilon(t) = 1 - \frac{2h_m(t)}{J}, \tag{1.4}$$

and

$$h_m = \frac{1}{\sqrt{\frac{1}{h_{m0}^2} + \frac{8F\sqrt{2gH}}{\eta\pi DL^3 g}}}, \tag{1.5}$$

where h_{m0} represents the minimum thickness of lubricant under static regime, and h_m represents the minimum lubricant thickness in the dynamic regime [2].

The instantaneous squeeze force has the following expression

$$\overline{F}_s = \frac{1}{A}\left[\overline{H}_s^{\,3}(1+A) - \overline{H}_s^5\right], \tag{1.6}$$

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

2. TEORETICAL RESULTS

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 [3].

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 F_1=1665 N, for H=20 cm we haveF_2=2356 N, and for H=40 cm we have F_3=3332 N. The static working conditions is presented for the following value H=0 cm [1].

The variations of the instantaneous carrying force, in relation to the non-dimensional thickness of the lubricant film for the three weight launching heights H are presented in figures 2.1 - 2.3 [4].

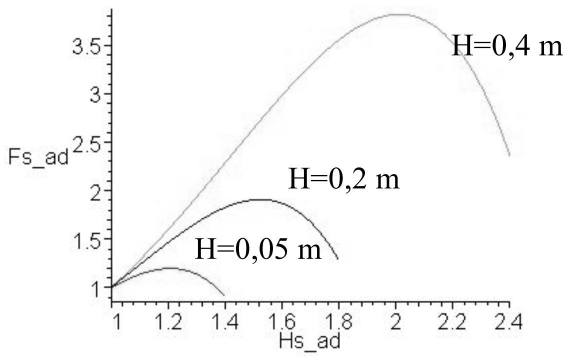

Fig. 2.1 *The instantaneous carrying force in relation to the non-dimensional thickness of the lubricant film (G=2250 N, n=370 rot/min, p_{in}=0,5 bar)*

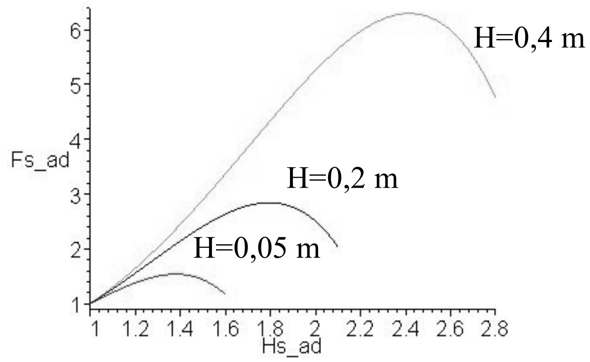

Fig. 2.2 *The instantaneous carrying force in relation to the non-dimensional thickness of the lubricant film (G=2250 N, n=600 rot/min, p_{in}=1,5 bar)*

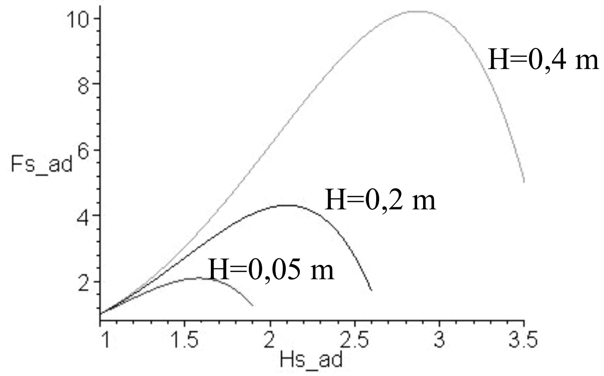

Fig. 2.3 *The instantaneous carrying force in relation to the non-dimensional thickness of the lubricant film (G=2250 N, n=960 rot/min, p_{in}=8 bar)*

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 2.4 - 2.6 [3].

