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A STUDY THE AXIAL AND RADIAL ROTOR STABILITY OF THE TURBO-MACHINERY WITH ALLOWANCE THE GEOMETRY OF THE SURFACE AND PROPERTIES OF THE LUBRICATING FLUID

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Abstract: *An ensuring of the axial and radial rotor stability is one of the main objectives in the design and operation of small-sized turbo-machinery. Hydrodynamic thrust bearings must securely restrain unbalanced axial forces arising from any possible operating modes. The using of different types of intermediate elements provides a stable position of the rotor in the radial bearings. Recently flexible rotors are widely used in small-sized turbo-machines. The complexity of the implementation of calculation methods for the dynamics of these rotors is the solving a system of motion equations. The system of motion equations, which is presented in this work, contains the motion equation of each bearing element, including floating bushings and elements of the rotor (wheels of the turbine and compressor, the central mass). As loads on bearings, we took into account not only the masses of the elements themselves, the reactions of lubricating layers and a journal imbalance, but also possible impact loads, forces and moments of force couples acting on the journal from other elements of the rotor. Gyroscopic moments caused by the precession of the rotor, the reaction of lubricant layers, as well as the force of the elastic interaction of the elements are also included in the equations of motion. To calculate the reactions of lubricating layers for two-layer bearings the hydrodynamic pressure field and the friction losses were considered for a real bearing design, on the surface of which the sources of lubrication is always located. As the thrust bearing the segments of various designs, including surfaces with the laser texturing, have been discussed. Trajectories of rotor elements of the turbocharger for various rotation velocities, as well as the elastic line of the rotor at different time points are represented as the results of calculation.*

Keywords: *thrust bearings, radial bearings, flexible rotors, load capacity, hydro-mechanical characteristics.*

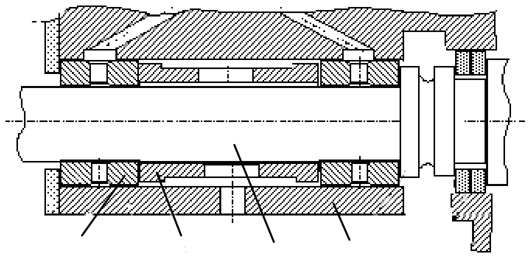
1. INTRODUCTION

Multilayer plain bearings are widely used to enhance the stability of the movement of the rotor of the turbo-machine. The rotating and non-rotating rings are used as intermediate elements of bearings. The number of intermediate elements, their design and the way they are installed in to the hull turbo-machine depends on operating conditions and design parameters. Three layers of the bearing are the most promising. It provides the greatest stability of the motion of the rotor.

At the same time, the using of multigrade oils that are prevalent in recent years, leads to a change of hydro-mechanical characteristics of bearings. This is due to the rheological properties of lubricants. The consideration of these properties approximates the math models to the real conditions of work of friction units.

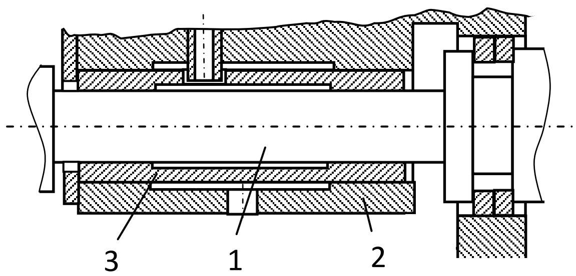
This study is part of the work to create the methodology for calculating hydro-mechanical characteristics of multilayer plain bearings, which are operating on non-Newtonian lubricants.

To reduce vibrations the rotor of a turbocharger the most common way is to use intermediate floating elements in the form of rotating floating (RF) (Fig. 1) or a non-rotating floating (NF) (Fig. 2) rings. Working surfaces of each ring with surfaces of the housing and the rotor are formed the several lubricating layers.



