TRIBOLOGICAL ANALYSIS OF A RADIAL PLAIN BEARING

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Abstract: The paper presents experimental determination of the friction coefficient variation at a radial plain bearing, depending on the values of the radial force R and the angular speed ω of the spindle at a constant lubricant temperature value.
Keywords: Tribology, radial plain bearing, friction coefficient.

1. Introduction

The tribological analysis of radial plain bearings (fig.1), lubricated in hydrodynamic regime, aims to determine the friction coefficient and the energy losses that cause the mechanical efficiency, heating temperature and wear intensity.

The friction coefficient is influenced by: the size of the bearing, the quantity and quality of the lubricant, the roughness of the contact surfaces (between the spindle and the inner ring of the bearing), the assembling precision, the magnitude of the load, the rotation speed and the temperature of the lubricant [6], [10], [11].

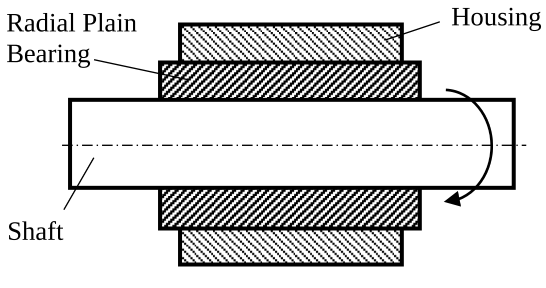

Fig. 1. Radial plain bearing.

The energy losses imply the expression of the friction moment and the power lost by friction depending on of the radial force and the relative angular speed of the shaft axis in relation to the fixed part of the bearing.

Plain bearings were analyzed in various conditions, from the tribological point of view [3], [5].

Numerical simulation studies were performed regarding the influence of clearance on stress state and contact pressure in plain bearings in [4], and regarding the friction forces that occur in different types of kinematical joints of mechanical systems in [7], [9], [12].

The paper shows the experimental determination of the variation of the experimental friction coefficient at a radial plain bearing, depending on the variation of the radial force, and of the shaft speed at a predetermined value of the lubricant temperature of 50 [˚].

2. Theoretical Approach

In a radial plain bearing with hydrodynamic lubrication, the theoretical friction coefficient is determined with the relations [1], [2], [8].:

$$\mu_t = 3{,}2 \cdot \eta \cdot \omega \cdot 10^{-3} / p \cdot \psi + 0{,}55 \cdot \psi \qquad (1)$$

where:

- η [Ns/m^2] is the dynamic viscosity of the lubricant, η = 0,11 [Ns/m^2] ;
- $\psi = (D - d) / d$ – the relative clearance between the inner diameter of the bearing D [mm] and the spindle of the shaft with the outer diameter d [mm], D = d = 30 [mm];
- l [mm] – contact length between spindle and bearing, l = 15 [mm];
- ω [s^{-1}] – spindle angular speed;
- p [MPa] – the average pressure between the spindle and the bearing, which is calculated with the following relation:

$$p = R / l \cdot d \text{ [MPa]} \qquad (2)$$

- R [N] is the radial force in the bearing.

The radial force R in the bearing is achieved by loading the system with levers with the force G, placing different weights on the plate of the system, and it has the following expression [13]:

$$R = G_0 + 5 \cdot G = 25 + 5 \cdot G \text{ [N]}, \quad (3)$$

where:
- $G_0 = 25$ [N];
- $G = 10; 20; 30; 40; 50$ [N].

The experimental friction coefficient in the bearing is calculated depending on the friction force F_f [N] and the radial force R [N], with the relation:

$$\mu_e = F_f / R \text{ [-]} \quad (4)$$

The friction moment and the friction force in the radial plain bearing are determined the with following relations [13]:

$$M_f = F_f \cdot \frac{d}{2} = F_0 \cdot a = 1 \cdot a \text{ [N mm]} \quad (5)$$

$$F_f = 2 \cdot M_f / d \text{ [N]} \quad (6)$$

where: a – is the length of the force arm $F_0 = 1$ [N], which is measured on the graded ruler of the balance scale, for the horizontal markings to coincide.

3. The Testing Stand

Fig. 2 shows the components of the experimental stand used for testing the radial plain bearing.

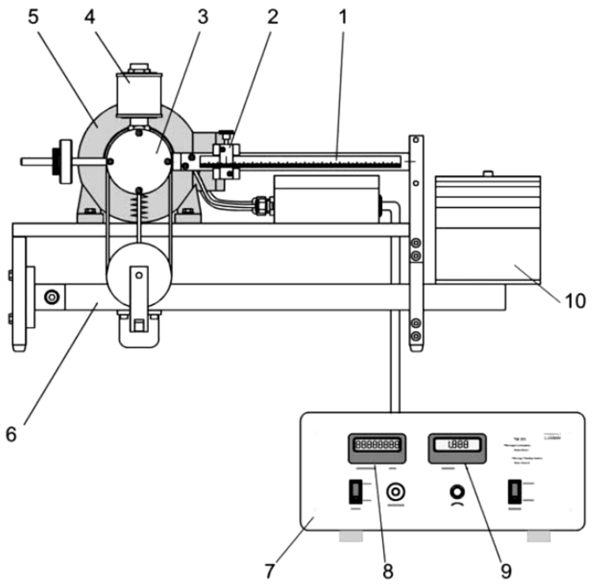

Fig. 2 Scheme of the testing stand [13].

