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Influence of the ultrafine oil additives on friction and vibration in journal bearings

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ABSTRACT

In order to study the influence of ultrafine additives on hydrodynamic lubrication in journal bearings, this paper combines experimental and simulative approaches. The lubricants' samples were obtained by means of adding the fullerene black, fullerenes, molybdenum disulfide and fluoropolymer in quantity no higher than 0.05% of the mass in base low-viscosity mineral oil. The experiment was made during the run-down of the rotor, and we measured the friction coefficient in the journal bearing and the vibrations of its housing. The simulation model based on the developed methodology of setting and solving of the variational problem, allowed to explain and reproduce the results of the experiment. Ultrafine additives significantly decreased the load-carrying capacity and the friction coefficient in the journal bearing, as well as the vibration level in the bearing. The rate of lowering of friction and vibration directly depended on the level of pseudoplastic properties of the sample. The best results were shown by samples with fullerene black and fluoropolymer.

1. Introduction

The basic principles of the hydrodynamic lubrication theory have been known for more than a century [1,2] and remain intact, but they constantly develop substantially. The role of the key tribological parameter – viscosity in the process of fluid friction – is contradictory and hard to formalize. On the one hand, the decrease of viscosity leads to an obvious decrease of the friction loss, on the other hand it brings the loss of the load-carrying and damping capacity of the fluid film. One way to improve operational characteristics of the lubricants is to combine the properties of solids and fluids. In theory, it can be expressed in the search of the perfect combination of the rheological models of liquid, plastic, granular, pore and other types of medium [3]. In practice, it means, that every synthetic lubricant consists of many organic and non-organic additives of various purpose [4].

In their classic work, the researchers at the Cornell University described the positive influence of the additives on friction and the load-carrying capacity of a plain journal bearing. Theoretically it may be explained by non-Newtonian properties of the modified oils. The additives were the 5-weight-percent polymer [5]. It is considered, that the dilatant lubricants help increase the load-carrying capacity relatively to an equivalent purely viscous fluid, while pseudoplastic lubricant helps reduce the load-carrying capacity [6–8]. It is obvious that the pressure field in fluid film may be re-distributed due to lubricating with the non-Newtonian fluids [9], however, it is not quite clear how it affects the dynamic properties of the bearings. In particular, in [8] it is shown, that under a constant load the pseudoplastic lubricants obtain better dynamic properties, compared to the Newtonian and dilatant lubricants. In other papers on the micropolar and polymer containing lubricants in various types of bearings there were also conclusions made on the positive influence of the additives on the dynamic characteristics of the bearings [10–13]. It is important to note, that the majority of the theoretical works are based on the Reynolds equation, which is initially not suitable for non-Newtonian fluids. This fact complicates and limits the process of solution. Moreover, there exists another point of view. For instance, it has been experimentally determined that the presence of the relatively big hard particles with size of a particle of $10\,\mu m$ in the bearing lubricant generates a much higher level of vibration [14].

The development of the nanotechnology brought new materials or familiar materials in some new capacities, in form of nanoparticles [15]. The best results were shown while applying these lubricants to the tribocoupling in the mixed and boundary lubrication regime [16–18]. The results of usage of ultrafine or nanoadditives in the tribocouplings with hydrodynamic friction are unambiguous. It is assumed, that solid ultrafine and nanoadditives provide lubricants with properties of the pseudoplastic fluids [19,20], which makes a negative impact on the load-carrying capacity of the fluid film [20].

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Nomenclature

i, j, k dummy indexes

Geometry parameters

a bipolar coordinates parameter
e_p eccentricity
h_0 average bearing gap
h or h_i Lamé's coefficients
I centroidal moment of inertia of the rotor
k_r, k_R auxiliary values in bipolar coordinates
l length of the bearing
$\vec{n}$ unit external normal vector with components n_i
R radius of the bearing
r radius of the bearing trunnion
S or S_i surface of the fluid flow region
X_i Cartesian coordinates of the center of the rotor (fixed coordinates)
x_i Cartesian coordinates of the flow region (rotating coordinates)
β_i bipolar coordinates of the flow region, $\beta_i^- \le \beta_i \le \beta_i^+$
$\tilde{\beta}_i$ dimensionless bipolar coordinates, $0 \le \tilde{\beta}_i \le 1$
γ central angle of the bearing trunnion's surface
θ_{ij} components of the cosines' matrix
Ω fluid flow region volume

