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ON THE THERMAL CALCULATION OF THE FLUID FRICTION SLIDING BEARINGS

The article deals with an approach to the heat flux distribution in fluid friction sliding bearings (FFSB) as a basis of the thermal calculation based on electrothermal analogy considering all the possible sources of thermogenesis in the FFSB and the way of its removal.

Keywords: heat flux, heat flux sources, lubricant temperature.

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ДО ПИТАННЯ ТЕПЛОВОГО РОЗРАХУНКУ ПІДШИПНИКІВ КОВЗАННЯ РІДИННОГО ТЕРТЯ

У статті розглянуто підхід до розподілу теплових потоків в підшипниках ковзання рідинного тертя (ПЖТ), як основа теплового розрахунку на базі електротеплової аналогії, що враховує всі можливі джерела теплоутворення в ПЖТ і шляхи його відводу.

Ключові слова: тепловий потік, джерела теплового потоку, температура мастила.

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К ВОПРОСУ ТЕПЛОВОГО РАСЧЕТА ПОДШИПНИКОВ СКОЛЬЖЕНИЯ ЖИДКОСТНОГО ТРЕНИЯ

В статье рассмотрен подход к распределению тепловых потоков в подшипниках скольжения жидкостного трения (ПЖТ), как основа теплового расчета на базе электротепловой аналогии, учитывающей все возможные источники теплообразования в ПЖТ и пути его отвода.

Ключевые слова: тепловой поток, источники теплового потока, температура смазки.

Relevance of the study

The thermal calculation is a basis for the calculation of fluid friction sliding bearing (FFSB) as a key parameter of a bearing, its bearing capacity, is determined by a design viscosity of the lubricant.

The purpose and objectives of the article

Thus, the primal problem of thermal calculation is a lubricant temperature test in both loaded and unloaded zones, which later determine the lubricant viscosity and the necessary amount of lubricant pumped through the bearing. The use of noncontact infra-red method of temperature measurement, the development of the device for direct temperature measurement of lubricant layer [1, 2] made it possible to determine experimentally that the temperature of the lubricant layer is not constant around the circumference of the FFSB under any operating conditions.

Substantiated exposition of the basic material and received results

The synchronous measurement of three operating parameters of the lubricant layer-the thickness, the pressure developed in it, and the temperature - using proposed methods and devices [3, 4, 5], confirmed the results of the method of thermal calculation of heat fluxes in the FSSB, considered below [6].

All existing methods of thermal calculation of the FFSB use the heat balance equation:

$$\theta = \theta_1 + \theta_2 + \cdots + \theta_n, \tag{1}$$

where θ $-$ is a heat (heat flux) separated as a result of frictional forces;

$\theta_1 + \theta_2 + \cdots + \theta_n$ - is a heat (heat flux) taken aside of FFS B by pumped lubricant, by heat transfer through rubbing surfaces etc.

The equation (1) shows that the amount of heat exuded from the FFSB is equal to the amount of heat exuded in it. A similar equation is known from the theory of electrical circuits [7] as the first Kirchhoff's law and is formulated as follows: "the sum of electric currents flowing to the node is equal to the sum of currents flowing from the node".

$$I = I_1 + I_2 + I_3 + \cdots + I_n, \tag{2}$$

where I – is a sum of currents flowing into a node;

$I = I_1 + I_2 + I_3 + \cdots + I_n$ – is currents flowing from the node.

If we compare the dependence [8] describing the process of thermal conductivity known as the Fourier's law and linking the heat flux density with the temperature gradient:

$$q_T = \lambda \overrightarrow{n_0} \frac{\partial T}{\partial n}, \tag{3}$$

and the dependence describing the process of electrical conductivity known as Ohm's law in differential form and relating the electric current density to the potential gradient:

$$q_Э = \sigma_э \overrightarrow{n_0} \frac{\partial \varphi}{\partial n}, \tag{4}$$

where q_T - is the heat flux density;

$q_Э$ - is the electric current density;

λ - is the thermal conductivity coefficient;

$\sigma_э$ - is the specific electrical conductivity;

T - is the temperature value at a given point;

φ - is the value of the electric potential at a given point;

$\overrightarrow{n_0}$ - is the unit vector, it is clear that processes of different physical nature - thermal conductivity and electrical conductivity - are described by the equations of the same structure.

