

# Correlations of the instantaneous carrying force and pressure distribution in the case of the narrow sliding radial bearing under hard shocks\*

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**Abstract.** We present the determining relationship of carriage in non-dimensional form for narrow radial bearings subjected to shocks and vibrations, as well as the determining relationships of the lubricant minimum thickness in relation to the dynamic loading. Due to the very short time of loading radial bearings subjected 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 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. The paper is focused on determining relationship of carriage in dimensional form for narrow radial bearings exposed to shocks and vibrations, as well as the determining relationships of the 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.

## 1. Introduction

The study of the behavior of radial bearings with hydrodynamic lubrication, functioning under conditions of shocks and vibrations, is carried out from a tribological point of view, observing the aspect of friction and lubrication, the lubricant film, by which the shock is damped [1]. In the case of bearings subjected 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 [2].

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 [1].

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].

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The modeling 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 ( $V = -\frac{\partial h}{\partial t}$ ) [3].

Analytical expression of the Reynolds equation corresponding to this study, with an isothermal approach is [4]:

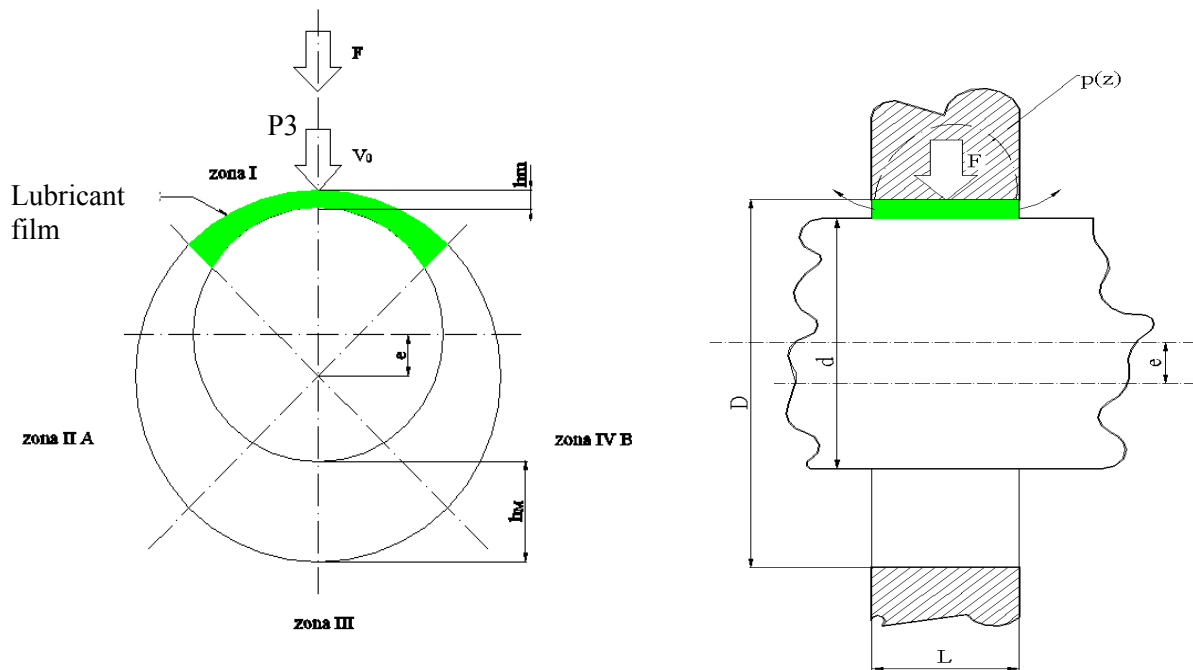
$$\frac{\partial}{\partial x} \left( h^3 \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial z} \left( h^3 \frac{\partial p}{\partial z} \right) = 12\eta \frac{\partial h}{\partial t} . \quad (1)$$

where  $\eta$ - viscosity of lubricant (Ns/m<sup>2</sup>);  $p$ -pressure (Pa);  $h$ - fluid film thickness (m).