1 – rotor, 2 – hull, 3 – rotating rings, 4 – stopper ring

Figure 1. Bearing with rotating rings



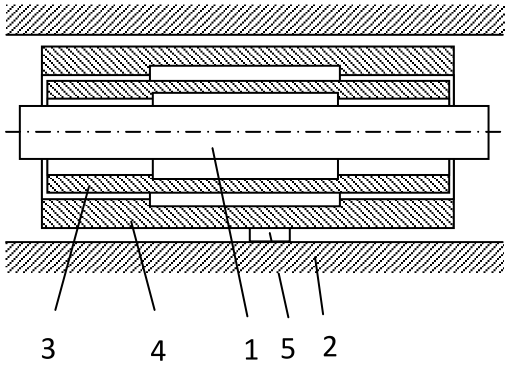
1 – rotor, 2 – hull, 3 – non-rotating ring

Figure 2. Bearing with non-rotating ring

Use of bearings with a package of the floating rings containing three lubricant layers is also perspective. In this case the third lubricant layer is an additional damper. The external ring is designed in the form of non-rotating mono-rings. Two autonomous rotating rings or non-rotating mono-ring are used as internal mobile elements (Fig. 3).

Dynamics of the rotor on bearings with floating rings are investigated in details [1-4]. In turbochargers of internal combustion engines the non-rotating floating rings are also used [5].

Edgar J. Gunter presents the linear and nonlinear dynamical behaviour of a typical turbocharger in floating ring bearings [6]. He computed the linearized a stability of the system for various ringing inner and outer clearance ratios.



1 – rotor, 2 – hull, 3,4 – non-rotating rings, 5 – pin

Figure 3. The package of the floating ring

He also represents the turbocharger model for critical speed analysis and showed several fluctuations forms of rotor.

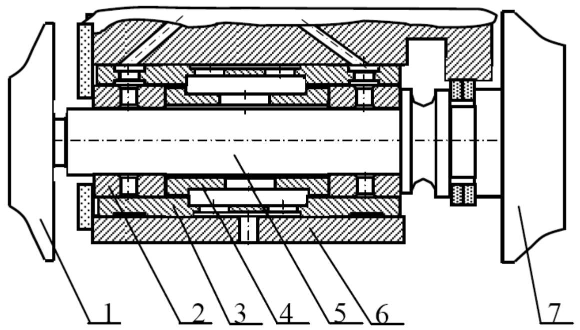
Tartara [7] made experiments and conclusions, that floating ring bearings are designed such that the rings may be properly floating and rotate correctly, remarkable effects of stabilizing the system can be expected. Namely, under suitable conclusions, the ring starts rotating as soon as the whirl occurs and as the shaft speed increases further stable operation is realized. The stabilizing effect of the floating-ring bearing is conspicuous when the ratio between inner and outer clearance and the absolute clearance are large.

The bearing design with three lubricating layers provides more stability of the rotor. It was shown in [8]. Example of calculation of the dynamics of flexible asymmetric rotor is presented in [9]. Newtonian and non-Newtonian fluids have been considered as a lubricant the bearings of the rotor.

However, theoretical studies of the stability of the rotor based on the properties of the lubricant developed not enough.

2. THEORETICAL ANALYSIS

The problem of the theory of hydrodynamic friction units is characterized by a set of methods for solving three interrelated tasks: to determine the conditions of stability and parameters of nonlinear oscillations the pin on the lubricating film, the calculation of its trajectory, the calculation of the field of hydrodynamic pressure in the lubricating layer, which is separating the surface friction of the shaft and bearing, with an arbitrary law of their relative motion, the calculation of the lubricating film temperature. The design of the turbocharger rotor with three layers bearings is shown in Figure 4.



1 – compressor's wheel, 2 – floating rotating rings, 3 – floating non-rotating ring, 4 - stopper ring, 5 – rotor, 6 – hull, 7 – turbine's wheel

Figure 4. The design of the rotor on multilayer plain bearings

2.1 The dynamics of the system

The dynamics of the rotor is considered as the movement of a pin on multilayer bearings, depending on the operating forces and initial conditions. Stability of motion of each movable element is a prerequisite for the effectiveness of the system as a whole. As a rule, the system is stable, if it deviates from its equilibrium position by an arbitrarily small value for any sufficiently small change the load. Effectiveness of different designs of bearings with intermediate elements is estimated by calculating the trajectories for which the geometric center of pin and rings are moving under the action of loads, calculating the characteristics of stability and a set of hydro-mechanical characteristics

A dynamic model of the rotor of the turbo-machine with asymmetrically arranged wheels, which based on two three-layered bearings, is presented in the form of five masses, connected by a weightless flexible rod (Fig. 5). The rotor, rings and hull are separated by thin layers of lubricants. Misalignment axis of the shaft and rings are not included. The axis *OZ* of the inertial coordinate system *OXYZ* is drawn through the geometric center of the bearing hull. The motion of a flexible rotor represent a superposition of the motion of the rotor axis as a rigid body, within clearances of bearings, and the elastic displacements of all elements of the rotor relative to the rigid axis.

