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RELIABILITY ASPECTS OF THE EFFECT OF APPLICATION 2% AND 8% FERRO-OIL AS A LUBRICANT IN A SLIDE JOURNAL BEARING ON ITS OPERATING PARAMETERS

Niezawodnościowe aspekty wpływu zastosowania 2% lub 8% ferro-oleju jako środka smarnego w poprzecznym łożysku ślizgowym na jego parametry eksploatacyjne

Abstract: *The purpose of this paper is to determine the impact of using ferro-oil with 2% or 8% concentration of the magnetic particles as the lubricant of the slide journal bearing on its operating parameters regarding the reliability aspect. There has been presented an analytical and numerical calculation model based on experimentally determined physical quantities describing the dependence of viscosity of ferro-oil on fundamental parameters in the paper. Numerical calculations were performed by solving the Reynold's type equation using the finite difference method using Mathcad 15 and own calculation procedures.*
Keywords: slide journal bearing, ferro-oil, operating parameters, reliability

Streszczenie: *Celem niniejszej pracy jest określenie jaki wpływ na zmianę parametrów eksploatacyjnych łożyska ślizgowego w aspekcie niezawodności może mieć zastawanie jako środka smarnego ferro-oleju o 2% lub 8% stężeniu cząstek magnetycznych. W pracy został przedstawiony analityczno-numeryczny model obliczeniowy bazujący na eksperymentalnie wyznaczonych fizycznych wielkościach opisujących zależność lepkości ferro-oleju od zmian podstawowych parametrów pracy. Obliczenia numeryczne wykonano rozwiązując równania typu Reynoldsa metodą różnic skończonych przy wykorzystaniu program Mathcad 15 i własnych procedur obliczeniowych.*
Słowa kluczowe: poprzeczne łożysko ślizgowe, ferro-olej, parametry eksploatacyjne, niezawodność

1. Introduction

The concept of using "smart" lubricants, such as ferro-oils, exhibiting susceptibility to control their physical properties, in modern high-tech mechanical constructions, is an attempt to meet the high expectations regarding their functioning in working conditions of often extreme nature [9,12,13,16,21-26,30,31]. However, the use of ferro-oil as a lubricant in the slide journal bearings, due to the economic aspects as well as technical complexity of these constructions, has its justification in only a few considered cases.

The first of these, the most natural case, will be the one when the use of "classical" lubricants will not be possible due to the unique nature of the extremely unfavorable environment conditions for these bearings, which will make impossible to maintain the lubricant in the lubrication gap of the bearing. Such a case of absolute necessity, is, for example, the functioning of the bearing under vacuum conditions, lack of gravity or under the influence of strong electromagnetic fields.

The second of these potential cases, is the situation when the use of "intelligent" lubricant like a ferro-oil, susceptible to the ability to control its properties, allows to obtain high precision of operation of mechanical devices as well as specialized and precisely functioning friction nodes of these devices, adapting to working conditions changes by adjusting the height of the lubrication gap or by adaptive vibration dampening.

The third one, nowhere considered case so far, is the aspect of the reliability or safety of technical facilities in which a such ferro-oil-lubricated bearing would be used. It seems that in principle would apply a situation when the journal slide bearings of such technical objects would be subject to external loads with a very wide range of values, or if these loads periodically assumed a very large amplitude values. The next part of this paper will be devoted to consideration on this third case.

2. Theoretical considerations

It seems that from the point of view of reliability, it would be necessary to consider two different cases of functioning of a slide journal bearing lubricated with ferro-oil. The first one would essentially concern a situation where such a bearing would operate under the rated load conditions and would only be periodically subjected to external loads of a significantly higher value going beyond the normal operating range. In a similar way, one could perceive a situation where in the "normal" mode of operation the bearing operates in classical environmental conditions and only occasionally is exposed to work in extreme environmental conditions.

In both cases, the "switching" of the external control magnetic field acting on the ferro-oil, would allow to change and adapt its viscosity. As a consequence, it would be possible to obtain an additional lift force in the bearing and to effectively counteract of the decrease in the height of the lubrication gap, which would protect against the occurrence of mixed friction and, as a result, damage to the bearing [2,8,10,11,15]. It seems that structurally such a case would be the same as a situation in which a cool parametric reserve exists in the reliability system [14, 17]. Such a reserve would be included in operational work, incidentally and excluded, except when its use would be necessary. The advantage of such a solution would be a compact design of the functional system of the bearing without the need to duplicate the system with additional elements, eg an additional static bearing and also simplification of the reliability structure of such a system. Also, the relative simplicity of control of bearing operating properties should be seen as another, significant advantage of such a solution [17, 20].

The second slightly different case is the situation when the slide journal bearing lubricated with ferro-oil functions constantly in a very wide range of external loads. In such a situation, the control system of the external magnetic field should constantly adaptively respond to changes in the bearing load and adjust the physical properties of the lubricant in such a way that the bearing operation takes place with optimal values of its operating parameters, ie carrying capacity coefficient, friction force and coefficient of friction. The possibility of performing work with optimal parameters would significantly increase the reliability and durability of such a bearing. Damage or intensive wear of the slide bearing in the most significant way [2, 11, 15] affects these moments of its operation when the transition from liquid friction to mixed friction occurs. This transition is accompanied by very intensive wear of the cooperating surfaces, change of bearing geometry and blurring [8,10]. The assumptions of the lubrication theory of slide bearings assume that the possibility of avoiding friction with a different character than a fluid one would allow to extend the life of the slide bearing even to infinity [2, 11, 20].