The experimental stand allows the determination of the energy losses from a radial plain bearing, depending on the radial force and the variation of the spindle speed.

The stand is actuated by an asynchronous electric motor with a frequency converter (5), the speed regulation of the radial plain bearing spindle being carried out by means of a speed regulator.

The values of the shaft rotation speed and of the temperature in the bearing are read on the displays no. (8) and (9) of the control system (7) [13].

The radial plain bearing is assembled in the oscillating cylindrical housing (3), on which the graded ruler (1), the slider (2) and the fine adjustment balance system are mounted. In the structure of the balancing, there are two trapezoidal belts mounted on the oscillating cylindrical housing, two belt pulleys with the related support on the lever (6) and a plate for the weights (10).

The radial force is applied on the bearing by placing the weights on the plate (10), for example G = 10; 20 [N], and the balancing of the friction moment will be achieved by moving the cursor (2) on the graded ruler (1) until the two horizontal marks (on the ruler and on the stand) align.

The lubrication of the radial plain bearing is hydrodynamic, being carried out automatically with the fine oil from the wick oiler (4) located in the upper part of the bearing housing. The oil in the bearing is directed via a pipe to the collector vessel, mounted under the oscillating cylindrical housing (3).

4. Testing Methodology

The testing steps are described as follows [13]:
- the radial force is applied on the bearing by placing a weight (G = 10 N) on the plate of the lever system;
- power is supplied to the experimental stand and it is started by setting the switch of the control system to position 1;
- adjusting the spindle rotation speed to the first speed step, n = 500 [rpm], by operating the potentiometer on the control system, and reading the bearing temperature [˚] and the cursor positioning distance "a" [mm] of the cursor (2) on the graded ruler (1) when the scale is in balance and the two horizontal marks overlap;
- changing the rotation speed to a new value from the series: n = 500, 1000, 1500, 2000, 2500 [rpm], and repeat the bearing test phases;
- changing the value of the radial force applied to the bearing by loading the weights G = 20; 30; 40; 50 [N] on the plate of the lever system and repeating the bearing testing for all preset rotation speed steps;
- the relations (1) – (6) are applied, for the parameters calculus of each test case.

5. Experimental Results

The testing was accomplished on the experimental stand presented in fig. 3.

The notations are the same as in chapter 2.

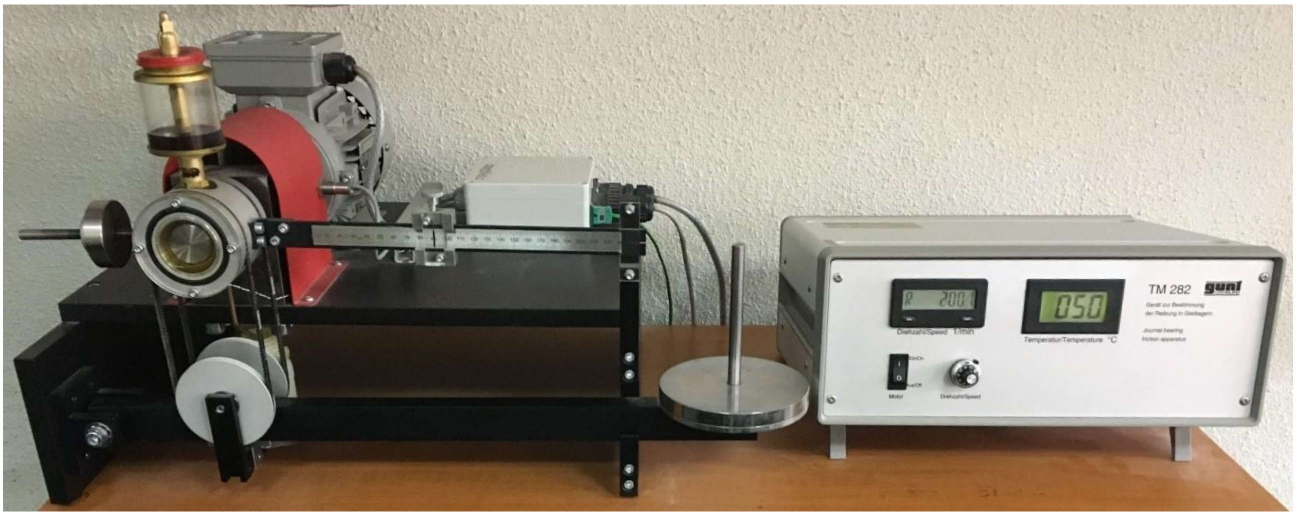

Fig. 3. The testing stand.

