VARIATION OF THE FRICTION COEFFICIENT DURING THE OPERATION OF A RADIAL SLIDING BEARING DEPENDING ON LUBRICANT TEMPERATURE

Daniel-Gheorghe Vela [1], Ion Vela [2], Calin-Octavian Miclosina [1]*

[1] Babeș-Bolyai University, Faculty of Engineering, Department of Engineering Science
Traian Vuia Square, no. 1-4, 320085 Reşiţa, Romania
[2] Technical Sciences Academy of Romania, Petroșani Branch
University Street, No. 20, 332006 Petroșani, Romania
* Corresponding author. E-mail: calin.miclosina@ubbcluj.ro

Abstract: The paper presents a theoretical analysis and experimental tests regarding the variation of the friction coefficient in a radial sliding bearing depending on lubricant temperature changes. The values of the radial force and the rotation speed of the spindle are kept constant.
Keywords: Radial sliding bearing, variation of friction coefficient, lubricant temperature.

1. Introduction

Friction directly influences the operation of mechanical systems, causing wear of components or even their blocking.

In order to reduce the frictional forces, machine parts called bearings are used, which, depending on the constructive form, can be rolling bearings and sliding (or plain) bearings.

Depending on the direction of force application, the bearings can be radial, axial, and radial-axial.

Sliding bearings operate in fluid lubrication regime (hydrostatic or hydrodynamic) when a continuous layer of lubricant is provided between the contact surfaces of the conjugated parts in relative motion.

The scheme of a radial sliding bearing is presented in fig. 1. The design of radial sliding bearing with hydrostatically suspended pads is presented in [11], and the thermohydrodynamic analysis is studied in [9].

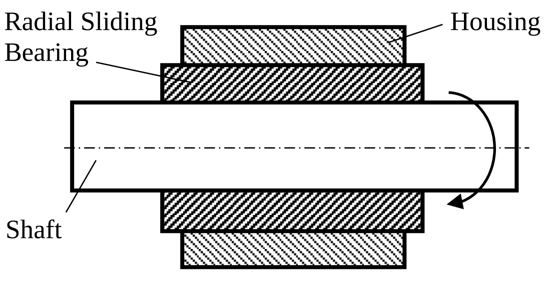

Fig. 1. Radial sliding bearing [15].

The characteristic tribological parameter is the friction coefficient, which influences directly the mechanical efficiency, temperature, and wear intensity.

The friction coefficient is influenced by the following factors: the size of the bearing, the quantity and quality of the lubricant, the roughness of the contact surfaces, the mounting precision, the load value, the rotation speed and the lubricant temperature [5], [10], [12], [15].

Sliding bearings were studied in different tribological situations [2], [4], [14], [15].

The influence of clearance on stress state and contact pressure in plain bearings was analyzed by numerical simulation in [3].

The friction forces that occur in usual types of kinematical joints were studied in [7], [8], [13].

The paper presents the research on the variation of the friction coefficient in the sliding bearing depending on lubricant temperature changes (35°, 40°, 45°, 50°, 55°, 60°). The testing was performed for certain rotation speeds of the shaft, loaded with constant radial forces.

2. Basic Theoretical Notions

The theoretical friction coefficient in a radial sliding bearing with hydrodynamic lubrication is computed with the formula [1], [6]:

$$\mu_t = 3{,}2\cdot\eta\cdot\omega\cdot10^{-6} / p\cdot\psi + 0{,}55\cdot\psi =$$
$$= 3{,}2\cdot0.018\cdot52.36\cdot10^{-6} / 869\cdot0.0035 + 0{,}55\cdot0.0035 =$$
$$= 0.011 \quad (1)$$

where:

- η [$N\cdot s/m^2$] is the dynamic viscosity of the lubricant; for an HLP ISO 32 oil [17], it can be calculated by using the relation [18]:
  $\eta = \nu\cdot\rho = 21\cdot10^{-6}\cdot869 = 0{,}018$ [$N\cdot s/m^2$] (2)
  where:
  $\nu = 21$ [cSt] $= 21\cdot10^{-6}$ [m^2/s] – is the kinematic viscosity of an ISO 32 oil at 50 [°C] [18];
  $\rho = 869$ [kg/m^3] – is the oil density [19]
- ω [s^{-1}] – spindle angular speed;
  $\omega = 2\cdot\pi\cdot n/60 = 2\cdot\pi\cdot500/60 = 52.36$ [s^{-1}] (3)