3. Analytical-numerical computational model

An analytical model of magnetohydrodynamic lubrication of slide journal bearings was derived from fundamental equations, ie equations of momentum conservation, equations of flow continuity, equations of energy conservation (1)-(3) as well as Maxwell's equations (4) in the following presented form [3,18,19,27-29]. The non-isothermal bearing lubrication model was assumed with a laminar and steady lubricant flow rate and the external magnetic field was adopted as stationary, transverse to the ferro-oil flow in the bearing gap.

$$0 = Div\,\mathbf{S} + \mu_o(\mathbf{N}\cdot\nabla)\mathbf{H} + \tfrac{1}{2}\mu_o rot(\mathbf{N}\times\mathbf{H}), \tag{1}$$

$$div(\rho\mathbf{v}) = 0, \tag{2}$$

$$div(\kappa\, grad\, T) + div(\mathbf{vS}) - \mathbf{v}\mathrm{Div}\mathbf{S} - \mu_o \mathrm{T}\frac{\partial \boldsymbol{N}}{\partial T}\frac{d\mathbf{H}}{d\,t} + \Omega = \rho\frac{d(c_v T)}{d\,t}, \tag{3}$$

$$\mathrm{rot}\,\boldsymbol{H} = 0, \quad \mathrm{div}\,\boldsymbol{B} = 0, \quad \boldsymbol{B}=\mu_o(\boldsymbol{H}+\boldsymbol{N}), \quad \boldsymbol{N}=\boldsymbol{H}\cdot\chi, \tag{4}$$

where:
$\boldsymbol{B}$ – magnetic induction vector in ferro-oil [T],
$\boldsymbol{H}$ – vector of magnetic field strength in ferro-oil [$A\cdot m^{-1}$],
$\boldsymbol{N}$ – vector of ferro-oil magnetization [$A\cdot m^{-1}$],
$\boldsymbol{S}$ – tensor of ferro-oil stresses with coordinates τ_{ij} for i,j=ϕ,r,z [Pa],
$\boldsymbol{v}$ – velocity vector of ferro-oil [$m\cdot s^{-1}$],
∇ – Nabla operator,
μ_o – magnetic permeability of the vacuum [$H\cdot m^{-1}$],
ρ – density of ferro-oil [$kg\cdot m^{-3}$],
χ – magnetic ferro-oil susceptibility factor.

As the constitutive equation for ferro-oil was adopted non-Newtonian viscoelastic model of Rivlin-Ericksen’ fluid. This relationship describes the relationship between stress tensor coordinates and the tensor of shear rate of the ferro-oil and it can be presented in the following form [18, 29]:

$$\boldsymbol{S} = -p\,\boldsymbol{I} + \eta\boldsymbol{A}_1 + \alpha\boldsymbol{A}_1\boldsymbol{A}_1 + \beta\boldsymbol{A}_2. \tag{5}$$

The following are the relationships (6) describing shear rate tensors [29]:

$$\boldsymbol{A}_1 \equiv \boldsymbol{L} + \boldsymbol{L}^{\mathrm{T}}, \quad \boldsymbol{A}_2 \equiv \mathrm{grad}\,\boldsymbol{a} + (\mathrm{grad}\,\boldsymbol{a})^{\mathrm{T}} + 2\boldsymbol{L}^{\mathrm{T}}\cdot\boldsymbol{L}, \tag{6}$$

and the acceleration vector in a formula (7):

$$\boldsymbol{a} \equiv \boldsymbol{L}\cdot\boldsymbol{v}, \quad \boldsymbol{L} \equiv \mathrm{grad}\,\boldsymbol{v}, \tag{7}$$

where:
$\boldsymbol{A}_1$ – first shear rate tensor [s^{-1}],
$\boldsymbol{A}_2$ – second shear rate tensor [s^{-2}],
$\boldsymbol{I}$ – unit tensor,
$\boldsymbol{L}$ – gradient of the velocity vector tensor [s^{-1}],
$\boldsymbol{a}$ – acceleration vector [$m\cdot s^{-2}$],
$\boldsymbol{p}$ – hydrodynamic pressure [Pa],
α, β – experimental factors determining viscoelastic properties of ferro-oil [$Pa\cdot s^2$],
η – dynamic viscosity coefficient [$Pa\cdot s$].

The material coefficients α, β of the lubricating liquid multiplied by the appropriate shear rate tensors take into account the additional stresses resulting from the viscoelastic, non-Newtonian character of the ferro-oil. The paper [19] presents a method for determining the values of these constants for the investigated ferro-oil.