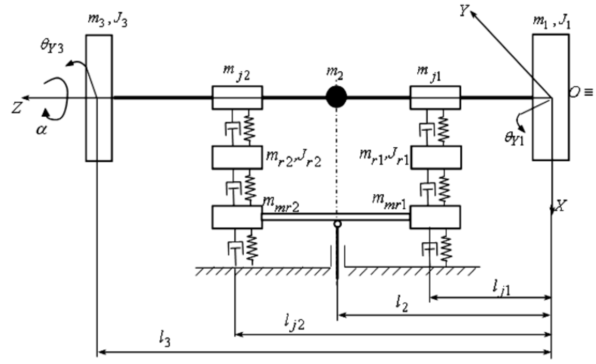


Figure 5. A dynamic model of the turbo-machine rotor

The system of equations of motion for a flexible asymmetric rotor, based on multilayer plain bearings, is presented in [10].

An analysis of the trajectories of each element tribo-system obtained by direct numerical integration of the equations of motion is the most accurate method for evaluating the performance of bearings with intermediates elements. For arbitrary initial values of the coordinates and velocities of the centres of mobile elements, you can not anticipate the kind of trajectories in which they move, and therefore we can not specify the number of time steps, sufficient for the analysis of these movements. We have developed an algorithm for integrating the equations of motion, which contains a specific procedure for automatic determination of the end of the path of finding. According this procedure, the individual turns of the trajectory must be placed inside a closed rectangular region.

2.2 Properties of the lubricating fluid

The lubricant is considered a design element of any friction units. At present, multigrade engine oils are widely used in the operation of the internal combustion engine turbochargers. It is known that such oils are non-Newtonian fluids. The main feature of the rheological behaviour of these oils is the non-linearity of viscosity as a function of shear rate. In other words, the viscosity is a function not only of temperature and pressure, but and shear rates, the values of which reach $3 \cdot 10^6 s^{-1}$ for the rotor bearing of the turbochargers. For these values of shear rates the viscosity of lubricant may be significant reduced. This fact must be taken into account when the hydro-mechanical characteristics of the bearings of the rotor are calculated.

Thus, the need for a comprehensive rheological model lubricant emerged. This model must take into account both the viscosity depends on the temperature lubrication, hydrodynamic pressure in a thin lubricating layer, shear, changing dramatically the thickness of the layer, relaxation (delay of viscosity change with rapid growth of hydrodynamic pressures) and highly viscous boundary layer. Following viscosity model was proposed:

$$\mu^*(T,p,\dot{\gamma}) = \begin{cases} \mu_1 \cdot C_1 e^{(C_2/(T_э+C_3))+\theta(T_э)\cdot p}, \\ \qquad 1 \le \dot{\gamma} \le 10^2; \\ (I_2)^{(n(T_э)-1)/2} \cdot C_1 e^{(C_2/(T_э+C_3))+\theta(T_э)\cdot p}, \\ \qquad 10^2 \le \dot{\gamma} \le 10^6; \\ \mu_2 \cdot C_1 e^{(C_2/(T_э+C_3))+\theta(T_э)\cdot p}, \\ \qquad \dot{\gamma} > 10^6, \end{cases} \quad (1)$$

where $\dot{\gamma} = \sqrt{I_2}$, $I_2 = (\partial V_x/\partial y)^2 + (\partial V_z/\partial y)^2$.

At section 1 in the range of shear rates from 1 to $10^2 s^{-1}$ oil behaves as a Newtonian fluid with a viscosity μ_1. For section 2 in the range of shear rates of 10^2 to $10^6 s^{-1}$ is characteristic decrease in viscosity as a power law. In section 3 at a shear rate greater than $10^6 s^{-1}$ oil behaves as a Newtonian fluid with a viscosity μ_2.