Nomenclature:

$L$ - length of bearing (m);  $\eta$ - viscosity of lubricant (Ns/m<sup>2</sup>);  $G$ - static loading (N);  $p$ -pressure (Pa);  $F$ - dynamic load (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 schematic of a narrow hydrodynamic radial bearing with circular bushing subjected to shocks, modeled in 4 areas, is presented in figure 1 [1].



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

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 IV 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 [5].

The circumferential pressure distribution is [6]:

$$p(\theta) = \frac{12\eta VB^2}{J^3(1 - \varepsilon \cos \theta)^3} \quad (2)$$

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

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

The relative eccentricity given by

$$\varepsilon(t) = 1 - \frac{2h_m(t)}{J} \quad (4)$$

and

$$h_m = \frac{1}{\sqrt{\frac{1}{h_{m0}^2} + \frac{8F\sqrt{2gH}}{\eta \pi D L^3 g}}} \quad (5)$$

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

$$F^* = 2 \int_0^{\pi/2} \frac{4\eta V D L^3 \cos^2 \theta \cdot d\theta}{J^3(1 - \varepsilon \cos \theta)^3} \quad (6)$$

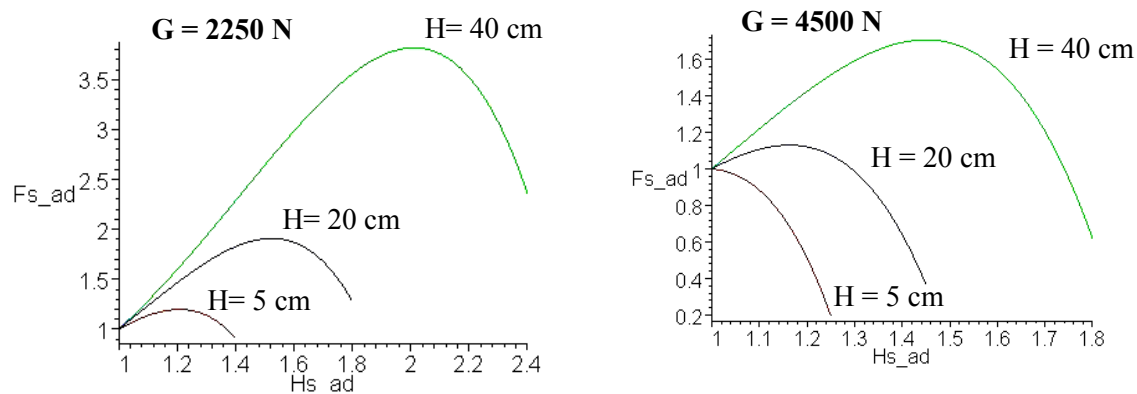
We can write

$$\bar{F}_s = \frac{1}{A} [\bar{H}_s^3(1 + A) - \bar{H}_s^5] \quad (7)$$

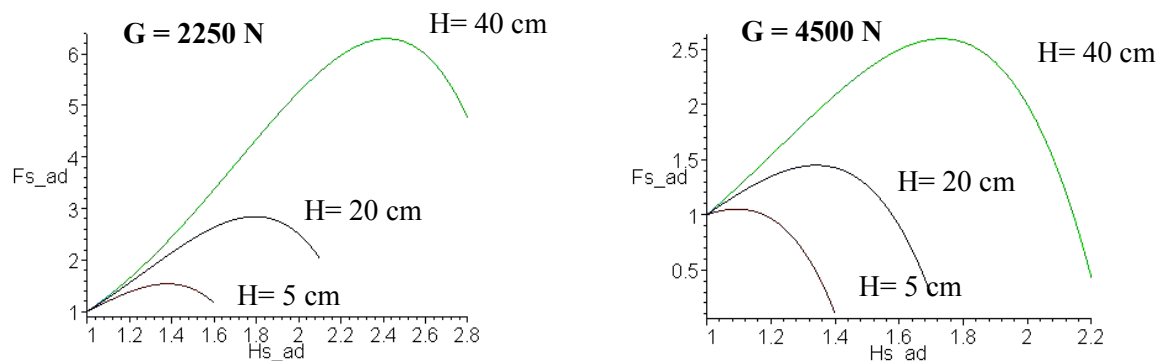
where  $A = 4\bar{F}\Pi$ ,  $\bar{H}_s = \frac{h_{m0}}{h_m} = H_{s-ad}$  and the parameters of lubricant film expulsion  $\Pi$  have the expression  $\Pi = \frac{H}{h_{m0}}$  ( $H$  being the height from which the weight dynamically loading the bearing is launched) [6].