The dependence of the dynamic viscosity of the ferro-oil on magnetic induction, temperature and pressure $\eta=\eta(B,T,p)$ was taken into account. Whereas the material coefficients α, β were assumed to be constant [19]. Based on the results of research on the properties of ferro-oils [4-7], it has been assumed that the characteristics of viscosity changes associated with changes in temperature and pressure will be modeled using exponential relations (9), (10). Changes in ferro-oil viscosity due to the influence of the external magnetic field have been modeled by power dependence (11):

$$\eta = \eta_o \eta_1; \quad \eta_1 = \eta_{1B}\eta_{1p}\eta_{1T}, \tag{8}$$

$$\eta_{1p}(\phi, z) = a_p e^{\zeta \cdot p_o \cdot p_1} = a_p e^{\zeta_p p_1}, \tag{9}$$

$$\eta_{1T}(\phi, z, r) \equiv a_T e^{-\delta_T (T - T_o)} = a_T e^{-Q_{Br} T_1}, \tag{10}$$

$$\eta_{1B}(\phi, z) = 1 + a_B (B_0 B_1)^{\delta_{1B}} = 1 + a_{B1} (B_1)^{\delta_{1B}}, \tag{11}$$

where:

η_1 – total dimensionless dynamic viscosity,

η_o – characteristic dimensional value of dynamic viscosity [Pa·s],

η_{1p} – dimensionless dynamic viscosity depending on the pressure,

η_{1T} – dimensionless dynamic viscosity depending on the temperature,

η_{1B} – dimensionless dynamic viscosity depends on magnetic field induction,

δ_{B1} – dimensionless material factor including changes in viscosity from magnetic field,

ζ, ζ_p – dimensional [1/Pa] and dimensionless material factor including changes in viscosity depended on hydrodynamic pressure,

δ_T – material factor including changes in viscosity depended on temperature,

a_B – proportionality factor $\left[\mathrm{T}^{-\delta_{B1}}\right]$,

a_{B1} – dimensionless proportionality coefficient,

a_p – proportionality factor,

a_T – proportionality factor,

κ_o – coefficient of thermal conductivity of ferro-oil,

ω – angular velocity,

B_o – dimensional value of magnetic field induction,

B_1 – dimensionless induction of magnetic field,

Q_{Br} – dimensionless coefficient of viscosity changes from temperature,

R – the radius of the pin,

p_1 – dimensionless hydrodynamic pressure,
p_o – dimensional value of hydrodynamic pressure.

Equations of motion (1)-(2) have been substituted for constitutive relationships (4) between stress tensor coordinates and shear rate tensor coordinates. There were omitted nonstationary units and units of inertia forces in equations of momentum. Such omission is reasonable in the slow and medium speed bearings. It can be obtained the full set of equations of motion for the classical, steady flow of lubricating oil in this way.

The next step in solving the system of equations is its equalization and the estimation of the order of values of the individual members. For this purpose, the dimensional and dimensionless marks and numbers [7, 18, 29], known in the literature have been assumed.

A system of equations in the dimensionless form contains units of the order of a unity and members negligibly small order of radial relative clearance $\psi \approx 10^{-3}$. By neglecting the members of the row of radial relative clearance, that is about a thousand times smaller than the values of the other members, a new simplified system of equations is obtained [7, 18]. For further analysis of the basic equations, it has been assumed that the dimensionless density $\rho_1=1$ of the lubricant is constant and independent of both temperature and pressure [7].

In order to determine the function of the desired values such as velocity vector components, hydrodynamic pressure, load carrying capacity, frictional force and friction coefficient, the *small parameter method* was used. This method consists in converting the sought dimensionless quantities to convergent series with respect to small parameters [18,27,29].

In order to determine the hydrodynamic pressure in the ferro-oil, the Reynolds boundary condition was taken as [18]:

$$p_1^{(0)} = 0 \text{ for } \phi=\phi_p, \quad p_1^{(0)} = 0 \text{ for } \phi \ge \phi_k, \quad \frac{\partial p_1^{(0)}}{\partial \phi} = 0 \text{ for } \phi=\phi_k,$$

$$p_1^{(0)} = 0 \text{ for } z_1=+1 \text{ and } z_1=-1,$$

$$p_{10}^{(1)} = 0 \text{ for } \phi=\phi_p, \quad p_{10}^{(1)} = 0 \text{ for } \phi \ge \phi_k, \quad \frac{\partial p_{10}^{(1)}}{\partial \phi} = 0 \text{ for } \phi=\phi_k,$$

$$p_{10}^{(1)} = 0 \text{ for } z_1=+1 \text{ and } z_1=-1,$$

$$p_{11}^{(1)} = 0 \text{ for } \phi=\phi_p, \quad p_{11}^{(1)} = 0 \text{ for } \phi \ge \phi_k, \quad \frac{\partial p_{11}^{(1)}}{\partial \phi} = 0 \text{ for } \phi=\phi_k,$$

$$p_{11}^{(1)} = 0 \text{ for } z_1=+1 \text{ and } z_1= -1,$$

$$p_1^{(1)} = 0 \text{ for } \phi=\phi_p,\ \ p_1^{(1)} = 0 \text{ for } \phi\geq\phi_k,\ \frac{\partial p_1^{(1)}}{\partial\phi} = 0 \text{ for } \phi=\phi_k,$$

$$p_1^{(1)} = 0 \text{ for } z_1=+1 \text{ and } z_1= -1, \tag{12}$$

These conditions mean that the hydrodynamic pressure equals the ambient (atmospheric) pressure to zero relative to the developed pressure in the bearing. Adoption of a zero value refers to a point $\phi=\phi_p$, i.e. an initial coordinate of approximately 4° in the direction of movement of the journal at the front end of the line of the centers, usually at the point where the oil enters the gap as well as at the position $\phi=\phi_k$, that is, the coordinate of the end of the oil film. This value is unknown for Reynolds conditions but is known to lie outside the back end of the line of centers.

