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EXPERIMENTAL RESEARCH CONCERNING THE RADIAL BEARINGS WITH HD LUBRICATION IN THE CASE OF HUGE CHALLENGING WORKING

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Abstract: *This paper presents a few experimental assessment concerning the function of radial bearings with HD lubrication in the case of huge challenging working. The experimentation methodology includes some practical experiments met during the working process of these bearings. It was focussed on the determination of the lubrifiant film resistivity, bearing's acceleration in dynamic charging conditions and pressure distribution from the film to be lubricated in various places of the bearing's body. It is showed the details during the measuring accomplishments and the experimental results are registered in a record of obtained results.*

Key words: *pressure distribution, radial hydrodynamic bearing, impulse loading, lubrifiant film resistivity.*

1. INTRODUCTION

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.

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).

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

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

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

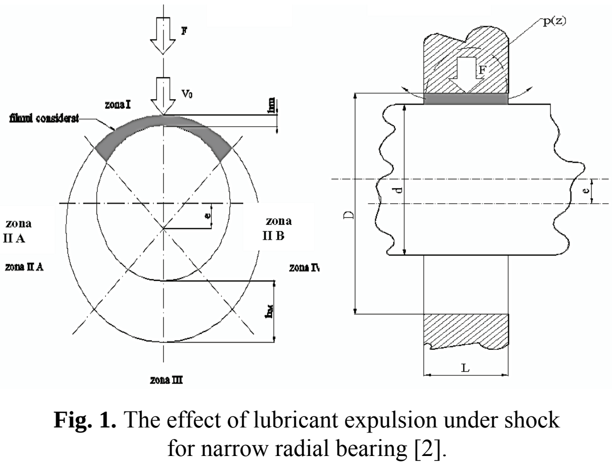

Fig. 1. The effect of lubricant expulsion under shock for narrow radial bearing [2].

The simplified modelling of the lubricant film thickness and carriage under the conditions of a closing motion of the spindle and bushing surfaces for the narrow radial bearing exposed to shocks (Fig. 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.

Taking into consideration the Stribeck curves, that can be applicable to sliding bearings it has been found that the minimum thickness of the lubrifiant film and the rubbing value can be modified depending on the challenging working, speed and oil dynamic viscosity [1].

În the case of sliding bearings the lubrifiant working conditions corresponding to $h_{\min}$ is imposed by rugosity; the medium height of the roughness can be reckoned for the spindle $R_{1\ \max} \cong 5$ μm; as for the bronze bushing $R_{2\ \max} \cong 5$ μm is to be considered [3]. So, in the case of fluid rubbing functioning conditions, the minimum lubrifiant thickness has to be bigger than an allowable value, $h_{\min,a} \geq 10$ μm.

2. EXPERIMENTAL DEVICES AND ACQUISITION CHAINS

The assessment was made on the experimental stand of the Tribology and Manufactural Engines Lab from the North University of Baia Mare (Fig. 2), making use of the modern technology concerning the results'processing and acquisition [1].

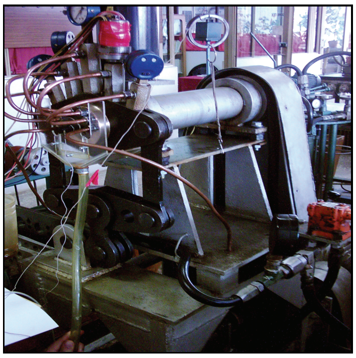

Fig. 2. The testing experimental devices.

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.

The dynamic loading of the bearing is made through the launcing 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 $F_1 = 1\,665$ N, for $H = 20$ cm we have $F_2 = 2\,356$ N, and for $H = 40$ cm, we have $F_3 = 3\,332$ N. The static working conditions is presented for the following value $H = 0$ cm.

All the tests were made at a 40°C of the lubrifiant, being constant, pressure distribution p_{in} having the following values, from 0.5 bar to 10 bar [1].

Using a lubrifiant oil for bearings of LA 32 STR 5152-89 type, with the viscosity of 31.3 cSt at 40°C, it was focussed on the determination pressure distribution from the film to be lubricated in various places of the bearing's body, with the help of pressure measuring dose with tensiometric translators put together through an amplifier placed at the acquisition plate ADuC 812.

Fig. 3 presents the bushing diagram in experimental assessments having $L/D = 0.5$.

Those 4 tensometric stamps are conected in a tensometric bridge diagram, being related by an amplifier at the acquisition plate ADuC 812 [5].

Fig. 4 presents the pressure measuring chain in the lubrifiant film.

