NUMERICAL CALCUALTION OF THE TEMPERATURE FIELD OF A ONE-STEP MAGNETOFLUID SEALING UNIT

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The temperature distribution in a cooled magnetofluid sealing unit is investigated numerically. The dependence of the maximum temperature in the sealing layer of magnetic fluid on the rate of axle rotation is obtained for various conditions of cooling.

At high rates of axle rotation, the temperature of the magnetic fluid in the gap of a magnetofluid sealing unit, rising on account of viscous dissipation, must be held below the limiting permissible level by means of a special cooling system. Calculations of the thermal conditions in the sealing layer of fluid show that, even for an idealized model with a fixed boundary temperature of the layer T = 10°C, heating to 100°C is observed even at $V_0 \approx 80$ m/sec [1]. With further increase in rate of axle rotation, it is obviously necessary to maintain a negative temperature at the boundaries of the fluid layer. This is usually achieved by passing coolant through annular channels in the polar endpieces at some distance from the fluid boundaries or else through an axial channel in the sealed axle [2]. Hence, heat transfer occurs through a layer of the material from which the axle and the polar endpiece are made.

To estimate the limiting possibilities of such a cooling system, the thermal conditions of a magnetofluid sealing unit cooled through the axle and the polar endpiece are numerically investigated in the present work. The temperature distribution in the central cross section of the one-step sealing unit is described by a one-dimensional heat-conduction equation in the coordinate system (r, φ, z) with discontinuous coefficients

$$\rho c \frac{\partial T}{\partial t} = \frac{1}{r} \frac{\partial}{\partial r}\left(\lambda r \frac{\partial T}{\partial r}\right) + q. \qquad (1)$$

Here change in r within the limits $R_0 < r < R_1$ corresponds to the axle cross section: $R_1 < r < R_2$ to the fluid and $R_2 < r < R_3$ to the polar endpiece. For the given ranges of r, the following values of the physical parameters were chosen

$$\rho c = \begin{cases} 3.75\cdot 10^6, & R_0 \leqslant r < R_1,\ R_2 < r < R_3, \\ 1.43\cdot 10^6 [1 - 8.8\cdot 10^{-4}(T-30)][1 + 1.93\cdot 10^{-3}(T-30)], & R_1 < r < R_2, \end{cases}$$

$$\lambda = \begin{cases} 48, & R_0 \leqslant r < R_1,\ R_2 < r < R_3, \\ 0.153\,[1 - 11.3\cdot 10^{-4}(T-30)], & R_1 < r < R_2. \end{cases}$$

The temperature dependences of the physical properties of the magnetic fluid are taken, by convention, to be identical to the dependence of the properties of the base — the organosilicon fluid PMS-50. Organosilicon fluid with low saturated-vapor tension may serve as the basis for magnetic fluids used in vacuum magnetofluid sealing units [3]. Steel St. 3 is chosen as the material of the axle and endpiece. The last term in Eq. (1) takes the form

$$q = \begin{cases} \alpha_1 (T - T_0), & R_0 \leqslant r < R_1, \\ \alpha_2 (T - T_0) + q_1, & R_1 < r < R_2, \\ \alpha_3 (T - T_0), & R_2 < r < R_3. \end{cases}$$

Here

$$q_1 = \eta \left[\frac{2R_1 R_2^2 V_0}{r^2 (R_2^2 - R_1^2)}\right]^2;\ \eta = 8.7\cdot 10^{-5} \exp\left(\frac{1864.2}{273 + T}\right).$$

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The heat source q_1 characterizes the viscous dissipation of the mechanical energy; the coefficients α_1, α_2, α_3 take account of the heat discharge in the direction of the z axis and are of the order of $2\lambda/\Delta z\delta$, where $\Delta z = 10^{-3}$-10^{-2} m is the linear scale at which the temperature difference $(T - T_0)$ is observed. The minimum temperature in the sealing system T_0 is taken to be 15°C.

The following boundary conditions are considered: when $r = R_0$ and $r = R_3$, the temperature is chosen equal to the temperature of the coolant passing through channels in the polar endpiece and axle

$$T|_{r=R_0} = T|_{r=R_3} = T_1, \tag{2}$$

and, if the axle has no cooling channel ($R_0 = 0$), then the symmetry condition is imposed at the axis

$$\left.\frac{\partial T}{\partial r}\right|_{r=0} = 0. \tag{3}$$

The grid method is used to solve the problem in Eqs. (1)-(3). The given geometry of the sealing unit is distinguished in that the gap $\delta = R_2 - R_1$ is ~10^{-4}-$5\cdot10^{-4}$ m, while $(R_3 - R_2)$ and $(R_1 - R_0)$ are two orders of magnitude larger. Therefore, in the gap and close to its boundaries, considerable temperature gradients are observed. Increase in accuracy of the calculations is associated with the selection of a sufficiently small-scale grid. To reduce the time required for the calculation, a nonuniform grid is chosen, taking the features of the problem into account: inside the gap it is uniform, consisting of 10-20 elements but, with increasing distance from the gap boundaries, the grid step increases in a geometric progression up to a specified maximum, after which it remains constant. As a result, the grid contains ~50-100 elements, while retaining the same error as a uniform grid of ~10^3-$2\cdot10^3$ elements.