Using the continuity equation and previously calculated peripheral and longitudinal components, after integrating the equation and applying appropriate boundary conditions, a velocity vector radial component and a Reynolds type equation are obtained in the following form [7]:

a) for the first set of equations – which takes into account the Newtonian properties and the influence of the magnetic field:

$$\frac{\partial}{\partial\phi}\left[\frac{h_{p1}^3}{\eta_{1B}}\left(\frac{\partial p_1^{(0)}}{\partial\phi} - M_1\right)\right] + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[\frac{h_{p1}^3}{\eta_{1B}}\left(\frac{\partial p_1^{(0)}}{\partial z_1} - M_3\right)\right] = 6\frac{\partial h_{p1}}{\partial\phi}, \tag{13}$$

b) for the second set of equations – it takes into account the influence of temperature on viscosity:

$$\frac{\partial}{\partial\phi}\left[\frac{h_{p1}^3}{\eta_{1B}}\left(\frac{\partial p_{10}^{(1)}}{\partial\phi}\right)\right] + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[\frac{h_{p1}^3}{\eta_{1B}}\left(\frac{\partial p_{10}^{(1)}}{\partial z_1}\right)\right] =$$

$$=12\left\{\frac{\partial}{\partial\phi}\left[\left(\int_0^{h_{p1}}\left(\int_0^{r_1} T_1^{(0)}\frac{\partial v_1^{(0)}}{\partial r_1}dr_1\right)dr_1 - \int_0^{h_{p1}}\frac{r_1}{h_{c1}}\left(\int_0^{h_{p1}} T_1^{(0)}\frac{\partial v_1^{(0)}}{\partial r_1}dr_1\right)dr_1\right)\right]+\right.$$

$$\left.+\frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[\int_0^{h_{p1}}\left(\int_0^{r_1} T_1^{(0)}\frac{\partial v_3^{(0)}}{\partial r_1}dr_1\right)dr_1 - \int_0^{h_{p1}}\frac{r_1}{h_{c1}}\left(\int_0^{h_{p1}} T_1^{(0)}\frac{\partial v_3^{(0)}}{\partial r_1}dr_1\right)dr_1\right]\right\}, \tag{14}$$

c) for the third set of equations – it takes into account the influence of pressure on viscosity:

$$\frac{\partial}{\partial\phi}\left[\frac{h_{p1}^3}{\eta_{1B}}\left(\frac{\partial p_{11}^{(1)}}{\partial\phi}\right)\right]+\frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[\frac{h_{p1}^3}{\eta_{1B}}\left(\frac{\partial p_{11}^{(1)}}{\partial z_1}\right)\right]=$$

$$=12\left\{\frac{\partial}{\partial\phi}\left[\int_0^{h_{p1}}\frac{r_1}{h_{c1}}\left(\int_0^{h_{p1}}p_1^{(0)}\frac{\partial v_1^{(0)}}{\partial r_1}dr_1\right)dr_1-\int_0^{h_{p1}}\left(\int_0^{r_1}p_1^{(0)}\frac{\partial v_1^{(0)}}{\partial r_1}dr_1\right)dr_1\right]+\right.$$

$$\left.+\frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[\int_0^{h_{p1}}\frac{r_1}{h_{c1}}\left(\int_0^{h_{p1}}p_1^{(0)}\frac{\partial v_3^{(0)}}{\partial r_1}dr_1\right)dr_1-\int_0^{h_{p1}}\left(\int_0^{r_1}p_1^{(0)}\frac{\partial v_3^{(0)}}{\partial r_1}dr_1\right)dr_1\right]\right\} \tag{15}$$

d) for the fourth set of equations – it takes into account the influence of non-Newtonian properties on viscosity:

$$\frac{\partial}{\partial\phi}\left(\frac{h_{p1}^3}{\eta_{B1}}\frac{\partial p_{1p}^{(1)}}{\partial\phi}\right)+\frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left(\frac{h_{p1}^3}{\eta_{B1}}\frac{\partial p_{1p}^{(1)}}{\partial z_1}\right)=$$

$$=12\left\{\frac{\partial}{\partial\varphi}\left(\frac{1}{\eta_{B1}}\int_0^{h_{p1}}\int_0^{r_3}\int_0^{r_2}F(\varphi,r_1,z_1)dr_1dr_2dr_3-\frac{h_{p1}}{2\eta_{B1}}\int_0^{h_{p1}}\int_0^{r_2}F(\varphi,r_1,z_1)dr_1dr_2\right)+\right.$$

$$\left.+\frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left(\frac{1}{\eta_{B1}}\int_0^{h_{p1}}\int_0^{r_3}\int_0^{r_2}G(\varphi,r_1,z_1)dr_1dr_2dr_3-\frac{h_{p1}}{2\eta_{B1}}\int_0^{h_{p1}}\int_0^{r_2}G(\varphi,r_1,z_1)dr_1dr_2\right)\right\}, \tag{16}$$

where:

$$v_1^{(0)}(r_1,\varphi,z_1)=\frac{1}{2\eta_{1B}}\left(\frac{\partial p_1^{(0)}}{\partial\phi}-M_1\right)\left(r_1^2-r_1h_{p1}\right)+1-\frac{r_1}{h_{p1}},$$

$$v_3^{(0)}(r_1,\varphi,z_1)=\frac{1}{2\eta_{1B}}\left(\frac{\partial p_1^{(0)}}{\partial z_1}-M_3\right)\left(r_1^2-r_1h_{p1}\right),$$

