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ANALYSIS OF THE INFLUENCE OF SELECTED FACTORS ON THE CAPACITY OF THRUST SLIDING BEARINGS LUBRICATED WITH MAGNETIC FLUIDS

ANALIZA WYPŁYWU WYBRANYCH CZYNNIKÓW NA NOŚNOŚĆ WZDŁUŻNYCH ŁOŻYSK ŚLIZGOWYCH SMAROWANYCH CIECZAMI MAGNETYCZNYMI

Key words: thrust bearing, ferrofluid, magnetorheological fluid.

Abstract The paper presents the results of modelling and experimental validation of a developed mathematical model, describing the influence of selected factors on the axial force generated by magnetic fluids used as lubricants in thrust slide bearings. The model takes into account the physical properties of the magnetic fluid (density, saturation magnetization), bearing geometry (diameter, gap height, magnetic induction in the gap), and bearing load. The proposed model has been validated by comparing the results obtained from the model with the experiments. Good model compatibility with the test results was obtained.

Słowa kluczowe: łożysko wzdłużne, ciecz ferromagnetyczna, ciecz magnetoreologiczna, siła normalna.

Streszczenie W pracy przedstawiono wyniki analiz modelowych oraz walidację eksperymentalną opracowanego modelu opisującego zmiany siły osiowej generowanej przez ciecz magnetyczną pracującą jako środek smarny w łożysku wzdłużnym. Opracowany model pozwala na przeprowadzenie oceny ilościowej wpływu istotnych parametrów węzła łożyska na siłę normalną. W modelu uwzględniono właściwości fizyczne cieczy magnetycznej (gęstość, magnetyzacja nasycenia), parametry geometryczne łożyska (średnica, wysokość szczeliny, indukcja magnetyczna w szczelinie) oraz warunki obciążenia łożyska. Zaproponowany model został poddany walidacji poprzez porównanie wyników otrzymanych z modelu z wynikami badań rzeczywistego układu łożyskowego. Uzyskano dobrą zgodność modelu z wynikami badań.

APPLICATION OF MAGNETIC FLUIDS IN BEARING ENGINEERING

Magnetic fluids are substances whose rheological properties can be actively influenced by magnetic fields. The potential ability to control the rheological properties of magnetic fluids offers new capabilities while developing mechanical devices, such as clutches, brakes, vibration dampers and, in particular, seals. An interesting area for the potential application of magnetic fluids is also bearing engineering. It is possible to minimize friction losses and maximize the durability of bearings by guaranteeing the maintenance of fluid film. In the common design of slide bearings, the effect of fluid film can be achieved by applying hydrostatic or hydrodynamic lubrication.

For those bearing types, important issues to be solved are the proper configuration of the sliding surfaces, and the method of applying the lubricant and spreading it on the bearing surfaces. Another problem is the choice of sliding materials and lubricant properties. Changes in the viscosity of the lubricant due to temperature should also be taken into account. Another important issue is to control oil film thickness at varying loads. A valuable option for hydrostatic bearings is the ability to control the thickness of the lubricating film or at least maintain a constant value, e.g., by varying the lubricant flow rate. The solution for these problems is related to the construction of complex systems that are susceptible to failure. The use of magnetic fluids as lubricants creates the possibility of developing new bearing types. It is possible to control the parameters of

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their work **[L. 1, 12, 13]**. By using magnetic fluids, it is possible to keep them in a predetermined position and control the direction and flow rate. An additional aspect of sliding bearings lubricated by magnetic fluids is the ability to generate normal force in the direction of the magnetic field vector.

The results of studies into thrust bearings lubricated with magnetic fluids were presented in **[L. 2, 3]**. It has been shown that the use of a magnetorheological fluid allows a load capacity to be obtained that is approximately ten times higher compared to a ferromagnetic fluid. However, this is related to a significant increase in the movement resistance. It has also been found that it is difficult to achieve constant load capacity under variable bearing conditions. The load-bearing capacity largely depends on the rotational speed of the bearing, the height of the gap, and the properties of the magnetic fluid. This is due to the dynamics of the cyclic formation and destruction of chain-like columnar structures inside the magnetic fluid. The research discussed in **[L. 4]** has shown that the use of magnetorheological fluid as a lubricant allows high bearing rigidity. This feature allows the requirements for the tolerance of the bearing sliding surfaces to be limited.