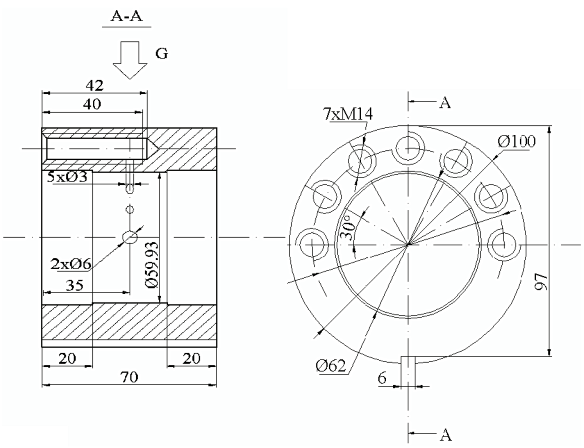

Fig. 3. The bushing of the experimental radial bearing.

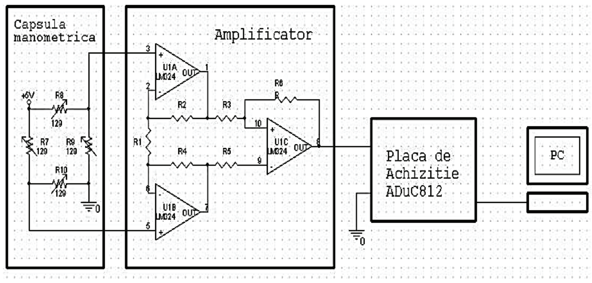

Fig 4. The pressure measuring chain in the lubrifiant film.

The acceleration of the moving bearing because of the shock, it determined with the help of acceleration sensor ADXL 190 WQC whose signal was acquisitioned with data acquisition system ADuC 812 (Fig. 5).

The exit signal of the senzor is the electric voltage, for 250 mV there is an acceleration of 9.8 m/s^2 [1].

The signal taken by the data acquisition system ADuC 812 have been analised by PC with the help of MATLAB 6.5.0.18091 3a program.

Fig. 6 presents the the ADXL 190 WQC acceleration senzor and the available supply source.

The lubricant film resistivity, it determined through the achievement of a circuit between spindle and bushing which include a standard resistance $R_{12} = 49$ kΩ [4].

Fig. 7 presents the lubricant film resistivity measuring chain.

ADuC 812 (Fig. 9), produce by Analog Device, has an 8051-compatible microcontroler core supported by 8kb Flash/EE program memory [5].

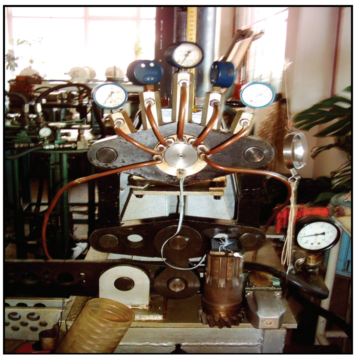

Fig. 5. Experimental stand with the ADXL 190 WQC acceleration senzor.

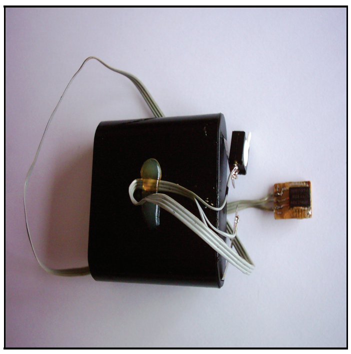

Fig. 6. The ADXL 190 WQC acceleration senzor and the available supply source.

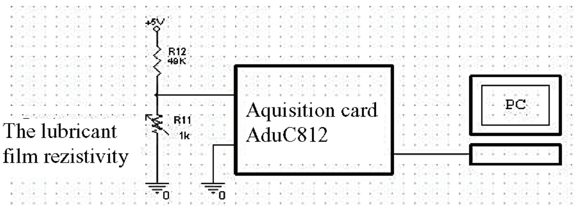

Fig. 7. The lubricant film resistivity measuring chain between spindle and bushing.

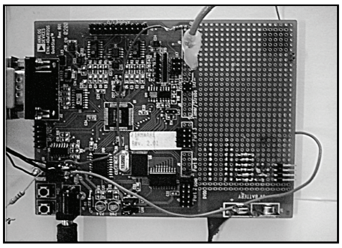

Fig. 9. The acquisition plate ADuC 812 [5].

3. EXPERIMENTAL RESULTS

The pressure distribution on the peripheric side of the bushing, depending on the available supply pressure, the static and dynamic charging conditions at different spindle's rotations are presented in Fig. 10, for $n = 370$ rot/min, $p_{in} = 0.5$ bar, and Fig. 11 for $n = 600$ rot/min, $p_{in} = 1.5$ bar.