$$T_1^{(0)}(r_1,\varphi,z_1)=1+\frac{1}{2}\eta_{1B}(1-2s)-q_{1c}^{(0)}h_{p1}s-\frac{1}{2}\Omega_1\left(h_{p1}s\right)^2-\frac{1}{6}h_{p1}^2\left(\frac{\partial p_1^{(0)}}{\partial\varphi}-M_1\right)s\,(3-3s+s^2)+$$

$$-\frac{1}{2}\eta_{1B}\left[\left(v_1^{(0)}\right)^2+\frac{1}{L_1^2}\left(v_3^{(0)}\right)^2\right]+\frac{1}{24\eta_{1B}}h_{p1}^4\left[\left(\frac{\partial p_1^{(0)}}{\partial\varphi}-M_1\right)^2+\frac{1}{L_1^2}\left(\frac{\partial p_1^{(0)}}{\partial z_1}-M_3\right)^2\right]s^3(s-2),$$

$$M_1 = R_f \chi \left[H_1 \frac{\partial H_1}{\partial \varphi} + \frac{1}{L_1} H_3 \frac{\partial H_1}{\partial z_1} \right], \quad M_3 = R_f L_1 \chi \left(H_1 \frac{\partial H_3}{\partial \varphi} + \frac{1}{L_1} H_3 \frac{\partial H_3}{\partial z_1} \right),$$

$$F(\varphi, r_1, z_1) \equiv \left(1 + 2\frac{\beta_o}{\alpha_o}\right)\left(\frac{\partial X_1}{\partial \varphi} + \frac{1}{L_1^2}\frac{\partial Z_1}{\partial \varphi}\right) - \frac{\partial X_1}{\partial \varphi} - \frac{1}{L_1^2}\left(\frac{\partial X_2}{\partial r_1} + \frac{\partial X_3}{\partial z_1}\right) - \frac{\beta_o}{\alpha_o}\left(\frac{\partial X_4}{\partial r_1} + 2\frac{\partial X_5}{\partial r_1}\right),$$

$$G(\varphi, r_1, z_1) \equiv \left(1 + 2\frac{\beta_o}{\alpha_o}\right)\left(\frac{\partial X_1}{\partial z_1} + \frac{1}{L_1^2}\frac{\partial Z_1}{\partial z_1}\right) - \frac{1}{L_1^2}\frac{\partial Z_1}{\partial z_1} - \left(\frac{\partial Z_2}{\partial r_1} + \frac{\partial Z_3}{\partial \varphi}\right) - \frac{\beta_o}{\alpha_o}\left(\frac{\partial Z_4}{\partial r_1} + 2\frac{\partial Z_5}{\partial r_1}\right),$$

$$X_1 \equiv \left(\frac{\partial v_1^{(0)}}{\partial r_1}\right)^2, \; X_2 \equiv \frac{\partial v_3^{(0)}}{\partial r_1} Y_4 - 2\frac{\partial v_1^{(0)}}{\partial r_1}\frac{\partial v_3^{(0)}}{\partial z_1}, \; X_3 \equiv \frac{\partial v_1^{(0)}}{\partial r_1}\frac{\partial v_3^{(0)}}{\partial r_1}, \; Y_1 \equiv \frac{\partial v_1^{(0)}}{\partial \phi},$$

$$Y_2 \equiv \frac{\partial v_2^{(0)}}{\partial r_1} \; Y_3 \equiv \frac{\partial v_3^{(0)}}{\partial z_1}, \; Y_4 \equiv \frac{\partial v_3^{(0)}}{\partial \varphi} + \frac{\partial v_1^{(0)}}{\partial z_1}, \; Z_1 \equiv \left(\frac{\partial v_3^{(0)}}{\partial r_1}\right)^2,$$

$$Z_2 \equiv \frac{\partial v_1^{(0)}}{\partial r_1} Y_4 - 2\frac{\partial v_3^{(0)}}{\partial r_1}\frac{\partial v_1^{(0)}}{\partial \varphi}, \quad Z_3 \equiv \frac{\partial v_1^{(0)}}{\partial r_1}\frac{\partial v_3^{(0)}}{\partial r_1},$$

$$X_4 \equiv \frac{\partial}{\partial r_1}\left(v_1^{(0)}\frac{\partial v_1^{(0)}}{\partial \varphi} + v_2^{(0)}\frac{\partial v_1^{(0)}}{\partial r_1} + \frac{1}{L_1^2} v_3^{(0)}\frac{\partial v_1^{(0)}}{\partial z_1}\right), \; X_5 \equiv \frac{\partial v_1^{(0)}}{\partial \phi}\frac{\partial v_1^{(0)}}{\partial r_1} + \frac{1}{L_1^2}\frac{\partial v_3^{(0)}}{\partial \varphi}\frac{\partial v_3^{(0)}}{\partial r_1},$$

$$Z_4 \equiv \frac{\partial}{\partial r_1}\left(v_1^{(0)}\frac{\partial v_3^{(0)}}{\partial \phi} + v_2^{(0)}\frac{\partial v_3^{(0)}}{\partial r_1} + \frac{1}{L_1^2} v_3^{(0)}\frac{\partial v_3^{(0)}}{\partial z_1}\right), \; Z_5 \equiv \frac{\partial v_1^{(0)}}{\partial r_1}\frac{\partial v_1^{(0)}}{\partial z_1} + \frac{1}{L_1^2}\frac{\partial v_3^{(0)}}{\partial z_1}\frac{\partial v_3^{(0)}}{\partial r_1},$$