## 2. Theoretical results

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 figure 2 and figure 3.

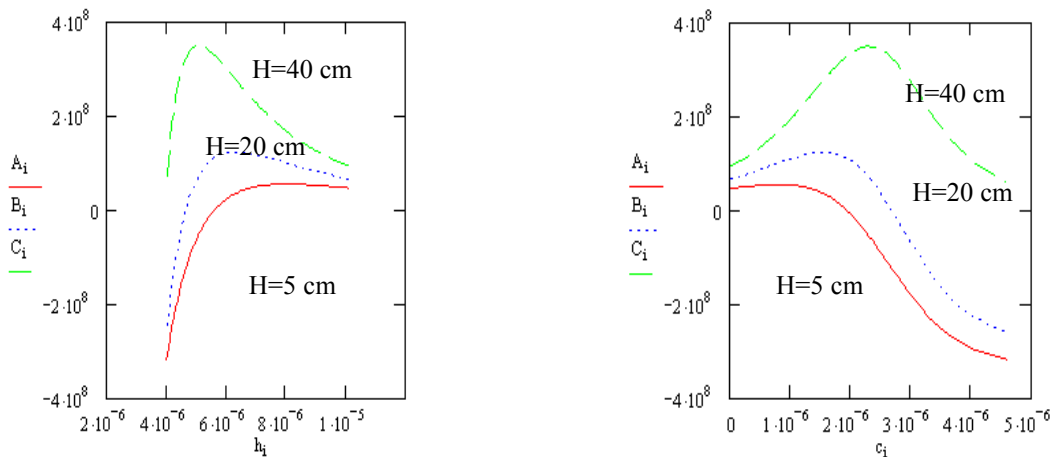


**Figure 2.** The instantaneous carrying force in relation to the non-dimensional thickness of the lubricant film ( $n=370$  rot/min,  $p_{in}=0.5$  bar).

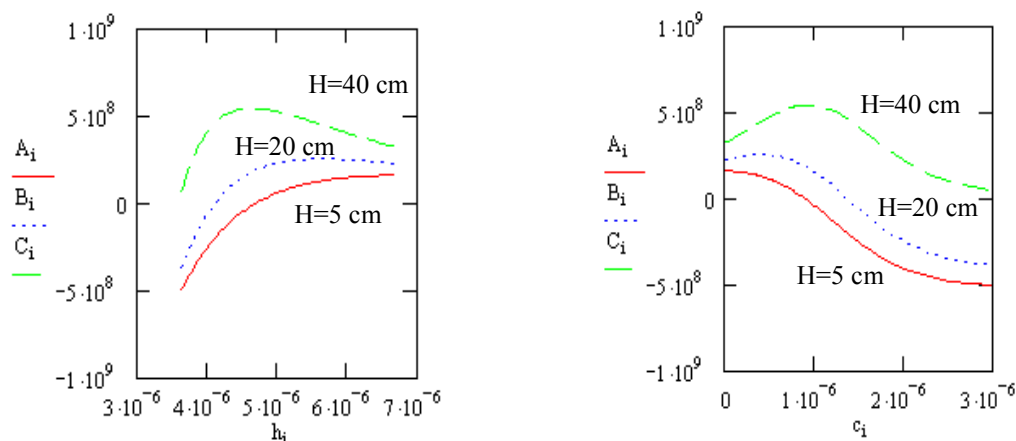


**Figure 3.** The instantaneous carrying force in relation to the non-dimensional thickness of the lubricant film ( $n=600$  rot/min,  $p_{in}=1.5$  bar).