On the assumption of the heat balance equation (1), the Kirchhoff's equation (2), the equations (3) and (4), we can draw an analogy between the thermal and electric parameters: : the temperature T is similar to the electrical potential φ, the heat flux Θ is similar to the electric current I, the heat flux density q_T is similar to the electric current density $q_э$, the thermal resistance R and thermal conductivity σ are similar to the electrical resistance $R_э$ and electric conductivity $\sigma_э$, the heat flux source Θ is similar to the electric current source *I*.

Developing the similarity further, it is clear that the known equations of:

a) the heat transfer by fluid with the thermal capacity C and the density γ in amount Q from a point with temperature T_1 to a point with temperature T_2

$$\Theta = c\gamma Q(T_1 - T_2), \tag{5}$$

b) the heat transfer in a solid with a thickness δ with a thermal conductivity λ, through a cross section with the square S from a point with temperature T_1 to a point with temperature T_2.

$$\Theta = \frac{\lambda}{\delta} SQ(T_1 - T_2), \tag{6}$$

c) the heat transfer by means of heat exchange between fluid and a solid with heat-exchange surface *S* heat transfer coefficient α from fluid with a temperature of T_1 to a body with a temperature T_2 or vice versa from a body to a fluid

$$\Theta = \alpha S(T_1 - T_2), \tag{7}$$

are similar to the known in the theory of electrical circuits expression [7] for electric current through a branch with a conductivity from a point with an electric potential φ_1 to a point with an electric potential φ_2.

$$I = \sigma(\varphi_1 - \varphi_2), \tag{8}$$

From the expressions 5,6,7,8 follows that the expressions $c\gamma Q$, $\frac{\lambda}{\delta} SQ$, αS are the thermal conductivities for each particular type of heat exchange.

There are well developed techniques in the theory of electrical circuits that allow on the basis of of the Kirchhoff's laws to calculate currents in branches or the values of electrical potentials at nodes -points of intersection of branches in circuits where the values of the resistance or conductivity of the branches are known and the values of current or voltage sources are set.

Using the similarity between thermal and electrical values, we will plot a thermal scheme reflecting the thermal bonds in the fluid friction sliding bearings and, using the methods of solving electric circuits, we will solve the problem of the values and distribution of lubricant temperatures in the FFSB.

The thermal scheme of the PZHT, plotted on the basis of the analogy of thermal and electrical processes, is shown in Fig. 1.

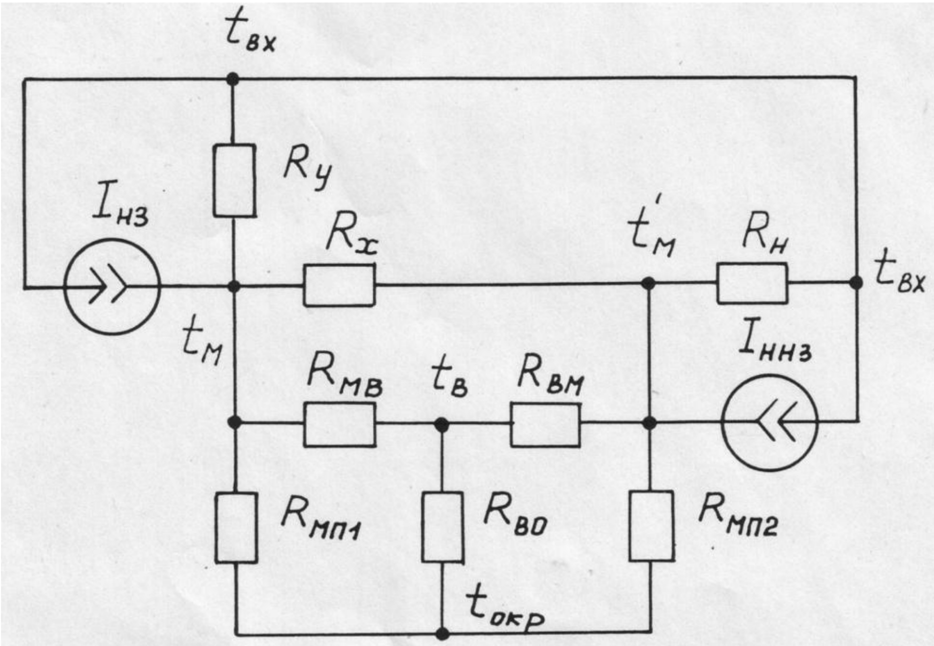

Fig. 1. The thermal scheme of the FFSB regarding the heat transfer along the shafts