$$h_{p1} = [1 + \lambda \cdot \cos\varphi + a_\gamma \cdot z_1 \cdot \cos(\varphi)], \quad a_\gamma = \frac{L_1}{\psi}\tan(\gamma),$$

$$0 \le r_1 \le h_{p1},\; 0 \le \varphi < \varphi_k,\; -1 \le z_1 < +1,\; s \equiv r_1/h_{p1},\; 0 \le s \le 1,\; 0 \le r_1 \le r_2 \le r_3 \le h_{c1}$$

a_γ – misalignment factor,
v_1, v_2, v_3 – dimensionless velocity vector components of ferro-oil,
r_1 – dimensinless radial coordinate,
$q_{1c}^{(0)}$ – dimensionless density of heat stream,
z_1 – dimensionless longitudal coordinate,
ϕ – peripheral coordinate,
γ – angle of misalignement,
λ – relative eccentricity,

χ	– magnetic susceptibility coefficient of ferro-oil,
ψ	– dimensionless value of radial relativ clearence,
Ω_1	– dimensionless heat supplied from outside sorces to ferro-oil,
H_1, H_2, H_3	– dimensionless vector components of magnetic field strength,
α_o, β_o	– dimensional values of ferro-oil material coefficients,
L_1	– dimensionless lenght of bearing,
R_f	– magnetic pressure number.

The total dimensional value of the carrying capacity coefficient in the slide journal bearing is determined from the commonly known relation [18,27]:

$$C_\Sigma = C_{1\Sigma} \cdot bR\eta_o\omega/\psi^2 . \tag{17}$$

The total dimensionless value of the carrying capacity coefficient $C_{1\Sigma}$ in the slide journal bearing lubricated with a ferromagnetic factor is calculated from the dependence [7,18]:

$$C_{1\Sigma} = C_1^{(0)} + Q_{Br}C_{10}^{(1)} + \zeta_p C_{11}^{(1)} + De_\alpha C_1^{(1)} + O\left(Q_{Br}^2\right) + O\left(\zeta_p^2\right) + O\left(De_\alpha^2\right) . \tag{18}$$

The total dimensional friction force Fr_Σ and total dimensionless friction force Fr_1 in the journal slide bearing gap are shown in the following relation [28]:

$$Fr_\Sigma = Fr \cdot (bR\eta_o\omega)/\psi \, ; \; Fr_1 = Fr_1^{(0)} + Q_{BR}Fr_{10}^{(1)} + \varsigma_p \cdot Fr_{11}^{(1)} + De_\alpha \cdot Fr_1^{(1)} . \tag{19}$$

Analogously, the total contractual coefficient of friction for ferro-oil taking into account the influence of magnetic field, pressure, temperature and non-Newtonian properties on the change of dynamic viscosity is determined from the following formula [7]:

$$\left(\frac{\mu}{\psi}\right)_\Sigma = \frac{Fr_\Sigma}{\psi \cdot C_\Sigma} = \left(\frac{\mu}{\psi}\right)_1^{(0)} + Q_{Br}\left(\frac{\mu}{\psi}\right)_{10}^{(1)} + \zeta_p\left(\frac{\mu}{\psi}\right)_{11}^{(1)} + De_\alpha\left(\frac{\mu}{\psi}\right)_1^{(1)}, \tag{20}$$

$$\left(\frac{\mu}{\psi}\right)_1^{(0)} = \frac{Fr_1^{(0)}}{C_1^{(0)}}, \tag{21}$$

$$\left(\frac{\mu}{\psi}\right)_{10}^{(1)} = \frac{\dfrac{Fr_1^{(0)} + Q_{Br}Fr_{10}^{(1)}}{C_1^{(0)} + Q_{Br}C_{10}^{(1)}} - \left(\dfrac{\mu}{\psi}\right)_1^{(0)}}{Q_{Br}}, \tag{22}$$

$$\left(\frac{\mu}{\psi}\right)_{11}^{(1)}=\frac{\dfrac{Fr_1^{(0)}+\zeta_p Fr_{11}^{(1)}}{C_1^{(0)}+\zeta_p C_{11}^{(1)}}-\left(\dfrac{\mu}{\psi}\right)_1^{(0)}}{\zeta_p}, \tag{23}$$

$$\left(\frac{\mu}{\psi}\right)_{10}^{(1)}=\frac{\dfrac{Fr_1^{(0)}+De_\alpha Fr_1^{(1)}}{C_1^{(0)}+De_\alpha C_1^{(1)}}-\left(\dfrac{\mu}{\psi}\right)_1^{(0)}}{De_\alpha}. \tag{24}$$

where:

μ – magnetic permeability of ferro-oil,

b – half the length of the bearing,

De_α – Deborah’s number.