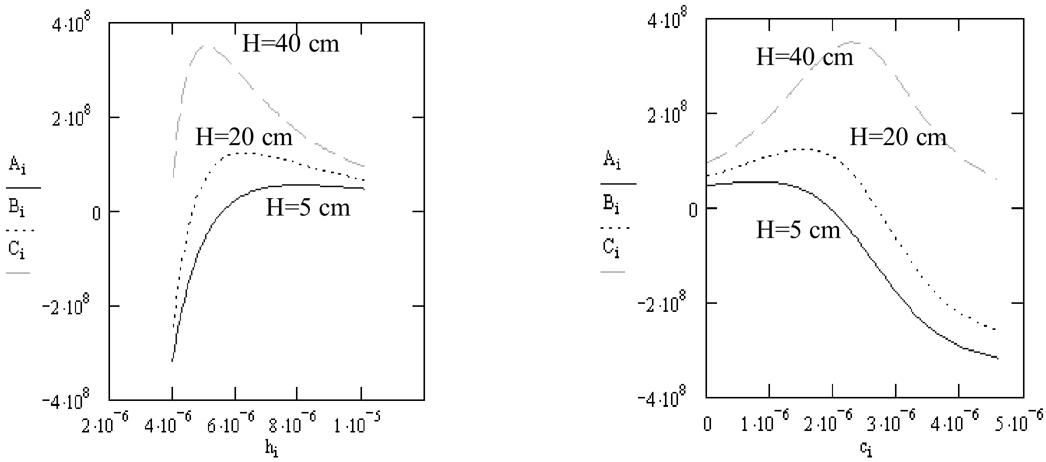

Fig. 2.4 *The instantaneous carrying force in relation to the dimensional thickness of the lubricant film (n=370 rot/min, p_{in}=0,5 bar, G=2250 N, h_{m0}=10,175 µm)*

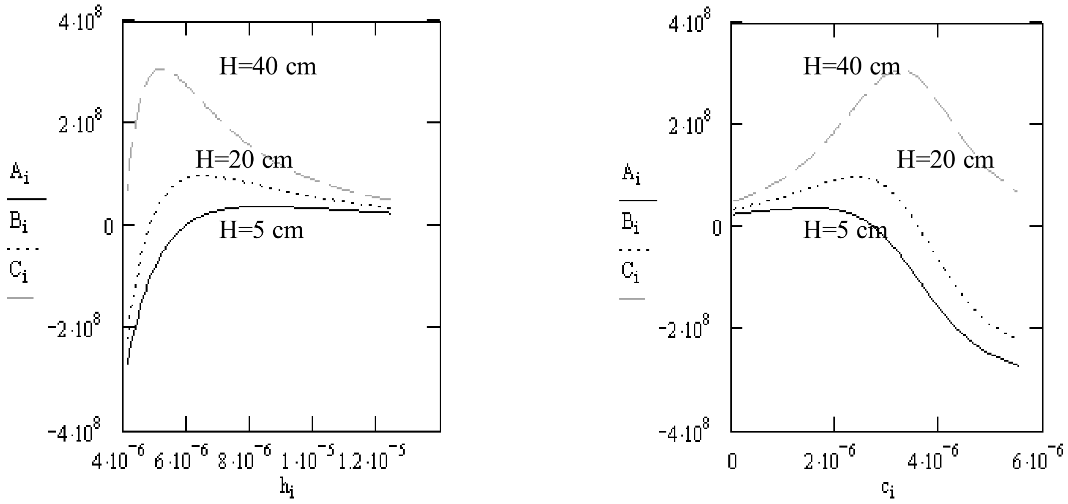

Fig. 2.5 *The instantaneous carrying force in relation to the dimensional thickness of the lubricant film (n=600 rot/min, p_{in}=1,5 bar, G=2250 N, h_{m0}=12,554 μm)*