In the case of thrust bearings, additional load capacity can be achieved by a self-sealing effect. This effect was used in the construction described in **[L. 5]**, where even the power pump was eliminated. This is associated with the ability to keep the magnetic fluid in a certain position by a magnetic field with appropriately shaped bearing surfaces. This limits the outflow of magnetic fluid from the bearing gap due to the presence of a magnetic barrier which counteracts the movement of the lubricant. Such barriers may arise from the local change in magnetic field induction, like in the case of magnetic fluid seals.

In sliding bearings, an additional buoyant force can be obtained because of the pressure generated in the magnetic fluid due to the influence of the magnetic field gradient. This phenomenon is discussed more extensively in other works **[L. 6, 7, 8]**.

Until now, there have been no universal mathematical models describing the dependence of the bearing capacity on the properties of magnetic fluids or working conditions. This difficulty is related to the complexity of physical phenomena occurring during the operation of this type of bearing. This paper proposes a model for determining the bearing capacity by using dimensional analysis. The model takes into account the relationship between the magnetic field parameters, the geometry of the bearing gap, bearing rotation speed, and magnetic fluid properties.

DESIGN OF THE BEARING

The scheme of the thrust bearing lubricated by the magnetic fluid operating under the influence of the magnetic field is shown in **Fig. 1**. The geometry of the bearing is defined as two parallel plates with flat surfaces. The magnetic fluid (2) is located between the moving plate (1) and the front surface of the electromagnet core (4). The value of the magnetic field is controlled by the current value in the winding of the electromagnet (5). The magnetic circuit is closed by the top and bottom housing (3). These elements are made of material with ferromagnetic properties, while the movable plate is made of a paramagnetic material. The direction of the magnetic field strength vector is perpendicular to the plate surfaces. More information about such bearings is in another publication from the authrs **[L. 9]**.

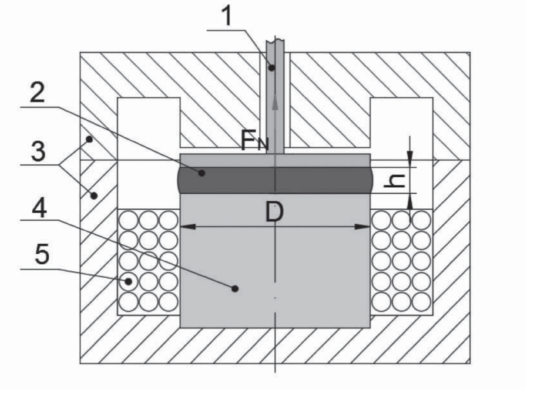


Fig. 1. The scheme of thenalysed bearing
Rys. 1. Schemat analizowanego łożyska

MATHEMATICAL MODEL

For the mathematical description of the axial force, a dimensional analysis was applied. This modelling approach is particularly useful when the equations describing the studied phenomenon are unknown. Dimensional analysis is based on the assumption that a physical quantity can be represented as a power-law monomial of other dependent physical quantities.

The base of dimensional analysis is the theorem Π **[L. 14]**, according to which the equation is dimensionally homogeneous and binding *n* dimensional quantities:

$$f\left(a_1, a_2 \ldots a_{k,} a_{k+1}, \ldots a_n\right) = 0 \tag{1}$$

of which *k* is dimensionally independent and can be represented by the relation *n-k* of mutually independent dimensionless parameters:

$$f\left(a_1, a_2 \ldots a_{k,} a_{k+1}, \ldots a_n\right) = 0 \tag{2}$$

where

$$\Pi_1 = \frac{a_{k+1}}{w_1}, \Pi_2 = \frac{a_{k+2}}{w_2}, \ldots \Pi_{n-k} = \frac{a_n}{w_{n-k}} \tag{3}$$

In the case of the axial (normal) force F_N model, it was assumed that it depends on the following: magnetic fluid saturation magnetization – M_S [A·m^{-1}], magnetic field induction inside the bearing gap – B [kg·A^{-1}·m^{-2}], bearing gap height – h [m], magnetic fluid density – ρ [kg·m^{-3}], bearing plate diameter – D [m], dynamic viscosity of the magnetic fluid – η [kg·m^{-1}·s^{-1}], and angular velocity – ω [s$_{-1}$].

It should be noted that the magnetic induction in the examined bearing is not constant throughout the gap diameter. There may be an increase in magnetic induction on the edges of the electromagnet core. In turn, in the middle of the core surface, where there is an opening in the upper housing, magnetic induction decreases **[L. 9]**. In the presented model, in order to simplify the analysis, it is assumed that B is a constant parameter.