The current sources $I_{\text{нз}}$ and $I_{\text{ннз}}$ are the sources of heat flux equal to the amount of heat exuded in lubricant as the result of friction force in loaded and unloaded zones correspondingly (the loaded zone is a development zone for hydrodynamic pressures, the unloaded zone is a zone through which the main quantity of lubricant is being pumped). As it was mentioned above, the heat flux sources are similar to the current sources and therefore have similar properties, i.e. their internal resistance equals infinity that, for the case of a thermal scheme, means that the heat flux cannot be limited by external thermal resistances or conductivities and is determined by the internal properties of the heat flux source. In our case, P is the external load, f is the coefficient, u is the sliding velocity, i.e. the heat flux source can develop a heat flux equal to $\Theta = Pfu$.

Another property of the current source is that when a resistance is connected to it, a potential proportional to the current of the current source and the value of the connected resistance will develop at the output of the current source. Similarly, a temperature proportional to the heat flux of the given source and the thermal resistance connected to it will develop in the source of the heat flux. That is, at a point or on a surface where the heat flux flows, the temperature will be higher than at a point or surface, from which the heat flux flows, that corresponds to the Fourier's law of thermal conductivity (3). Obviously, the lower the thermal resistance or the greater the thermal conductivity connected to the heat flux source, the smaller will be the temperature difference on the thermal resistance and the lower will be the temperature developed by the heat flux source. Or as applied to the FFSB, if heat is being removed from the loaded bearing zone by a large amount of lubricant Q-formula (5), the thermal conductivity with $c\gamma Q$ becomes large, and since the amount of heat that must be removed is constant, the temperature difference ($T_1 - T_2$) will decrease. But the T_2 value is constant, as, for example, the ambient temperature. Therefore, the temperature of the lubricant T_1 in the loaded zone will also decrease.

The nodes in the thermal scheme denote the corresponding nodal temperatures. The scheme is a scheme with lumped parameters. That is why the nodal temperatures and thermal resistances are average at a part of the FFSB: $t_{\text{м}}$ is the average lubricant temperature in the loaded zone; $t_{\text{м}}^1$ is the average lubricant temperature in the unloaded zone; $t_{\text{в}}$ is the shaft surface temperature; $t_{\text{вх}}$ is the average lubricant temperature at the entry to the bearing; $t_{\text{окр}}$ is the ambient temperature; R_x the thermal resistance for heat flux Θ_x, which is transferred from the loaded zone of the FFSB to the unloaded as a result of the lubricant overflow captured by the rotating shaft ($1/c\gamma Q_x$, where Q_x is the amount of the lubricant overflow); R_y the thermal resistance for heat flux Θ_y, which is transferred from the loaded zone of the FFSB as a result of the lubricant overflow in the ends of the loaded zone ($1/c\gamma Q_y$, where Q_y is the amount of the lubricant overflow), $R_{\text{мв}}$ is the thermal resistance for the heat flux transferred from the lubricant in the loaded zone to the shaft; $R_{\text{вм}}$ is the thermal resistance for the heat flux transferred from the shaft to the lubricant in the unloaded zone; $R_{\text{мп}1}$ is the thermal resistance for heat flux Θ_1 transferred from the lubricant in the loaded zone to the ambient through the bearing liner or vice versa; $R_{\text{мп}2}$ is the thermal resistance for heat flux Θ_2 transferred from the lubricant in the unloaded zone to the ambient through the bearing liner or vice versa; $R_{\text{нн}}$ is the thermal resistance for heat flux $\Theta_{\text{нн}}$, transferred by in the unloaded zone the lubricant pumped through it ($1/c\gamma Q_{\text{нн}}$, where $Q_{\text{нн}}$ is the amount of lubricant pumped through the unloaded zone); $R_{\text{во}}$ is the thermal resistance for the heat flux transferred from the shaft to the ambient.