Fig. 12 presents the bushing acceleration from the moment of shock for each position P1-P5 from the periphery of the bushing at the following rotation $n = 370$ rot/min. In a similar way (Fig. 13), the case of shock at $n = 600$ rot/min, $p_{in} = 1.5$ bar [1].

Minimal resistance on the lubricant film, depending on the available supply pressure, the static and dynamic charging conditions, are presented in Fig. 14, for $n = 370$ rot/min, $p_{in} = 0.5$ bar, and Fig. 15 for $n = 600$ rot/min, $p_{in} = 1.5$ bar.

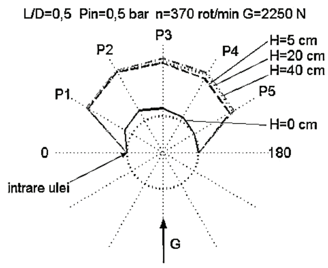

Fig 10. The dynamic pressure distribution on the peripheric side of the bushing (n = 370 rot/min, p_{in} = = 0.5 bar).

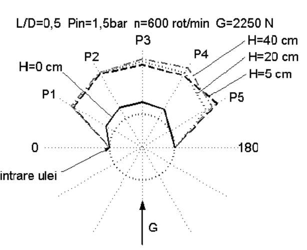

Fig 11. The dynamic pressure distribution on the peripheric side of the bushing ($n = 600$ rot/min, p_{in} = = 1.5 bar).

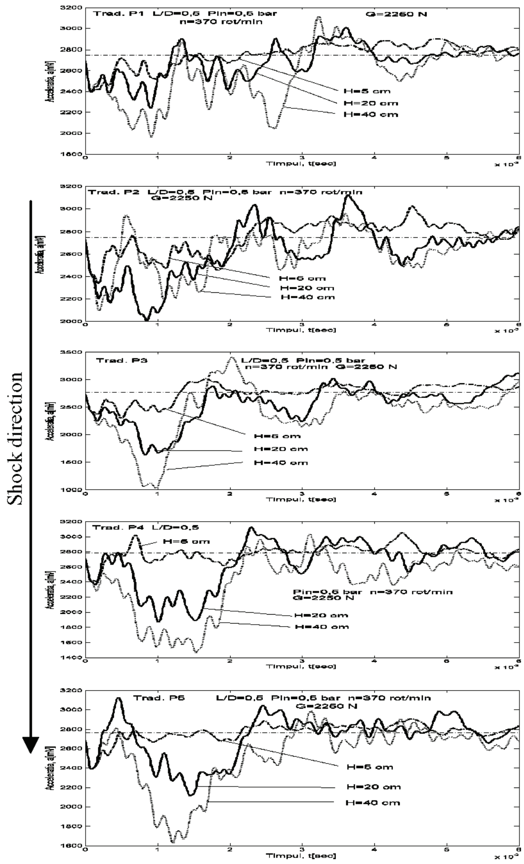

Fig. 12. The bushing acceleration in the moment of shock at $n = 370$ rot/min, $p_{in} = 0.5$ bar [1].

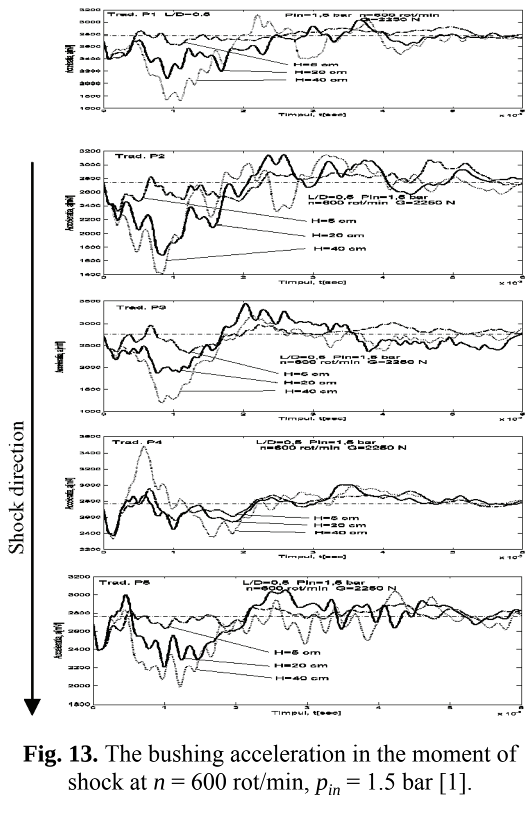

Fig. 13. The bushing acceleration in the moment of shock at $n = 600$ rot/min, $p_{in} = 1.5$ bar [1].