Three dimensionless parameters were selected and determined: Π_1, Π_2, Π_3. They can be represented as:

$$\Pi_1 = \frac{F_n}{M_S^{a_1} \cdot B^{b_1} \cdot h^{c_1} \cdot \rho^{d_1}} \tag{4}$$

$$\Pi_2 = \frac{D}{M_S^{a_2} \cdot B^{b_2} \cdot h^{c_2} \cdot \rho^{d_2}} \tag{5}$$

$$\Pi_3 = \frac{\eta}{M_S^{a_3} \cdot B^{b_3} \cdot h^{c_3} \cdot \rho^{d_3}} \tag{6}$$

After determining the values of the corresponding exponents a, b, c, and d using a dimension matrix, the dimensionless parameters have the following form:

$$\Pi_1 = \frac{F_n}{M_S \cdot B \cdot h^2} \tag{7}$$

$$\Pi_2 = \frac{D}{h} \tag{8}$$

$$\Pi_3 = \frac{\eta}{(M_S \cdot B \cdot \rho)^{0,5} \cdot h} \tag{9}$$

According to the theorem Π, the dimensional equation can be written as:

$$\Pi_1 = C_1 \cdot \Pi_2^{\alpha} \cdot \Pi_3^{\beta} \tag{10}$$

where
α, β – real numbers,
C_1 – dimensionless proportional coefficient.

Taking into account the earlier relationships, Equation (10) can be written as:

$$\frac{F_n}{M_S \cdot B \cdot h^2} = C_1 \cdot \left(\frac{D}{h}\right)^{\alpha} \cdot \left(\frac{\eta}{\sqrt{M_S \cdot B \cdot \rho \cdot h}}\right)^{\beta} \tag{11}$$

Or

$$F_n = C_1 \cdot \left(M_S \cdot B\right)^{1-\frac{\beta}{2}} \cdot \rho^{\beta} \cdot h^{2-\alpha-\beta} \cdot D^{\alpha} \cdot \eta^{\beta} \tag{12}$$

VALIDATION OF THE MODEL

In order to validate the model, a series of experiments were performed to determine the value of the model coefficients. Then, a comparison of the normal force values obtained from the experiments and calculated from the model was made. A compatibility model with experimental data was estimated by the coefficient of determination R^2.

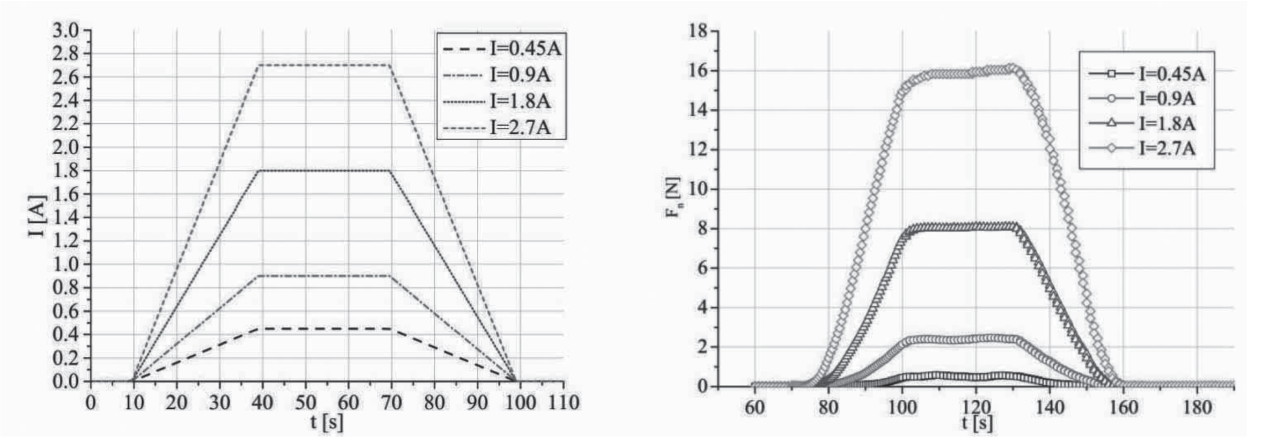
The study was performed on a Physic MCR 301 rotary rheometer, equipped with a measuring cell enabling samples to be examined in the magnetic field. Plate-plate geometry (**Fig. 1**) of D = 20 mm diameter was used. The experiments were performed at a constant temperature of t = 25°C. The volume of the magnetic fluid was v = 175 μl.