Thermal resistances and their corresponding thermal conductivities will be defined below. The thermal resistance $R_{\text{во}}$ is assumed to be equal to infinity, because usually the heat transfer along the shaft to the ambient is very small compared to other ways of heat transfer.

In this case, the thermal scheme in Fig. 1 will be as shown in Fig.2.

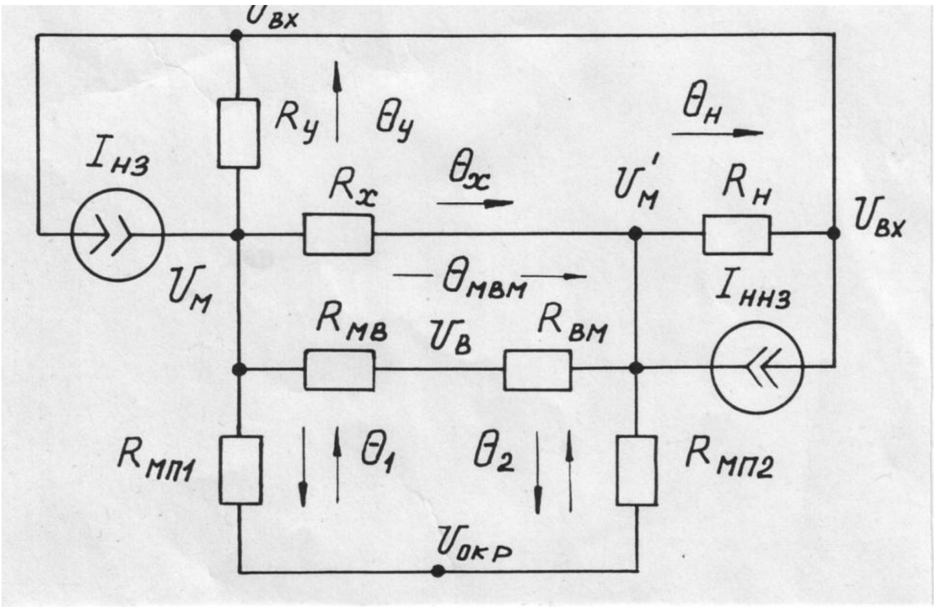

Fig. 2 The thermal scheme of the FFSB

We use the method of nodal potentials to calculate the thermal scheme [7].When solving using this method, one of the nodes is decided to be common and has a zero potential, in our case zero temperature. Assuming it is the node $U_{\text{вх}}$, the temperature of which equals $U_{\text{вх}}= t_{\text{вх}} - t_{\text{вх}}$. Then:

$$U_{\text{м}}= t_{\text{м}}-\ t_{\text{вх}}\,,$$
$$U^1_{\text{м}}= t^1_{\text{м}}-\ t_{\text{вх}}\,,$$
$$U_{\text{окр}}=t_{\text{окр}}-\ t_{\text{вх}}\,. \tag{9}$$

The thermal conductivities used in the node potential method of calculation are determined by thermal resistances and are equal to: $\sigma_{\text{х}}=1/R_x= c\gamma Q_x;$

$\sigma_y=1/R_y= c\gamma Q_y;$ $\sigma_{\text{мп}1}=1/R_{\text{мп}1};$ $\sigma_{\text{мп}2}=1/R_{\text{мп}2};$ $\sigma_{\text{н}}=1/R_{\text{н}}= c\gamma Q_{\text{н}};$ $\sigma_{\text{мв}}=1/R_{\text{мв}};$ **;** $\sigma_{\text{вм}}=1/R_{\text{вм}}$.

The sum of conductivities $\sigma_{\text{мв}}$ is $\sigma_{\text{вм}}$ indicated as *σ:*

$$\sigma\ = \frac{\sigma_{\text{мв}}\sigma_{\text{вм}}}{\sigma_{\text{мв}}+\sigma_{\text{вм}}}, \tag{10}$$

The heat flux sources are equal to: $\sigma_{\text{мв}}$

$$I_{\text{нз}}\ = Pf_{\text{нз}}u, \tag{11}$$

$$I_{\text{ннз}}= Pf_{\text{ннз}}u, \tag{12}$$

where $f_{\text{нз}}$, $f_{\text{нз}}$ are friction coefficients in the loaded and unloaded zones.