$n = 500$ [rot/min] – is the rotation speed;

- p [MPa] – the average pressure between the spindle and the bearing, computed by using the formula [7]:

  $p = R / (l \cdot d) = 125/(45 \cdot 30) =$
  $= 0.093$ [N/mm^2] $= 0.093$ [MPa] (4)

  R = 125 [N] is the radial force acting on the bearing;

  l = 45 [mm] – the contact length between spindle and bearing;

- $\psi = (D - d) / d$ – the relative clearance between the bearing inner diameter D [mm] and the spindle outer diameter d [mm]; the nominal diameters of both bearing and spindle are the same, 30 [mm]; since the fit is E6/h9 [16], the tolerance limits are as follows:

  $D = 30^{+.053}_{+.040}$ [mm] and $d = 30^{.000}_{-.052}$ [mm] [20];

  $\psi_{max} = (D_{max} - d_{min}) / d_{min} =$
  $(30.053 - 29.948) / 29.948 = 0.0035$ [-] (5)

3. The Experimental Stand

The components of the experimental stand are presented in fig. 2.

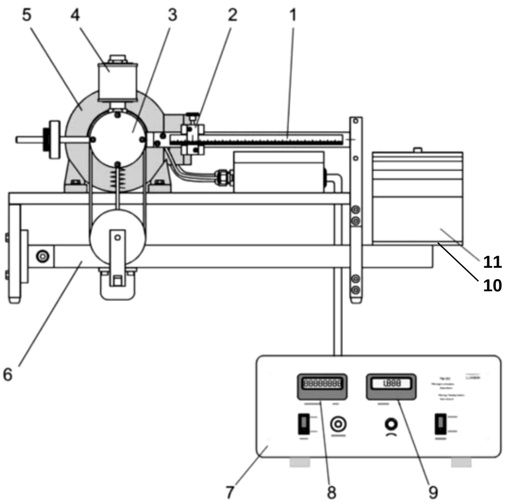

Fig. 2. Scheme of the experimental stand [15], [16].

By using the experimental stand, the energy loss from a radial sliding bearing is determined, depending on lubricant temperature, for certain rotation speeds of the shaft, loaded with constant radial forces.

The rotational movement of the shaft is provided by an electric motor, controlled by a speed regulator.

The values of the spindle rotation speed are read on the display no. (8) and the values of the lubricant temperature in the sliding bearing are read on the display no. (9) of the control system (7) [16].

The bearing is mounted in the oscillating cylindrical housing (3), on which are found, on one side, the graded ruler (1) and the related slider (2), and on the opposite side, a threaded rod with two counterweights for fine adjustment.

The radial force acts on the bearing by means of two belts, which connect the oscillating housing and two belt wheels mounted on a support fixed on the lever (6), on which the plate (10) for the weights (11) is found.

The value of the radial force is reached by loading the system with levers with the force G, by means of weights (11) placed on the plate (10), and the balancing of the friction moment is ensured by moving the cursor (2) in relation to the graded ruler (1) until the horizontal marks on the ruler and on the stand are aligned. The force R can be computed with the following formula [15], [16], [17]:

$$R = G_0 + 5 \cdot G = 25 + 5 \cdot G \text{ [N]}, \qquad (6)$$

where $G_0 = 25$ [N] and G = 20; 40 [N].

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

$$\mu_e = F_f / R \text{ [-]} \qquad (7)$$

The friction moment and the friction force in the radial sliding bearing are computed with the following expressions [15], [16], [17]:

$$M_f = F_f \cdot \frac{d}{2} = F_0 \cdot a = 1 \cdot a \text{ [N·mm]} \qquad (8)$$

$$F_f = 2 \cdot M_f / d \text{ [N]} \qquad (9)$$

where:

- a – is the length of the force arm, which is measured on the graded ruler of the balance scale, for the horizontal markings to coincide;
- $F_0 = 1$ [N], from [17].