Based on the results of a previous study **[L. 3],** MRF-122EG magnetorheological fluid from LORD Corporation was selected for the experiment. This fluid is characterized by the saturation magnetization M_s = 361 kA/m, the density ρ = 2.38 g/cm^3, and the viscosity η = 203.4 Pa·s.

The experiment was based on measuring the response of the magnetic fluid sample to the magnetic field induction when B = 0.1; 0.2; 0.4; 0.6 T, which corresponds to the current value in the coil: I = 0.45; 0.9; 1.8; 2.7 A. During the experiments, the current change over time was as in **Fig. 2a**. In the first stage, a linear increase in current was applied, and then the current value was stabilized at a constant value for about 30 s, after which there was a linear decrease in the current. During the tests, the normal force F_N changes were recorded – **Fig. 2b**. Measurements were made at shear rates $\dot{\gamma}$ = 0; 1; 10; 100 s^{-1}. The speed of rotation of the measuring plate was set as the shear rate $\dot{\gamma}$ [s^{-1}], but for the dimensional analysis, it was necessary to go to the basic unit ω [s^{-1}] according to the following relation:

$$\omega = \frac{\dot{\gamma} \cdot h}{2 \cdot \pi \cdot D / 2} \tag{13}$$

Figure 2b shows an example of the normal force measurement. In order to validate the model (12), the maximum values of the normal force obtained during the tests had been chosen.



a) b)

Fig. 2. a) Current change during the tests, b) Sample measurement results for I = idem., γ = 10 s^{-1}, h = 0.25 m

Rys. 2. a) Przebieg zmian prądu, b) Przykładowy wynik pomiaru F_n dla I = idem., $\dot{\gamma}$ = 10 s^{-1}, h = 0,25 mm

Figure 3 shows the relationship between the normal force and magnetic field induction for different gap heights, when the movable plate is stopped (ω = 0 s^{-1}). Based on experimental data, the coefficients of Equation (12) were determined as C_1 = 0.00023, and α = 2.7, β = -1. In the analysed range of parameter variability, the mathematical model with experimental data was highly consistent. This confirms the high value of R^2 (above 0.99).

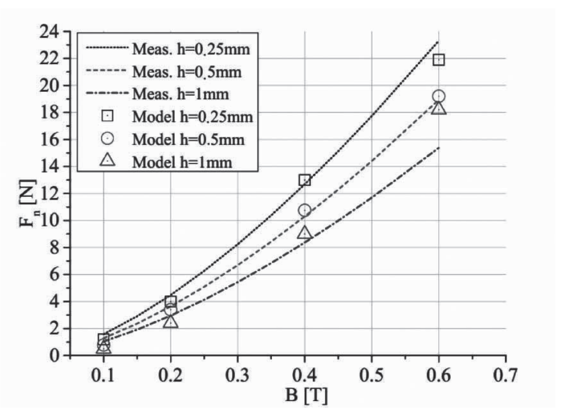


Fig. 3. F_N vs. B for ω = 0 s^{-1}, *h* = 0.25; 0.5; 1m

Rys. 3. Zależność siły F_N od B dla ω = 0 s^{-1}, *h* = 0,25; 0,5; 1 mm

When analysing the case in which the measuring plate was rotating (ω ≠ 0), the centrifugal force in Equation (12) should be taken into account. In this case, the equation for normal force is expressed as follows:

$$F_n = C_1 \cdot \left(M_S \cdot B\right)^{1,5} \cdot \rho^{0,5} \cdot h^{0,3} \cdot D^{2,7} \cdot \eta^{-1} - C_2 \cdot \rho \cdot \omega^2 \cdot D^4 \tag{14}$$

The determined values of the coefficients for equation (14) are as follows: C_1= 0.0002, C_2= 1.300.

Figs. 4, 5, and 6 show the dependence of force vs. magnetic field induction for different gap heights (*h* = 0.25; 0.5; 1 mm) and different shear rates ($\dot{\gamma}$ = 1; 10; 100 s^{-1}).