The system of equations, compiled using the method of nodal potentials, will be the following:

$$\begin{cases} U_{м}(\sigma_y + \sigma_{мп1} + \sigma_{х} + \sigma) - U_{м}^{1}(\sigma_{х} + \sigma) - U_{окр}\sigma_{мп1} = I_{нз}, \\ - U_{м}(\sigma_{х} + \sigma) + U_{м}^{1}(\sigma_y + \sigma_{мп1} + \sigma_{х} + \sigma) - U_{окр}\sigma_{мп2} = I_{ннз}. \end{cases} \quad (13)$$

Applying the values of conductivity the determinant of system (13) will be:

$$\Delta = c\gamma Q_{н}(c\gamma Q_x + c\gamma Q_y + \sigma + \sigma_{мп1}) + \sigma_{мп2}(c\gamma Q_x + c\gamma Q_y + \sigma + \sigma_{мп1}) + (c\gamma Q_x + \sigma)(c\gamma Q_y + \sigma_{мп1}), \quad (14)$$

Applying that $f = f_{нз} + f_{ннз}$ the determinants of temperatures $U_{м}$, $U_{м}^{1}$ will be:

$$\Delta U_{м} = (c\gamma Q_x + \sigma)(Pfu + U_{окр}\sigma_{мп1} + U_{окр}\sigma_{мп2}) + (Pf_{нз}u + U_{окр}\sigma_{мп1})\, c\gamma Q_{н} + (Pf_{ннз}u + U_{окр}\sigma_{мп1})\, \sigma_{мп2}, \quad (15)$$

$$\Delta U_{м}^{1} = (c\gamma Q_x + \sigma)(Pfu + U_{окр}\sigma_{мп1} + U_{окр}\sigma_{мп2}) + (Pf_{ннз}u + U_{окр}\sigma_{мп1})\, c\gamma Q_y + (Pf_{ннз}u + U_{окр}\sigma_{мп2})\, \sigma_{мп1}. \quad (16)$$

Determine the node temperatures as $U_{м} = \Delta U_{м}/\Delta$, $U_{м}^{1} = \Delta U_{м}^{1}/\Delta$ and considering equations (9), we obtain expressions for the lubricant temperature in the loaded and unloaded zones:

$$t_{м} = U_{м} + t_{вх} = t_{вх} + \frac{1}{\Delta}\{Pfu + (t_{окр} - t_{вх})(\sigma_{мп1} + \sigma_{мп2})](c\gamma Q_x + \sigma) + [Pf_{нз}u + (t_{окр} - t_{вх})\,\sigma_{мп1}]\, c\gamma Q_{н} + [Pf_{ннз}u + (t_{окр} - t_{вх})\,\sigma_{мп1}]\,\sigma_{мп2}, \quad (17)$$

$$t_{м}^{1} = U_{м}^{1} + t_{вх} = t_{вх} + \frac{1}{\Delta}\{[Pfu + (t_{окр} - t_{вх})(\sigma_{мп1} + \sigma_{мп2})](c\gamma Q_x + \sigma) + [Pf_{ннз}u + (t_{окр} - t_{вх})\,\sigma_{мп1}]\, c\gamma Q_y + [Pf_{ннз}u + (t_{окр} - t_{вх})\,\sigma_{мп2}]\,\sigma_{мп1}. \quad (18)$$

The obtained formulas can be used for calculating both lightly loaded and heavily loaded sliding bearings. One can see what affect the ambient temperature has, for example, when $t_{окр} < t_{вх}$, as if often happens, the terms, that have a multiplier $t_{окр}$- $t_{вх}$ are subtracted from the terms Pfu, $Pf_{нз}u$, $Pf_{ннз}u$, characterizing the sources of heat fluxes, which is equivalent to reducing them and, correspondingly, reducing the lubricant temperature in the loaded and unloaded zones. From the formulas obtained, one can see the effect of heat transfer through the bearing line (terms $\sigma_{мп1}$, $\sigma_{мп2}$), heat transfer through the shaft to the lubricant flowing through the unloaded zone from lubricant in the loaded zone (term σ). For the case of heavily loaded FFSBs, when heat transfer to the ambient can be neglected, thermal conductivities $\sigma_{мп1}$ and $\sigma_{мп2}$ equal zero:

$$t_{м} = t_{вх} + \frac{Pfu(c\gamma Q_x + \sigma) + Pf_{нз}u\, c\gamma Q_{н}}{c\gamma Q_{н}(c\gamma Q_x + c\gamma Q_y + \sigma) + c\gamma Q_y(c\gamma Q_x + \sigma)}, \quad (19)$$

$$t_{м}^{1} = t_{вх} + \frac{Pfu(c\gamma Q_x + \sigma) + Pf_{ннз}u\, c\gamma Q_{н}}{c\gamma Q_{н}(c\gamma Q_x + c\gamma Q_y + \sigma) + c\gamma Q_y(c\gamma Q_x + \sigma)}. \quad (20)$$

It has been experimentally proved [9] that the amount of lubricant flowing into the ends of the loaded zone in heavily loaded FFSBs is negligible small. I.e. $Q_y = 0$. Then (19), (20) can be represented in the following form:

$$t_{м} = t_{вх} + \frac{Pfu}{c\gamma Q_{н}} + \frac{Pf_{нз}u}{c\gamma Q_x + \sigma}, \quad (21)$$

$$t_{м}^{1} = t_{вх} + \frac{Pfu}{c\gamma Q_{н}}, \quad (22)$$

Or $$t_{м} = t_{м}^{1} + \frac{Pf_{нз}u}{c\gamma Q_x + \sigma}. \quad (23)$$

As one can see from the formulas (21), (22), (23) the lubricant temperature in the unloaded zone is determined by the amount of lubricant pumped through the unloaded zone, and the lubricant temperature in the loaded zone if all other conditions are equal (load, friction coefficient, sliding speed) is determined by the value of the lubricant temperature in the unloaded zone and highly depends on the value of the thermal conductivity σ, which characterizes the heat transfer developed in the lubricant of the loaded zone to the lubricant pumped in an unloaded zone through a rotating shaft.

When the shaft rotates, the temperature of each of its points on the surface, depending on the lubricant temperature, changes its temperature. For a fixed mode of operation of the FFSB, periodic change of the temperature is harmonic. The process of temperature changes during one period of rotation may be varied. For example, the temperature can vary sinusoidally, continuously increase, then continuously decrease or change impulsively. Thus, one can assume that when the surface temperature of the shaft is lower than the lubricant temperature, it accumulates heat from the lubricant, or when the surface temperature is higher, the shaft transfers heat to the lubricant. This assumption was made in the paper [9]. There are given the heat transfer coefficients of the lubricating layer to the shaft, according to which the amount of heat transferred by the shaft is determined directly from the formula $\Theta = \alpha S(T_1 - T_2)$. However, the paper [9] does not take into account the true ability of the shaft to accumulate and give off heat.

To do this one has to consider the possibility of a rotating shaft to transfer the heat generated in the loaded zone to the lubricant pumped through the unloaded zone and solve the problem of the form of the temperature field in the shaft, and then the problem of the heat flux through the shaft. In addition, one first has to assume that all heat from the lubricant in the loaded zone is transferred to the shaft and from the shaft all heat is transferred to the lubricant in an unloaded lubricant, which is equivalent to the fact that each point of the shaft surface repeats the temperature of each point of the lubricant layer, and then to make a correction taking into account the finiteness of the heat transfer coefficients.

Since the shaft does not have internal sources of thermal energy, to determine the temperature field in it, we can consider the Fourier's heat equation in the following form:

$$\frac{\partial U}{\partial t} = a\nabla^2 U \,, \tag{24}$$

where a is the heat conductivity coefficient, U is the temperature function, t is the time.

This assumes that the temperature function U does not obey any temporal initial condition, i.e. the previous duration is so great that the influence of the initial temperature distribution is no longer displayed.