The wick oiler (4), mounted on the bearing housing, provides the lubricant required for hydrodynamic lubrication. From the housing, the oil is circulated to the collector vessel, assembled in the lower part of the oscillating housing (3).

4. Experimental Steps

The following experimental steps are carried out [16]:
1) the experimental stand is powered;
2) the weight (11), G = 20 [N], is placed on the plate (10), thus obtaining the radial force $R_1 = 125$ [N] for loading the bearing;
3) adjust the shaft rotation speed to n = 500 [rpm], and reading the oil temperature of 35°, when occurring, and the cursor positioning distance “a” of the cursor (2) on the graded ruler (1) when the scale is in balance and the two horizontal marks overlap;
4) the previous step is repeated for the following temperature values: 40°, 45°, 50°, 55°, 60°;
5) modifying the rotation speed to the next values successively: n = 1000, 1500, 2000 [rpm], and repeating the steps 2 - 4;
6) changing the value of the force applied to the bearing by loading the weight G = 40 [N] on the plate of the lever system and repeating the bearing testing for steps 2 - 5;

- the expressions (1) – (9) are used for the computing of parameters corresponding to all experimental scenarios.

5. Experimental Results

The experimental stand is shown in fig. 3; the notations have the same meanings as in the second chapter.

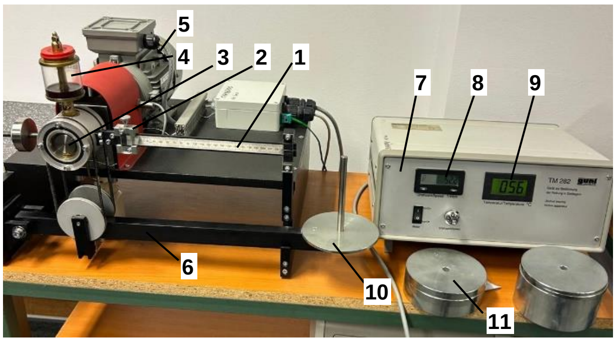

Fig. 3. The experimental stand.

The experimental results are shown in tables 1 and 2.

Table 1. Experimental results for loading force R = 125 [N].

Crt. no.	Temperature [°C]	Loading Force R_1 = 125 [N] Rotation Speed n_1 = 500 [rpm] Distance a [mm]	Friction coefficient μ_e [-]	Friction moment M_f [N·mm]
1.	35	65	0,035	65
2.	40	32	0,017	32
3.	45	25	0,013	25
4.	50	17	0,009	17
5.	55	12	0,006	12
6.	60	9	0,005	9
		Rotation Speed n_2 = 1000 [rpm]		
1.	35	120	0,064	120
2.	40	88	0,047	88
3.	45	68	0,036	68
4.	50	48	0,026	48
5.	55	34	0,018	34
6.	60	25	0,013	25
		Rotation Speed n_3 = 1500 [rpm]		
1.	35	178	0,095	178
2.	40	128	0,068	128
3.	45	105	0,056	105
4.	50	85	0,045	85
5.	55	56	0,030	56
6.	60	45	0,024	45
		Rotation Speed n_4 = 2000 [rpm]		
1.	35	192	0,102	192
2.	40	158	0,084	158
3.	45	125	0,067	125
4.	50	105	0,056	105
5.	55	68	0,036	68
6.	60	60	0,032	60

Table 2. Experimental results for loading force R_2 = 225 [N].

Crt. no.	Temperature [°C]	Loading Force R_2 = 225 [N] Rotation Speed n_1 = 500 [rpm] Distance a [mm]	Friction coefficient μ_e [-]	Friction moment M_f [N·mm]
1.	35	54	0,016	54
2.	40	36	0,011	36
3.	45	22	0,007	22
4.	50	15	0,004	15
5.	55	10	0,003	10
6.	60	7	0,002	7
		Rotation Speed n_2 = 1000 [rpm]		
1.	35	130	0,039	130
2.	40	96	0,028	96
3.	45	78	0,023	78
4.	50	58	0,017	58
5.	55	28	0,008	28
6.	60	20	0,006	20
		Rotation Speed n_3 = 1500 [rpm]		
1.	35	184	0,055	184
2.	40	130	0,039	130
3.	45	115	0,034	115
4.	50	92	0,027	92
5.	55	60	0,018	60
6.	60	56	0,017	56
		Rotation Speed n_4 = 2000 [rpm]		
1.	35	210	0,062	210
2.	40	172	0,051	172
3.	45	130	0,039	130
4.	50	110	0,033	110
5.	55	72	0,021	72
6.	60	68	0,020	68