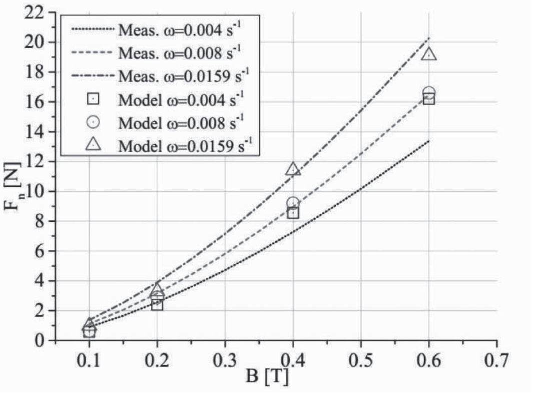


Fig. 4. F_N vs. B for γ = 1 s^{-1}, *h* = 0.25; 0.5; 1 mm

Rys. 4. Zależność siły F_N od B dla γ = 1 s^{-1}, *h* = 0,25; 0,5; 1 mm

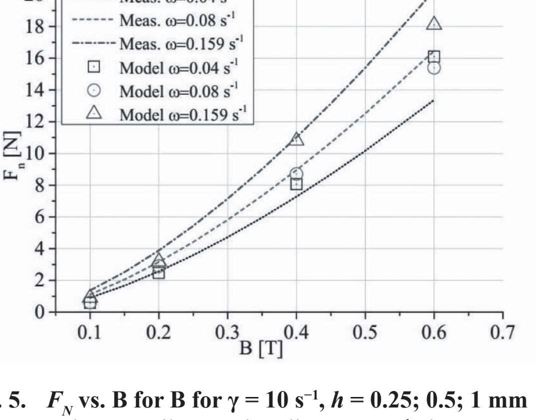


Fig. 5. F_N vs. B for B for γ = 10 s^{-1}, *h* = 0.25; 0.5; 1 mm

Rys. 5. Zależność siły F_N od B dla $\dot{\gamma}$ = 10 s^{-1}, *h* = 0,25; 0,5; 1 mm

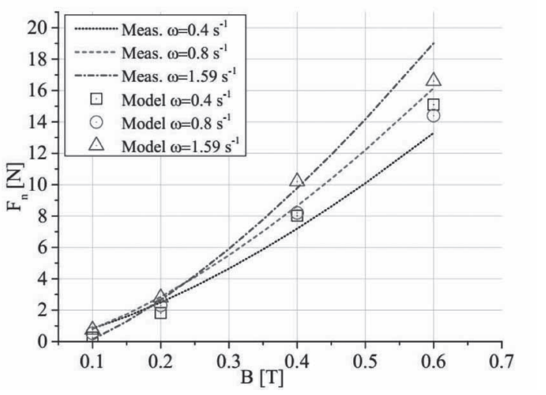


Fig. 6. F_N vs. B for $\gamma = 100\ s^{-1}$, $h = 0.25; 0.5; 1$ mm

Rys. 6. Zależność siły F_N od B dla $\dot{\gamma} = 100\ s^{-1}$, $h = 0{,}25; 0{,}5; 1$ mm

In addition, for this case ($\omega \neq 0$) in the analysed range of parameter variability, the mathematical model was highly compatible with experimental data. This confirms the high value of the coefficient determination R^2 above 0.99.

SUMMARY AND CONCLUSIONS

The main purpose of the study was to obtain and validate a model that would describe the influence of geometry and working conditions on axial force generated in thrust bearings lubricated with magnetic fluids.

The proposed model illustrates the relationship between normal force and parameters, such as the magnetic fluid saturation magnetization, the mean value of the magnetic induction flux inside the bearing gap, and the physical parameters of the fluid, such as dynamic viscosity and density.

The presented results showed that, in the range of variability of the analysed parameters, the developed model describes the normal force well. The obtained correlation indicates that increasing the height of the working gap in the bearing should result in an increase in the normal force (accordance with the power exponent 0.3).

It has also been shown that the value of the normal force is related to the diameter of the bearing plate by the exponent 2.7. Therefore, this parameter significantly affects the load capacity of the bearing. Equation (14) indicates that the rotation causes a decrease in the normal force. It should be noted that a high value of the coefficient determination for the model was achieved.

Thrust bearings lubricated with magnetic fluids have not yet found wide-ranging technical applications. The bearing capacity values obtained during the tests are several orders lower than those lubricated hydrostatically or in a hydrodynamic manner. This probably excludes their use in typical applications.

Possible areas of application for this type of bearing are systems in which controlling the rheological properties of the lubricant is important. Research concerning their basic parameters, in particular torque and load, is described in **[L. 11]**. These bearings can also be used under zero gravity working conditions, because the magnetic fluid can be held in place by a magnetic field. This eliminates lubricant leaks. Attempts to use such bearings in conjunction with classical solutions are described in **[L. 10]**.

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