One of the solutions of the Fourier's equation is the product of two functions, one of which is connected only with the coordinates, and the other with time:

$$U = \psi(x,y,z)\varphi(t), \tag{25}$$

When any point on the surface of the shaft is heated, its initial temperature T_1 is less than its temperature at the time t and less than the lubricant temperature T_0, i.e. $T_1 < T_t < T_0$. When any point on the surface of the shaft is cooled, it is vice versa: $T_1 > T_t > T_0$. Herewith the dimensionless quantity $(T_t - T_0)/ \ (T_1 - T_0)$ in the first and second cases is equal to 1 at the initial moment, and by equilibrium $T_t = T_0$, is equal to 0.

This condition is met by the exponential function

$$\varphi(t) = \mathrm{e}^{-\mathrm{pt}} \,, \tag{26}$$

where p is an AQ. As a result, the solution of the equation of the temperature field takes the following form:

$$U = U_{\mathrm{A}} e^{-y\sqrt{\pi/at_0}} \cos\left(\frac{2\pi}{t_0} t - y\sqrt{\frac{\pi}{\mathrm{at_0}}}\right) \,, \tag{27}$$

Where U_A - is the peak value of the temperature function;

y - is the coordinate in the direction of the radius of the shaft;

t_0 - is the period of oscillation (in this case, the period of the shaft rotation.

We will determine the heat flux through the shaft as:

$$\Theta = \lambda_{\text{в}} S \int \left(\frac{\partial \mathrm{U}}{\partial \mathrm{y}}\right) \mathrm{dt} \,, \tag{28}$$

where $\lambda_{\text{в}}$ - is the heat conductivity coefficient of the material of the shaft;

S - is the shaft surface area.

Integration of the equation (28) in half the period taking into account that the heat conductivity coefficient $a = \lambda_{\text{в}}/\gamma_{\text{в}} c_{\text{в}}$ will give an expression for the heat flux carried by the shaft:

$$\Theta = U_A\, S\sqrt{\frac{2}{\pi}\ \lambda_в \gamma_б c_в t_0}\ , \tag{29}$$

where $\gamma_в -$ is the density of the material of the shaft;
$c_в -$ is the heat capacity of the material of the shaft.

The equation (29) shows that if the temperature of the shaft changes according to the harmonic law, the heat will be accumulated by the shaft from the loaded zone, and half to the lubricant in the unloaded zone for a half of the rotation period of the shaft. The experimental studies [1, 2, 3, 4, 5] with simultaneous measurement of the lubricant layer thickness, hydrodynamic pressure developed in it and temperature along the circumference of the FFSB have shown that the hydrodynamic pressure development zone, i.e. the loaded zone is much smaller. Taking into account the finiteness of the heat transfer coefficients from the lubricant to the shaft in the loaded zone and from the shaft to the lubricant in the unloaded zone, and also the representation of the periodic temperature function [10] using harmonic analysis in the form of a sum of different cosine waves, the determination for each cosine of the corresponding temperature, the summation of these temperatures, and for the total temperature determines the heat flux accumulated by a shaft with diameter d, sliding velocity u, in a loaded FFSB l long in the pressure development zone t:

$$\Theta = \frac{2l}{\pi}\sqrt{u\, d \quad \lambda_в \gamma_в c_в}\left[\sum_{k=1}^{\infty}\frac{\eta_0^1}{k^{3/2}}\left(\sin\frac{k\omega\tau}{2}\right)^2 \cos\varepsilon_0^1\ \right](\,t_м - t_м^1)\ , \tag{30}$$

where η_0^1, $cos\,\varepsilon_0^1$ are coefficients that take into account the final value of the heat transfer coefficient, recommended [11] for fluid flows in flat channels, in which the channel length considerably exceeds its height, the coefficient before $t_м$- $t_м^1$ by similarity with (7) is the conductivity $\sigma_{мв}$, i.e. the conductivity from lubricant to the shaft in the loaded zone.

Conclusions and prospects for further researches

Based on the above-stated information, a methodology for the thermal calculation of heavily loaded FFSBs can be proposed, consisting of preliminary and clarified thermal calculations, which allow to determine the average lubricant temperatures in the loaded and unloaded zones, the required amount of pumped lubricant, the friction coefficient, the minimum thickness of the lubricant layer. An electrothermal similarity for determining the distribution of heat fluxes in the FFSB, which takes into account all heat sources and ways of heat removal from the FFSB is proposed as a basis for thermal calculation.

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