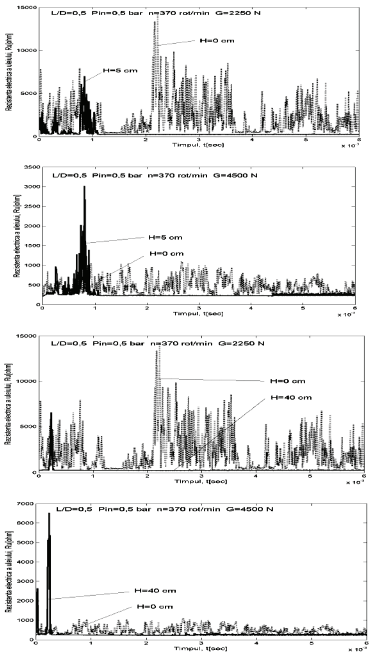

Fig. 14. The lubricant film resistivity for $n = 370$ rot/min, $p_{in} = 0.5$ bar, depending on the static and dynamic charging conditions.

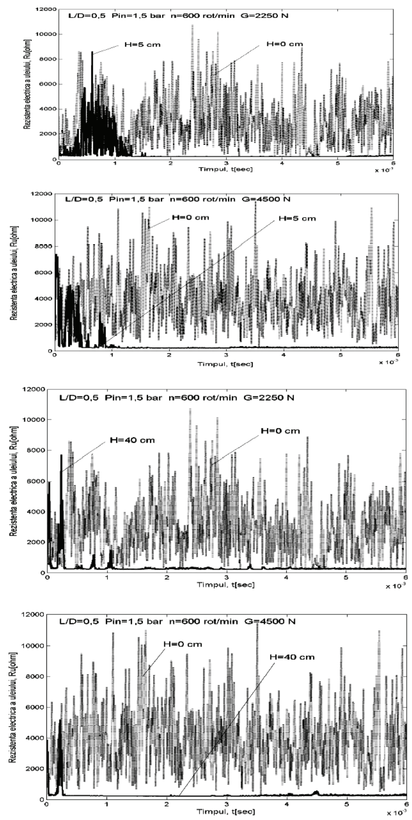

Fig. 15. The lubricant film resistivity for $n = 600$ rot/min, $p_{in} = 1.5$ bar, depending on the static and dynamic charging conditions.

4. CONCLUSIONS

The dynamic pressure from the moment of shock is increased when increasing the dynamic charging conditions; this incresing process refers to the all portant zone, the dynamic pressure having values from 3.37 to 118 static pressure, depending on the studied position of the peripheric zone of the bushing;

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 shock amplitude is bigger for the maximum position of the static pressure from the spindle, so: for the following rotations $n = 370$ rot/min and $n = 600$ rot/min this maximum is the same with P_3 position from the spindle;

Maintaining the constant dynamic charging, at the same time as the static charging increasing, the shock taken by the bushing – lubrifiant film – spindle system becomes lower (the lubrifiant film thickness in the case of static functioning becomes lower, so as the absorbtion of the shock made by the lubrifiant film becomes lower);

The mean value of the lubricant film resistivity, which estimate the lubricant film's minimum thickness, at revolutions $n = 370$ rot/min, it's approximate three times bigger in the static case, than in dynamic case, for the static change $G_1 = 2\,250$ N, respective two times bigger in the static case, than in dynamic case, for the static change $G_2 = 4\,500$ N; to be noticed the sudden decrease of lubricant film's thickness in area wich corresponds to the shock's time.

REFERENCES

[1] Alexandrescu, I. M., Pay, E., Buchman, A. (2005). *Experimental results regarding hard shocks behaviour to the radial hydrodynamic working bearings*, MicroCAD International Scientific Conference Machine and Construction Design, University of Miskolc, pp. 1–12, ISBN 963-661-646-9.

[2] Alexandrescu, I. M., Pay, E., Pascovici, M. D. (2005). *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, pp. 1–6, ISSN 1224-3264, ISBN 973-87237-1-X.

[3] Pay, E., Alexandrescu, I. M. (2004). *Searches Regarding Radial Journal Bearings with HD Lubrication*, Romanian Journal of Technical Sciences Applied Mechanics, Tome 49 (Special Number), International Conference on Manufacturing Systems (ICMaS 2004), Bucharest, Editura Academiei Române, ISSN 0035-4074, ISBN 973-27-1102-7.

[4] *** (2001). *National Instruments*, Data Acquisition Product Guide.

[5] *** (2003). *ADuC 812 MicroConverter*, Analog Devices Inc.

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