4. Results of modelling

Numerical calculations of the dimensionless friction force and the dimensionless friction coefficient were performed by solving the Reynold’s type equation using the finite difference method using Mathcad 15 and own calculation procedures. There have been adopted the following dimensional and dimensionless quantities for all calculations of operational parameters: a low-speed bearing with an angular velocity of the journal ω=20s^{-1} was assumed; the journal radius was R=0.15 m and the dimensionless bearing length L_1=1; a constant dimensionless radial relative clearance value ψ=0.003has been adopted; the ferro-oil thermal conduction coefficient was established as unchangeable and was equal κ=0.15; material coefficients of ferro-oil were respectively: α=0.000020 a β=-0.000010; the value of the magnetic field intensity vector was assumed at the level ensuring full magnetic saturation of the ferro-oil H_o=280000 A·m^{-1}. In the calculations carried out, it was assumed that the bearing placement effect will not be taken into account in the model, hence the crossover angle was γ=0°. In addition, the characteristic dimensional dynamic viscosity values η_o for T=T_o=90°C and p=p_{at} were respectively: $\eta_{o(0\%)}$=0.0015Pa·s, $\eta_{o(2\%)}$=0.0194Pa·s and $\eta_{o(8\%)}$=0.0923Pa·s as well as the values of magnetic susceptibility coefficients for 2% and 8% for ferro-oil were determined experimentally and amounted to $\chi_{(2\%)}$=0.060073 oraz $\chi_{(8\%)}$=0.143877 [1].

For research purposes, the LongLife Gold Penzzoil mineral oil with SAE 15W-40 viscosity grade was selected and the same oil was used as the base for a colloidal mixture with iron oxide Fe_3O_4 solid particles. 2% - the content (by volume) of magnetic particles, adopted for calculation, is the optimal concentration of magnetic particles in the ferro-oil, in the context of the impact on the flow and operating properties of slide journal bearings lubricated with such lubricant. Such a conclusion was derived from the author's own research on the concentration of ferro-oils in tribological applications [7]. In addition, the

8% concentration of magnetic particles in ferro-oil, selected for comparative purposes, is the highest concentration of these particles tested by the author for highly concentrated ferro-oil.

Table 1 below presents, the received and previously determined by experimental studies, values of viscosity coefficients in the equations (9-11).

Table 1

Values of viscosity coefficients

Values of viscosity coefficients	Magnetic particles concentration n_{cs}		
	0%	2%	8%
values of a_B [$T^{-\delta 1B}$]	0	0,571693	1,380766
values of δ_{1B} [-]	1	0,246007	0,210347
values of a_T [-]	0,9353	0,7372	0,6732
values of δ_T [K^{-1}]	0,04805	0,05261	0,05749
values of a_p [-]	1,35221	1,60595	1,92889
values of ζ [Pa^{-1}]	$4{,}59 \cdot 10^{-8}$	$6{,}53 \cdot 10^{-8}$	$6{,}15 \cdot 10^{-8}$

Table 2 contains the values of small parameters determined and adopted in further analytical and numerical studies.

Table 2

Small parameters values

	Q_{Br}	ζ_p	De_α
0%	0.044603	0.001579	0.025856
2%	0.06121	0.002815	0.020626
8%	0.31839	0.012612	0.004334

Below, in figs. 1, 2 and 3, there are presented obtained characteristics of changes in operating parameters of the slide journal bearing, such as: dimensionless load carrying capacity, dimensionless friction force and dimensionless coefficient of friction, lubricated with base oil and ferro-oils with two adopted concentrations of magnetic particles. These graphs present the total values of the mentioned parameters, which include components taking into account the impact of changes in the dynamic viscosity of the lubricant caused by the external magnetic field, by pressure changes, by temperature changes and resulting from non-Newtonian properties of the tested lubricants. The values of these component coefficients have been multiplied by the respective values of small parameters Q_{Br}, ζ_p and De_α presented in tab. 2.

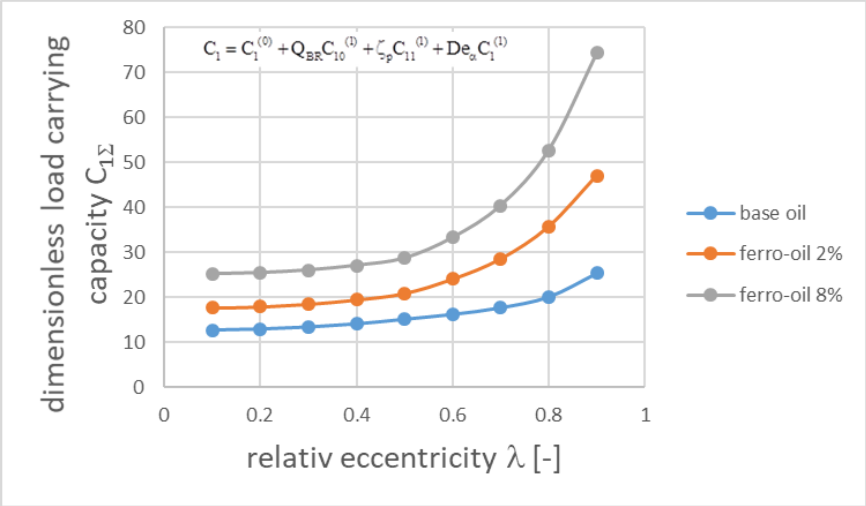

Fig. 1. Characteristics of changes of the dimensionless load carrying capacities due to the relative eccentricity