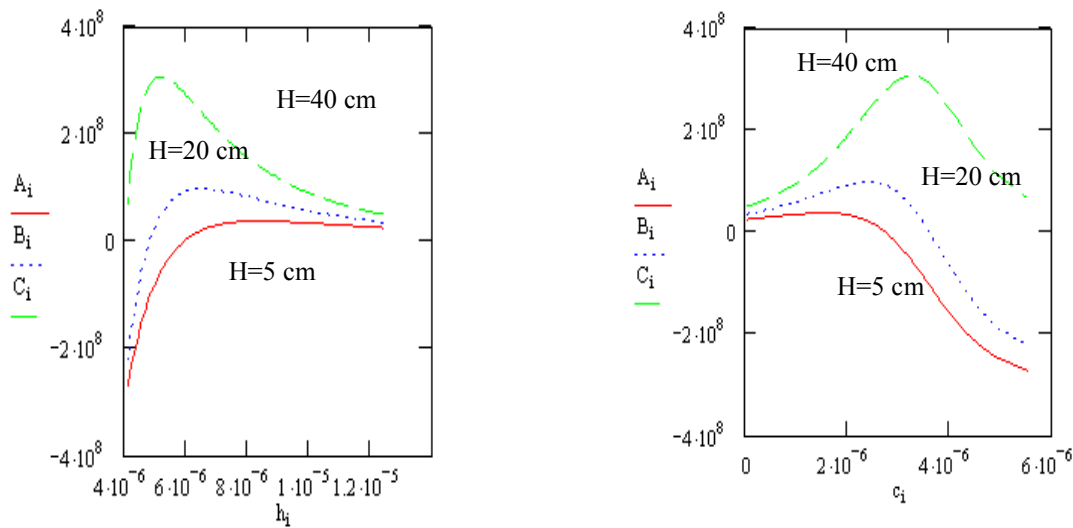
The variations of the instantaneous carrying force, in relation to the dimensional thickness of the lubricant film and in relation to the time of shock, for the three weight launching heights  $H$  are presented in figure 4 – 7.



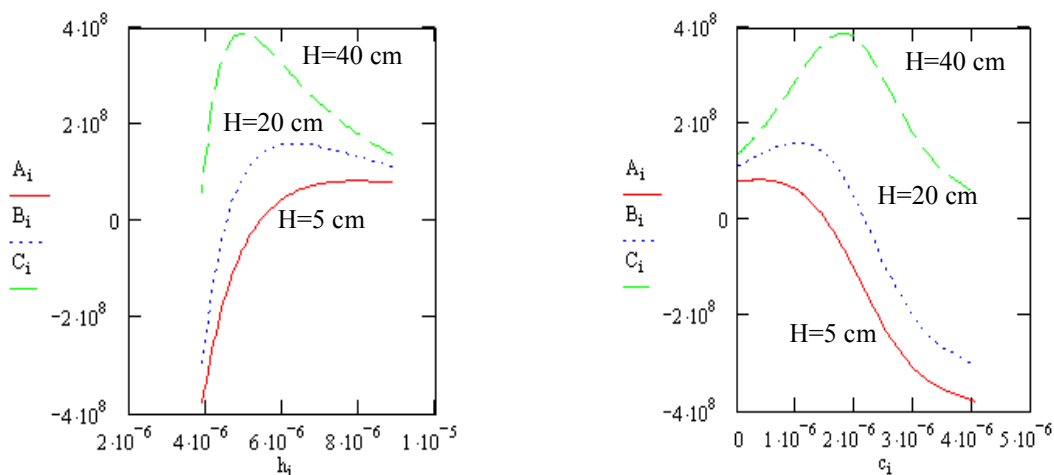
**Figure 4.** The instantaneous carrying force in relation to the dimensional thickness of the lubricant film ( $n=370$  rot/min,  $p_{in}=0.5$  bar,  $G=2250$  N,  $h_{m0}=10.175$   $\mu\text{m}$ ).