The experimental results are presented in Table 1.

Table 1. Experimental results.

Crt. no.	Shaft rotation speed n [rev/min]	Loading Force R_1 = 75 [N]		
		Distance a [mm]	Friction coefficient μ_e [-]	Friction moment M_f [N·mm]
1.	500	15	0.013	15
2.	1000	39	0.034	39
3.	1500	66	0.058	66
4.	2000	90	0.08	90
5.	2500	108	0.096	108
		Loading Force R_2 = 125 [N]		
6.	500	12	0.0064	12
7.	1000	41	0.021	41
8.	1500	75	0.04	75
9.	2000	98	0.052	98
10.	2500	108	0.057	108
		Loading Force R_3 = 175 [N]		
11.	500	-	-	-
12.	1000	36	0.013	36
13.	1500	65	0.024	65
14.	2000	85	0.032	85
15.	2500	110	0.041	110
		Loading Force R_4 = 225 [N]		
16.	500	-	-	-
17.	1000	35	0.010	35
18.	1500	70	0.020	70
19.	2000	103	0.030	103
20.	2500	115	0.034	115
		Loading Force R_5 = 275 [N]		
21.	500	-	-	-
22.	1000	35	0.0042	35
23.	1500	70	0.0084	70
24.	2000	95	0.011	95
25.	2500	115	0.013	115

The variation graphs of friction coefficient - radial force, and friction coefficient - shaft rotation speed, are presented in figures 4 and 5, respectively.

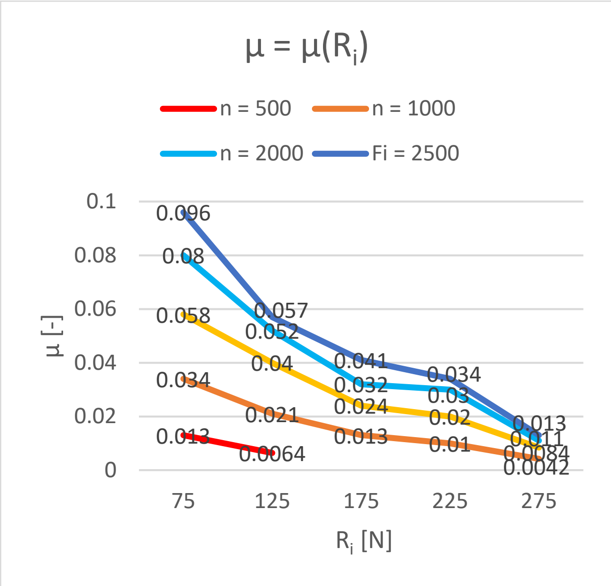

Fig. 4. The variation friction coefficient - radial force, μ = μ(Ri).

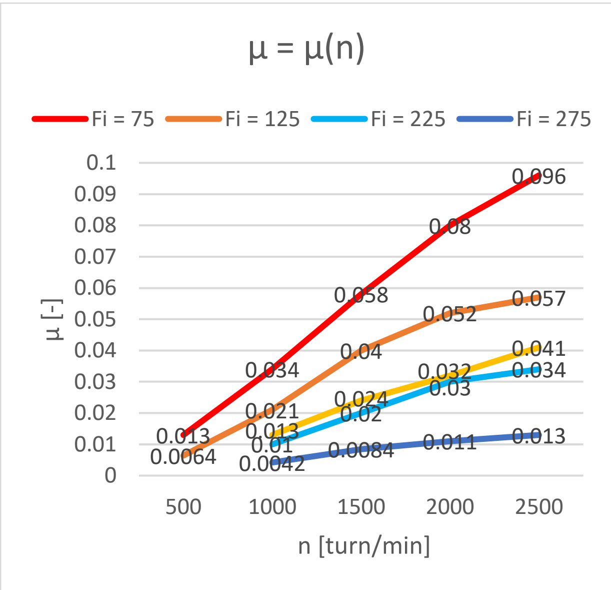

Fig. 5. The variation friction coefficient - shaft rotation speed, μ = μ(n).

5. Conclusions

Following the tests performed, at a constant temperature of the lubricant, it is observed that the increase of the radial force causes the decrease of the friction coefficient (fig. 4). Thus, between the values of 75 [N] – 275 [N] of the radial force, it is observed that the coefficient of friction is relatively constant.

Regarding the graph in fig. 5, the influence of the shaft rotation speed on the friction coefficient can be observed. Thus, the increase of the rotation speed determines the increase of the coefficient of friction.

As further research, the influence of temperature on the friction coefficient can be analyzed.

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