Kinematic parameters

A, B, C, D, M, N auxiliary values in ψ
n revolutions per minute of the rotor (rotor speed)
T_ξ strain rate tensor with components ξ_{ij}
V velocity of the center of the rotor
$\vec{v}$ fluid flow velocity vector
v^p, v^τ normal and tangential components of surface velocity accordingly
$\alpha_k = [\alpha_k^I, \alpha_k^{II}]$ unknown coefficients in correction function Φ
$H = \sqrt{2\xi_{ij}\xi_{ji}}$ shear strain rate intensity
Φ correction function as a part of flow function Ψ
χ additional correction function as a part of flow function Ψ
Ψ flow function
ψ main solution as a part of flow function Ψ
ω angular velocity of the rotor

Static parameters

D_σ deviator part of the stress tensor with components s_{ij}
F resulting force vector
f friction coefficient
M resulting torque
p_0- fluid flow pressure
T_σ stress tensor with components σ_{ij}
$\vec{\sigma}^n, p^n, \tau^n$ full, normal and tangential surface stresses accordingly
$T = \sqrt{s_{ij}s_{ji}/2}$- shear stress intensity

Dynamic parameters

B_{ij} components of the damping matrix
Ext or Ext_i power of external forces
g free fall acceleration
Int power of internal forces
J target functional
K_{ij} components of the stiffness matrix
L integral limitation of the target functional's value
m mass of the rotor
$m\Delta$ imbalance of the rotor
q, z viscosity power law parameters
λ Lagrange's multiplier
μ dynamic viscosity coefficient (viscosity)

The aim of the present paper is to implement an integrated theoretical and experimental research of the influence of the solid ultrafine additives of different types and dimensions on hydrodynamic friction processes and vibrations in journal bearings.

2. Lubricants' samples and their rheology

As a basic lubricant we used low-viscosity oil I-12A of the same viscosity class as, for example, oil «Shell Vitrea 22». Oil I-12A is applied to the journal bearings of light, high-speed rotors and the machines, where anti-oxidant and anti-corrosion properties of the lubricant are not really important. As additives we used the following four types of ultrafine materials (Fig. 1): fullerene black; mixture of fullerenes of the following content: C_{60} (63.12%), C_{50}–C_{58} (14.69%), C_{70} (13.25%), $C_{62}-C_{68}$ (5.88%), C_{72}–C_{92} (3.06%); molybdenum disulfide MoS_2; polytetrafluorethylene (fluoropolymer) $(C_2F_4)_n$. Additional information on basic and additional materials is reflected in Table 1.

The preparation of the lubricants excluded the use of solvents due to low solubility of some additives. Instead, the samples were mixed mechanically in order to become homogenous. Mass fraction of additives in samples did not exceed 0.05%.

The results of the viscosity tests of these 5 materials are shown in Fig. 2. We used inertial viscometer, which is the modification of the capillary viscometer, save the fact that the test medium moves in a closed circuit due to the inertial forces [21].

The experimental results are convenient to approximate using the power law (Fig. 2) [22]:

$$\mu = q\xi_{12}^{z-1}, \sigma_{12} = 2q\xi_{12}^{z}, \tag{1}$$

where μ-dynamic viscosity coefficient (viscosity), q and z-power law parameters, ξ_{12}-strain rate tensor component, in this case, the only non-zero component, σ_{12}-stress tensor component.

The following form of the power law [27] is the generalization of the Eq. (1) for the case of an arbitrary stress–strain state:

$$\mu = 2^{z-1}qH^{z-1}, T = 2^{z-1}qH^{z}, \tag{2}$$

where T-shear stress intensity, H-shear strain rate intensity.

According to the hypothesis of single curve [27], the form of the rheological functions in terms of intensity is invariant to the type of the stress–strain state and may be defined only by properties of the material.

Parameters of the rheological models of lubricant samples are shown in Table 2. The z parameter of the power law characterizes the type of non-Newtonian behavior of the fluid: when $z<1$, the viscosity of the fluid decreases with the increase of the shear rate, and the fluid may be called pseudoplastic, when $z>1$, the viscosity of the fluid increases with the increase of shear rate, and the fluid may be called dilatant [22].