**Figure 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,  $h_{m0}=6.723$   $\mu\text{m}$ ).



**Figure 6.** 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,  $h_{m0}=12.554$   $\mu$ m).



**Figure 7.** The instantaneous carrying force in relation to the dimensional thickness of the lubricant film ( $n=600$  rot/min,  $p_{in}=1.5$  bar,  $G=4500$  N,  $h_{m0}=8.493$   $\mu$ m).

### 3. Experimental devices and acquisition chains

The experimental research was carried out 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 assessment was made on the experimental stand of the Tribology Engines Lab from the Technical University of Cluj-Napoca, North University Center of Baia Mare, making use of the modern technology concerning the results' processing and acquisition [1].

The dynamic loading of the bearing is achieved through the launching of a weight which hits the bearing at different heights. Assessments for heights between 5 and 40 cm were carried out, 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. The static working conditions is presented for the following value  $H=0$  cm [7].

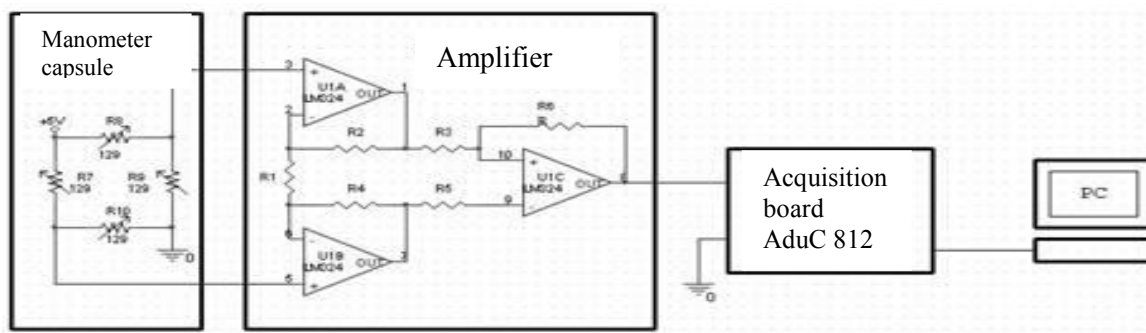
All the tests were conducted at a 40°C of the lubricant, being constant, pressure distribution  $p_{in}$  having the following values, from 0.5 bar to 10 bar.

Using a lubricant oil for bearings of LA 32 STR 5152-89 type, with the viscosity of 31.3 cSt at 40°C, it was focused 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 tensometric translators put together through an amplifier placed at the acquisition plate ADuC 812 [8].

In dynamic loading conditions, 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 transducers.

Those 4 strain gages are connected in a tensometric bridge and connected to an amplifier at the acquisition plate ADuC 812 [9].

Figure 8 presents the pressure measuring chain in the lubricant film [10].