Accounting for highly viscous boundary layer is calculated as follows [11]:

$$\mu_i = \mu^*(T,p,\dot{\gamma}) + \mu_S \exp\left(-\frac{h_i}{l_h}\right), \quad (2)$$

where l_h - characteristic parameter with dimension of length, the value of which is specific to each combination of lubricating oil and solid surface μ_s – parameter representing the equivalent viscosity at infinitely small distance from the boundary surface; μ_0 - viscosity oil in volume.

2.3 The calculation the temperature of the lubricating film

Thermal processes in oil film of heavy-loaded bearing are considered on based of the solutions of the generalized energy (heat) equation for a thin layer of viscous incompressible fluid between two moving surfaces**.** It takes into account both the convective heat transfer, which are implemented lubrication, and heat transfer by conduction. The temperature distribution in the lubricating film is described by the equation [10]:

$$\rho c_0 \frac{\partial T}{\partial t} + \rho c_0 \left(V_x \frac{\partial T}{\partial x} + V_y \frac{\partial T}{\partial y} + V_z \frac{\partial T}{\partial z} \right) - \lambda_0 \frac{\partial^2 T}{\partial y^2} = D. \quad (3)$$

where ρ - the density of lubricant; c_0 , λ_0 - specific heat capacity and thermal conductivity grease (usually assumed constant); $T(x,y,z,t)$ - the temperature at coordinates x,y,z; t - time; V_x, V_y, V_z - velocity components unit volume lubrication, located between two moving surfaces conjugation; D - dissipation function.

Depending on the assumptions used for the temperature distribution in a thin lubricating layer, three approaches to the integration of the generalized energy equation can be applied: thermo-hydrodynamic (non-isothermal), adiabatic, isothermal.

For the isothermal approach the three-constant Fogel's formula works very well

$$\mu(T) = C_1 \cdot \exp(C_2/(T + C_3)), \quad (4)$$

where C_1, C_2, C_3 - constants, which are empirical features of lubricant.

When thermo-hydrodynamic approach is used the temperature change is expected in all directions [12], including across the lubricating layer. In this case, the boundary conditions are the most adequate to the actual thermal processes. This approach provides information on the local properties of the temperature field of the lubricant layer: the maximum, instantaneous average temperatures, areas of increased thermal stress.

To determine the temperature variation in thickness of the lubricating layer we perform a discretization across the layer. This allows you to take into account the temperature field, dependence of the viscosity of the lubricant from the second invariant of the shear rate, and the effect highly viscous boundary layer, which adsorbed on the surfaces of friction, that is, to take into account non-Newtonian properties of the lubricating oil.

2.4 Determination of hydrodynamic pressure in the lubricating layer

Among the techniques that take into account properties of the lubricants is best known approach of modifying the equations for the field of hydrodynamic pressure through a variety of rheological laws of the behaviour of lubricant. For example, the equation of Reynolds considering the power law of viscosity takes the form:

$$\frac{1}{r^2}\frac{\partial}{\partial\varphi}\left[\left(\varphi_2-\frac{\varphi_1}{\varphi_0}\varphi_1\right)\rho\frac{\partial p}{\partial\varphi}\right]+$$
$$+\frac{\partial}{\partial z}\left[\left(\varphi_2-\frac{\varphi_1}{\varphi_0}\varphi_1\right)\rho\frac{\partial p}{\partial z}\right]= \tag{5}$$
$$=\frac{\partial}{\partial t}\left(\bar{h}\rho\right)+\frac{\partial}{\partial\varphi}\left\{\left[\omega_1+\left(1-\frac{\varphi_1}{h\varphi_0}\right)\omega_{21}\right]\rho h\right\}.$$

where $\bar{\varphi}_k=\int_0^1\frac{\bar{y}^k}{\bar{\mu}^*}d\bar{y}$, $\bar{h}(\varphi,\bar{t})=1-\chi\cos(\varphi-\delta)$ – dimensionless film thickness; χ – the relative eccentricity.

Equation (5) is integrated with the boundary conditions of Swift-Shtiber:

$$\begin{aligned}&\bar{p}(\varphi,\bar{z}=\pm a)=0;\ \bar{p}(\varphi,\bar{z})=\bar{p}(\varphi+2\pi,\bar{z});\\&\bar{p}(\varphi,\bar{z})\geq 0;\ \bar{p}(\varphi,\bar{z})=\bar{p}_s.\end{aligned} \tag{6}$$

where $(\varphi,\bar{z})\in\Omega_S$, $S=1,2\ldots S^*$ – source lubrication area in which the pressure is constant and equal to the pressure supply $\bar{p}_S$; S^* – number of sources.