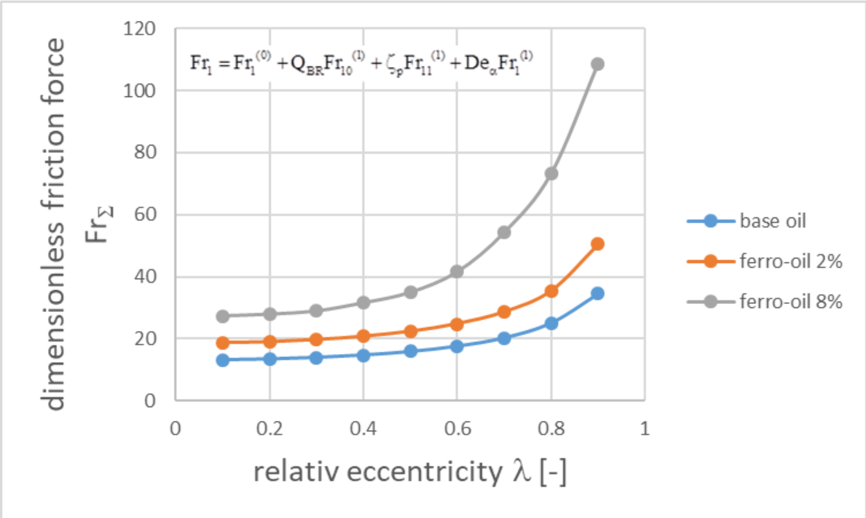

Fig. 2. Characteristics of changes of the dimensionless friction force due to the relative eccentricity

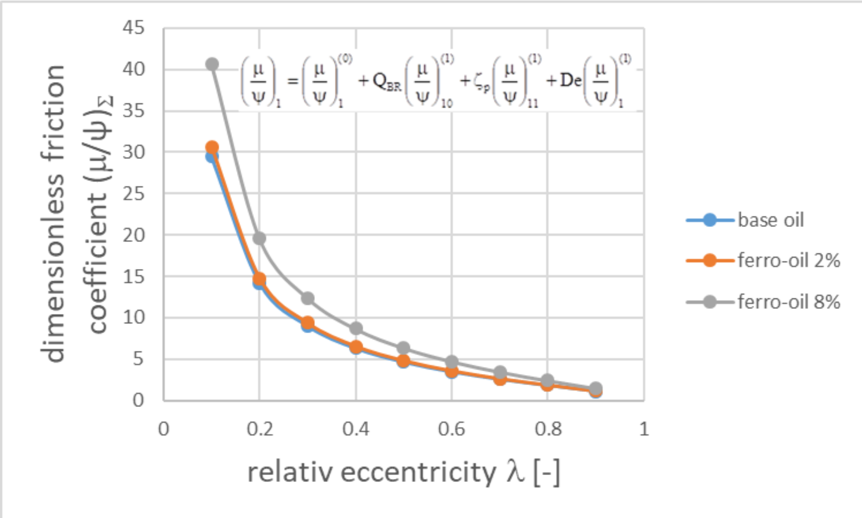

Fig. 3. Characteristics of changes of the dimensionless friction coefficient due to the relative eccentricity

5. Observations and conclusions

Analyzing the obtained characteristics of operating parameters, it can be noticed that the use of ferro-oil in place of classical mineral oil results in an increase of dimensionless load carrying capacity regardless of the value of the assumed concentration of magnetic particles. This increase is significantly greater for the highly concentrated 8% ferro-oil, in particular in the area of higher relative eccentricity λ i.e. above 0.5. For the highest value of relative eccentricity λ = 0.9, the value of load carrying capacity for 8%ferro-oil is almost three times higher than for base oil and almost twice as high as for low-concentrated 2% ferro-oil.

On the other hand, analyzing the characteristics of dimensionless friction force for bearings lubricated with the mentioned oils, it can be noticed that the use of 2%ferro-oil results in a relatively small increase in the friction force parameter over the entire and considered range of relative eccentricity values. However, the use of highly concentrated 8%ferro-oil very significantly increases its value, i.e. even four times for the highest value of relative eccentricity λ = 0.9.

The analysis of the dimensionless coefficient of friction leads to the conclusion that the use of highly concentrated ferro-oils leads to a significant increase in the coefficient of friction, which translates directly into a deterioration of friction conditions in the journal slide bearing. This effect is not observed in the case of 2% ferro-oil for which the coefficient of friction almost coincides with the values for the base oil.

Referring observations to the reliability aspect, it can be assumed that the use of ferro-oil in place of the classic bearing lubricating oil may result in an increase in the reliability and durability of the technical system in which the bearing is integrated. The selection of an appropriate, in this context, concentration of magnetic particles of such a ferro-oil will be strongly dependent on the working conditions as well as the nature of bearing use.

If the main purpose adopted in operation is to protect the bearing against damage under the influence of very high values of external incidental loads, the priority will be to obtain maximum values of bearing load carrying capacity which will constitute a specific buffer of operational safety. These conditions correspond to a combination of a high-concentrated 8%ferro-oil in a bearing and a control module that activates the appropriate, saturation, external and targeted magnetic field at critical load points.

On the other hand, working conditions characterized by high variability of bearing load values but not necessarily of extremely high values would indicate the use of 2% ferro-oil lubricating of a slide journal bearing operating in a continuous mode under the action of the external magnetic field control. This type of operating would allow adaptive adjustment of the instantaneous load capacity to the changing of bearing load conditions without the need to give up the optimal lubrication conditions. What's more, it would also be possible to maintain the value of the lubrication gap height for which the determined bearing life would be the maximum.

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