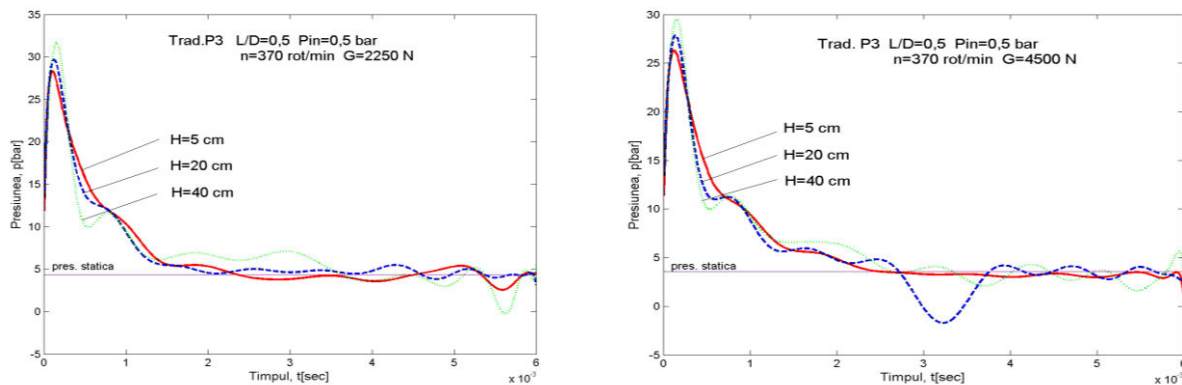


**Figure 8.** The pressure measuring chain in the lubricant film.

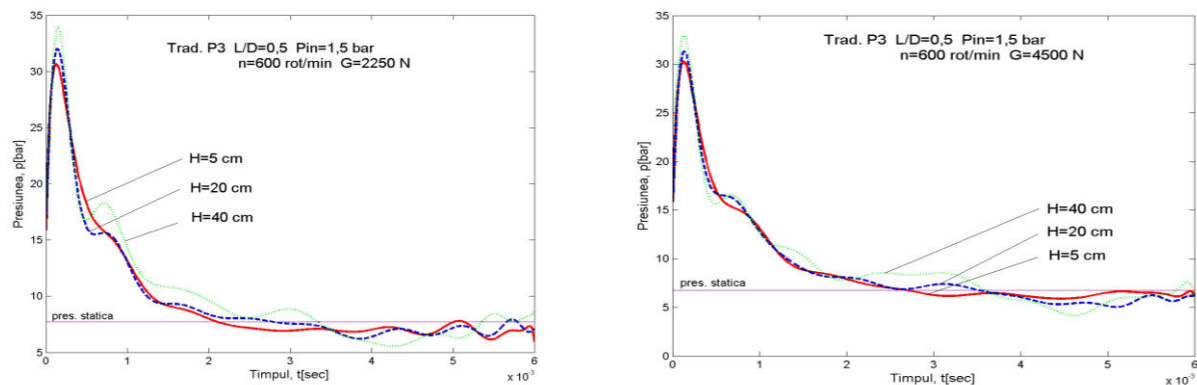
The pressure was measured in the case of dynamic charging, on the manometer's exhibition stand, using the above chain, focusing on the variation exit sign and registering amplifier and the acquisition plate ADuC 812. The pressure increase was made bar by bar, the dose distortion being linear with the pressure. It was established the dependency relation between the pressure and the tension in the exit point in mV (2.3 mV = 1bar  $\Delta p$ ) [11].

#### 4. Experimental results

The pressure distribution at points P3 of the bearing's body, which corresponds to the direction of charging application, depending on the available supply pressure, the static and dynamic charging conditions at different spindle's rotations are presented in figure 9 for  $n=370$  rot/min,  $p_{in}=0.5$  bar, and figure 10, for  $n=600$  rot/min,  $p_{in}=1.5$  bar.



**Figure 9.** The dynamic pressure distribution depending on the static and dynamic charging conditions of the bearing ( $n=370$  rot/min,  $p_{in}=0.5$  bar).



**Figure 10.** The dynamic pressure distribution depending on the static and dynamic charging conditions of the bearing ( $n=600$  rot/min,  $p_{in}=1.5$  bar).

## 5. Conclusions

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

- 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;
- 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 decrease of the lubricant film minimum thickness along with the increase of static loading;
- 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 dynamic pressure from the moment of shock is increased when increasing the dynamic charging conditions; for the studied position, P3, the dynamic pressure rise at the same time with the rise of dynamic charging, the pressure leap being between 5.95 and 7.45 multiplied with static pressure for the HD bearing with  $L/D=0.5$ ;
- 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;
- the draught's pressure in dynamic conditions has a slightly shifting to the entrance zone of the lubricant when static charging conditions are increasing;
- for the studied P3 position, at the same time with the rise of static charging, the static pressure decrease.

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