We define the trajectory of moving parts, the position of the rotor axis and the hydro-mechanical characteristics of friction units when we solve together the equations of motion, the equations for determining the pressure field and the generalized energy (heat) equation for finding the temperature in each film of lubricant.

2.5 Mathematical model for calculating thrust bearings

Usually thrust bearing consists of the shoe rotating together with a rotor and a motionless thrust bearing. Hydrodynamic thrust bearings with macro-profiling of a surface of friction of a thrust bearing were widely adopted. Usually the thrust bearing is carried out in the form of segments: inclined surfaces, "Rayleigh's steps" or other profile. At the same time micro-profiling of surfaces of friction receives bigger distribution for increase of the bearing ability.

Laser texturing (creation of certain properties of a micro-profile of a surface of friction by means of processing by laser impulses) is the most perspective for increase of the bearing ability of various tribo-units.

Therefore we used this type of profiling for calculation of thrust bearings of sliding of turbocharger of internal combustion engines. Calculations of load capacity and load characteristics for the four types of structures of hydrodynamic thrust bearing segment were performed.

A scheme of the hydrodynamic thrust bearing is shown in Fig. 6. The rotor rotates at a constant speed relative to the housing turbocharger. An axial load on the rotor is constant and balanced reactions lubricating layer. Thrust bearing axis coincides with the rotational axis of the rotor.

The surface of a thrust bearing consists of eight blocks (segments). Lubricant under pressure moves through an internal ring groove of a thrust bearing. Pressure on the external radius of a thrust bearing is accepted equal to the atmospheric. Blocks are divided by radial grooves. The pressure in grooves is distributed under the linear law. The load capacity of the bearing is formed due to formation of system of hydrodynamic wedges when involving lubricant in the narrowed axial gap.

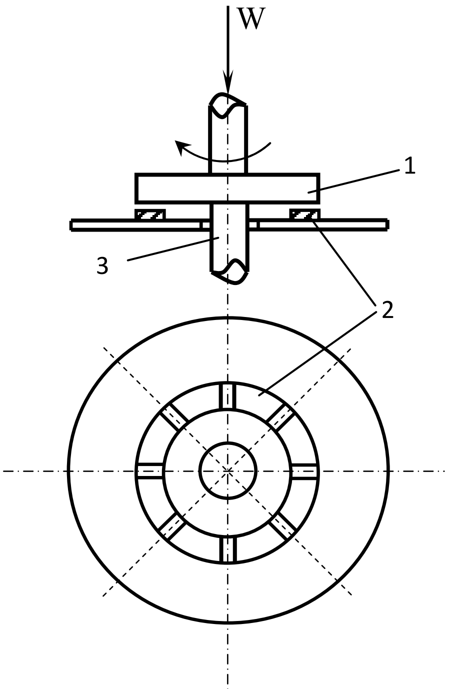


Figure 6. Scheme thrust hydrodynamic bearing: 1 – heel; 2 – blocks; 3 – rotor

The planes of blocks can be processed by means of various technologies. We considered 4 types of persistent bearings (Fig. 7). A design 1 is Rayleigh's step ($h_1/h_y=1{,}87$, $l_2=l/3{,}59$). A design 2 is a block with two inclined planes in the circumferential direction ($\gamma=0{,}3^\circ$ $h_1/h_y=1{,}87$, $l_2=l/3{,}59$). A design 3 is the inclined plane in the circumferential direction $\gamma=0{,}3^\circ$. A design 4 is a block surface with laser texturing.

Extensive experimental and theoretical studies of thrust bearings have been performed by Etsion and his colleagues [13].

The textured surface represents the plane with micro-deepening of the set diameter, depths and density of their distribution [13]. Laser texturing can be used for part of the bearing or for all plane of the bearing.