The graphical dependence of friction coefficient - temperature, for loading force R_1 = 125 [N] and R_2 = 225 [N], is presented in figures 4 and 5, respectively.

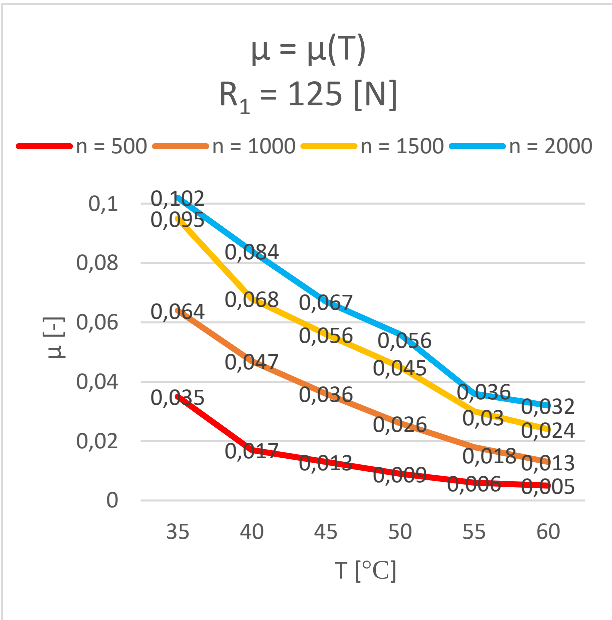

Fig. 4. Friction coefficient vs. temperature, μ = μ(T), for R_1 = 125 [N].

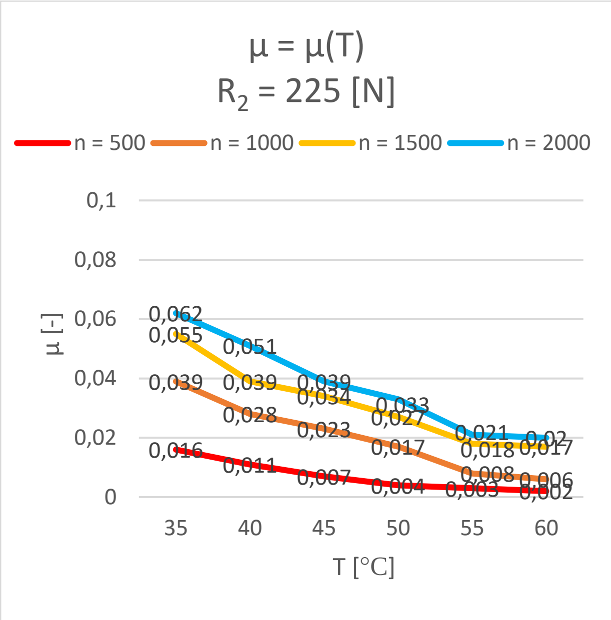

Fig. 5. Friction coefficient vs. temperature, $\mu = \mu(T)$, for $R_2 = 225$ [N].

5. Conclusions

Following the tests performed, for a constant radial force, it is observed that the increase in the temperature causes the decrease in the friction coefficient (fig. 4, 5), and the increase in the radial force also causes the decrease in the friction coefficient.

Comparing the theoretical friction coefficient value computed with the relation (1) to the experimentally determined value (table 1, rotation speed 500 rpm, row 4) for the radial force R = 125 [N] and temperature T = 50 [°C], it can be observed that they are close: $\mu_t = 0.011$ and $\mu_e = 0.009$.

As further research, the shaft material can be changed and the variation of the friction coefficient value can be studied.