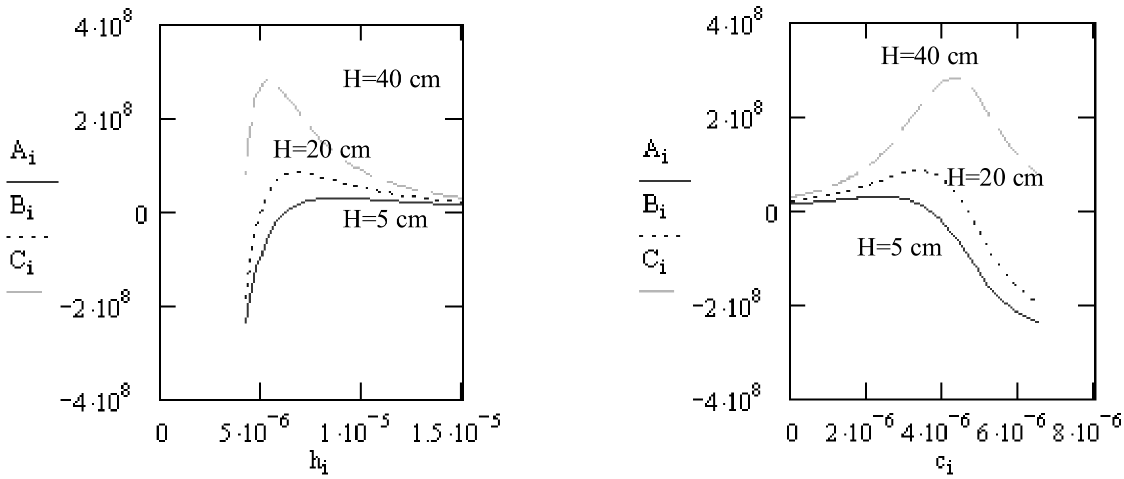

Fig. 2.6 *The instantaneous carrying force in relation to the dimensional thickness of the lubricant film (n=960 rot/min, p_{in}=8 bar, G=2250 N, h_{m0}=15,159 μm)*

3. CONCLUSIONS

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

- the insignificant influence of the feeding pressure on the minimum thickness of the lubricant for the same rotation of the spindle;
- 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 ratio of film thickness Hs_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;
- 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);
- the static charging conditions of the bearing does not have an important influence regarding the changing in the pressure's values, as the static charging conditions gets bigger, so as the dynamic pressure is bigger.

4. REFERENCES

[1] ALEXANDRESCU, I. M., PAY, E., BUCHMAN, A., **Experimental results regarding hard shocks behaviour to the radial hydrodynamic working bearings**. microCAD International Scientific Conference. Machine and Construction Design, University of Miskolc, 2005, pp. 1-12, ISBN 963-661-646-9

[2] ALEXANDRESCU, I. M. - PAY, E. - PASCOVICI, M. D.: **The effect of lubricant film expulsion under shock in the case of the narrow sliding radial bearing (L/D ≤ 0,6)**. Scientific Bulletin, Serie C, Vol. XIX. Fascicle: Mechanics, Tribology, Machine Manufacturing Technology. Part I. International Multidisciplinary Conference, 6-th Edition, North University of Baia Mare, 2005, pp. 1-6, ISSN 1224-3264, ISBN 973-87237-1-X

[3] ALEXANDRESCU, I. M.: **Studiul comportării lagărelor radiale cu ungere hidrodinamică în condițiile funcționării cu şocuri şi vibrații**. Teza de doctorat, Universitatea Tehnică Cluj-Napoca, 2005, Conducător ştiințific: prof. dr. ing. Eugen PAY Dr. H.C.

[4] ALEXANDRESCU M., COTETIU, R., UNGUREANU, N.,: **Theoretical correlations regarding the instantaneous carrying force in the case of the narrow sliding radial HD working bearing under hard shocks**. Automation in Production Planning and Manufacturing. 8th International Scientific Conference, 02 – 04 May 2007, Žilina – Valča, Slovak Republic, ISBN 978-80-89276-03-5, pag. 9-13.

[5] BOWDEN, F. P., TABOR, D., **The Friction and Lubrication of Solids**. Oxford at the Claderendon Press, 1950, pp. 259-283

[6] KHONSARI, M. M., BOOSER, E. R., **Applied Tribology**. Bearing Design and Lubrication. John Wiley & Sons, INC., New York, 2001

[7] MOORE, D. F., **Viscoelastic Machine Elements. Elastomers and lubricants in machine system**. Oxford, Butterworth-Heinemann Ltd., 1993, pp. 139-163