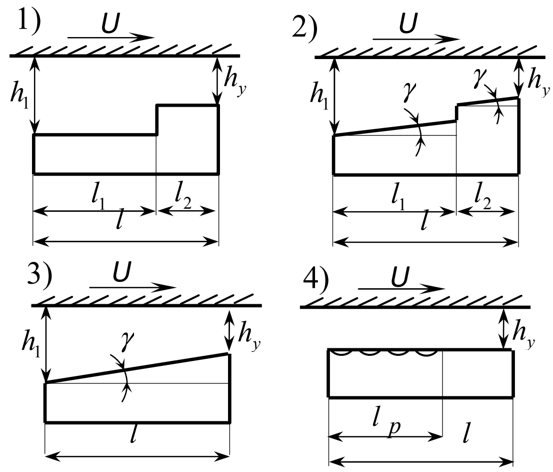


Figure 7. Design of blocks

The following parameters of laser texturing were accepted in calculations: socket depth $h_p=50\cdot10^{-6}\,м$,socket radius $r_p=50\cdot10^{-6}\,м$, density of distribution of sockets $S_p=0{,}6$, $l_p=l\cdot0{,}6$.

3. RESULTS

The equation (5) was written in term of finite differences and integrated by a Gauss-Seidel method using of three levels of grids. The finest grid contains 96 intervals on an axis φ and 25 ones on an axis z. Each time, when condition $p>0$ during calculation is not satisfied in any point of a difference grid, the pressure in this point is setting equal to zero. All integrals were evaluated by Simpson method.

The distribution of the hydrodynamic pressure in a lubricant layer with sources of lubricants on a surface of the bearing is given in Figure 8.

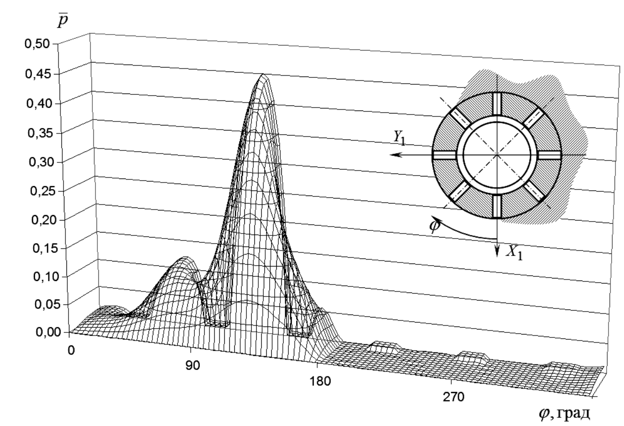


Figure 8. Distribution of the hydrodynamic pressure

The ratio of radial clearances $C2/C1$ were changed in the course of researches. Results showed that the increase in the relation of external and internal clearances leads to sharp increase in amplitude of fluctuation of a rotor $A1$, $A2$ (Fig. 9).

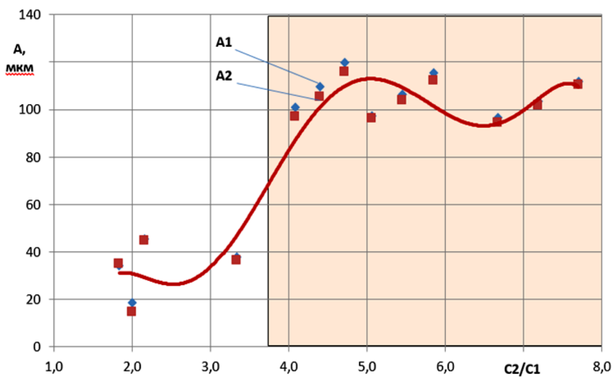


Figure 9. Amplitudes of fluctuation of a rotor

Speed of rotation of a rotor also influences to amplitude of fluctuations of a rotor. Additionally with increase in frequency of rotation the form of fluctuations of a rotor changes too. Changing the position of the rotor elements of a turbocharger with time is shown in Figure 10.