6. References

[1] Anghel. St., Ianici, S.: ”Testarea Organelor de Mașini / Testing of Machine Parts ”, Eftimie Murgu Publishing House, Resita, 1998.
[2] Brinkmann L., Kock S., Lang J., Knoll G.: “Tribological analysis of the plain bearings in an external gear pump”, 19th Drive Train Technology Conference (ATK 2021), IOP Conference Series-Materials Science and Engineering, Vol. 1097, Art. No. 012014, 2021.
[3] Cojocaru V., Miclosina C.-O.: “Numerical Analysis of the Influence of Clearance on Stress State and Contact Pressure in Plain Bearings”, Robotica & Management, ISSN 1453-2069, Vol. 22, No. 2, December 2017, pp. 4-7.
[4] Filgueira F.I.C.M., Bottene A.C., Silva E.J., Nicoletti R.: “Static behavior of plain journal bearings with textured journal- Experimental analysis”, Tribology International, Vol. 159, Art. no. 106970, 2021.
[5] Ianici, S.: “Organe de Mașini / Machine Parts”. Eftimie Murgu Publishing House, Resita, 2001.
[6] Ianici, S.: “Ingineria Sistemelor Mecanice / Mechanical System Engineering”. Eftimie Murgu Publishing House, Resita, 2010.
[7] Ianici, D., Ianici, S.: “Testarea Sistemelor Mecanice / Testing Mechanical Systems”, Laboratory handbook, Eftimie Murgu University of Resita, 2010.
[8] Ianici, D., Ianici, S.: “Elemente de Inginerie Mecanică / Elements of mechanical engineering”, Laboratory handbook, Eftimie Murgu University of Resita, 2015.
[9] Li S.S., Shen Y, Lu Y.X.: “The Thermohydrodynamic Analysis of Sliding Bearing High-Speed Motorized Spindle by Rotor Dynamic”, The 3rd International Conference on Mechatronics and Mechanical Engineering (ICMME 2016), Oct. 21-23, 2016, Shanghai, China, vol. 95, art. no. 12012, 2017.
https://doi.org/10.1051/matecconf/20179512012
[10] Manea, Gh.: “Organe de Mașini / Machine Parts”. Vol. I-II., Politechnic Institute Iași, 1978.
[11] Martsinkovsky V., Yurko V., Tarelnik V., Filonenko Y.: “Designing Radial Sliding Bearing Equipped with Hydrostatically Suspended Pads”, Procedia Engineering, Vol. 39, 2012, pp. 157-167,
https://doi.org/10.1016/j.proeng.2012.07.020
[12] Mădăras, L.: “Organe de Mașini / Machine Parts”. Vol. I-II”. Politehnica University Timișoara, 1990.
[13] Miclosina C.-O., Cojocaru V., Vela D.-G.: „Friction Forces in Numerical Simulations of Kinematical Joints of Mechanical Systems”, Robotica & Management, Vol. 26, No. 1, June 2021, pp. 9-12.
https://doi.org/10.24193/rm.2021.1.2
[14] Shutin D., Kazakov Y.: “Theoretical and Numerical Investigation of Reduction of Viscous Friction in Circular and Non-Circular Journal Bearings Using Active Lubrication”, Lubricants, vol. 15, issue 5, art. no. 218, 2023.
https://doi.org/10.3390/lubricants11050218
[15] Vela D.G., Vela I., Miclosina C.-O.: “Tribological Analysis of a Radial Plain Bearing”, Robotica & Management, Vol. 27, No. 1, 2022, pp. 41-44,
https://doi.org/10.24193/rm.2022.1.8
[16] ***: “Equipment for Engineering Education – Mechatronics”, GUNT, Hamburg, 2007.
[17] ***: “Experiment Instructions. Journal Bearing Friction Apparatus”, GUNT, Hamburg, 2007.
[18] ***: Hydraulic Oil Viscosity
https://fluidpower.pro/oil-viscosity/
[19] ***: “ISO Grade Oils - Viscosities and Densities”
https://www.engineeringtoolbox.com/iso-grade-oil-d_1207.html
[20] ***: “ISO Tolerances for Holes (ISO 286-2)”
https://simplybearings.co.uk/shop/Info-Pages-ISO-Limits/c4746_4779/index.html#Q2