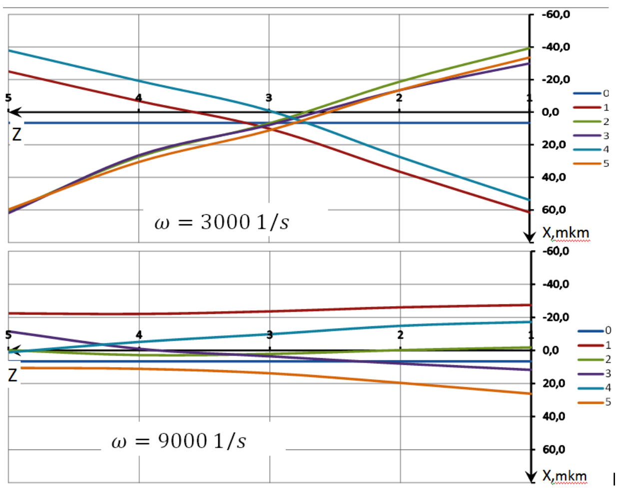


Figure 10. Elastic lines of a rotor in different time points

Trajectories of the movement of the centres of elements of a rotor in designations of Figure 5 are presented on Figure 11.

At increase in speed of rotation of a rotor from 3000 to 9000 1/s amplitudes of fluctuations of its elements decrease by 2.5 times. The form of fluctuations changes from conic on the cylindrical. The further increase in speed of a rotor leads to sharp increase in amplitudes.

During numerical researches of the thrust bearing the new technical solution was proposed. It combines advantages of "Rayleigh's step" and a friction surface with laser texturing.

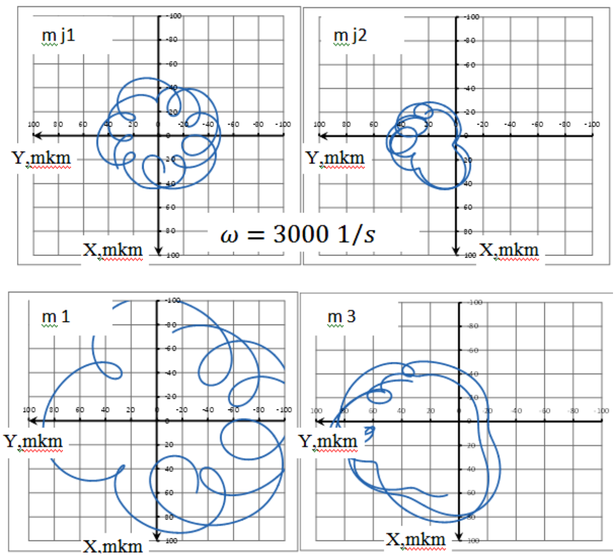


Figure 11. Trajectories of the movement of the centres of rotor elements

The schematic diagram of a segment is submitted in Figure 12.

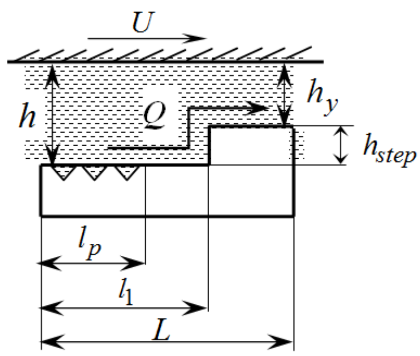


Figure 12. The scheme of a segment – "Rayleigh's Step" with laser texturing

Advantage of the developed design was shown on the example of comparison of results of calculation of the load capacity and the maximum pressure for four forms of segments: I – laser texturing of a surface, II – an inclined surface, III – "Rayleigh's step", IV – "Rayleigh's step" with laser texturing of a surface (Table 1).

4. CONCLUSION

Non-Newtonian properties of the lubricant have a significant impact on the stability of flexible rotors of turbomachines. The calculation results showed that the properties of the lubricant affect on the trajectory of moving parts of the bearing and on its hydro-mechanical characteristics.

Table 1. Results of calculation of the thrust bearing

Characteristics	The design of the thrust bearing			
	I	II	III	IV
$P_{\max} / P_{\max\, IV}$	0.62	0.66	0.99	1
W/W_{IV}	0.47	0.59	0.60	1

Accounting for one of the properties of the lubricant does not reflect the processes taking place in a thin lubricating film. Each of these properties of lubrication and viscosity dependence of one of the parameters (and more) or improves or worsens the hydro mechanical characteristics of friction units. Therefore, the choice of the rheological models used in the calculation of complex-loaded friction units of machines, depending on the type, operating conditions and lubricant tribo-unit as well as the objectives pursued by the design engineer.

Results of the authors and other researchers show that the properties of lubricants affect the characteristics of friction units as much as a geometric factor. However, the parameters of rheological models should be determined on the basis of experimental data for each grade lubricants.

ACKNOWLEDGEMENT

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