Equation (1) is approximated by an implicit, conservative, continuous-computation difference scheme [4]. The solution is obtained by the method of three-point fitting, with three iterations in each time layer to correct the nonlinear dependence of the physical characteristics on the temperature [1].

Analysis of the results obtained shows that, holding the wall temperature of the cooling channel in the polar endpiece equal to T_1 = 15°C, the maximum fluid temperature cannot be limited to 100°C even when V_0 = 30-35 m/sec if there is no heat discharge in the direction of the z axis along the axle and the body of the endpiece (Fig. 1, curves 1 and 2). Earlier [1], it was shown that the influence of the term $\alpha_2(T - T_0)$ on the discharge of the heat liberated in the gap is insignificant. To evaluate the contribution of the terms $\alpha_1(T - T_0)$, $\alpha_3(T - T_0)$, calculations are performed with $\alpha_1 = \alpha_3 = 10^5$ ($\Delta z = 10^{-3}$), which corresponds to the limiting heat transfer under the conditions $\alpha_1 = 0$, $\alpha_3 = 10^5$ and $\alpha_1 = 10^5$, $\alpha_3 = 0$.

It is found that the term $\alpha_3(T - T_0)$ characterizing the cooling of the body of the endpiece has practically no influence on the maximum temperature. The term $\alpha_1(T - T_0)$, the heat transfer along the axle, has the greatest effect. As a result, a decrease in maximum temperature by approximately 30-50°C is observed, which allows the peripheral velocity of the axle to be increased to $V_0 \approx 55$ m/sec, taking the limiting temperature to be T = 100°C (Fig. 1, curve 4).

Obviously this value is close to the limiting peripheral velocity of the axle, above which temperature stabilization of the magnetic fluid at a level of 100°C is impossible if a coolant at a temperature of 15°C (tap water) is used. In fact, calculation of the construction of a cooling system with heat transfer by passing coolant along an axial channel in the axle shows that it allows the same peripheral velocity to be reached (Fig. 1, curves 3 and 5). For higher peripheral velocities, a cryogenic coolant must be used. Thus, for example, if T_1 = −100°C is maintained by passing coolant through the axial channel in the axle and a channel in the polar endpiece, the fluid in the gap is heated to 100°C even at $V_0 \approx 70$ m/sec (Fig. 1, curve 6). It is clear that, in the realization of such cooling, a system of automatic control of the coolant supply must be created, to ensure operation of the sealing unit at different V_0.

In practice, it is probably possible to reach higher velocities if the possibility of convective motion of the fluid with slight radial and axial displacements of the sealed axle is taken into account. On the other hand, the maximum temperature chosen as permissible in

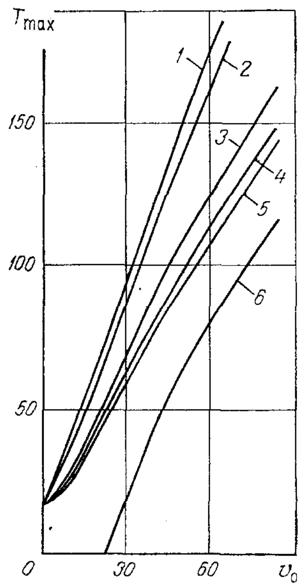

Fig. 1. Dependence of the maximum temperature in the sealing layer of fluid on the linear velocity of the axle surface: 1) $R_0 = 0$; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 7.02 \cdot 10^{-2}$ m; $\alpha_{1,3} = 0$; 2) $R_0 = 0$; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 4.52 \cdot 10^{-2}$ m; $\alpha_{1,3} = 0$; 3) $R_0 = 0.5 \cdot 10^{-2}$ m; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 5.02 \cdot 10^{-2}$ m; $\alpha_{1,3} = 0$; 4) $R_0 = 0$; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 7.02 \cdot 10^{-2}$ m; $\alpha_{1,3} = 10^5$ W/m$^3 \cdot$°K; 5) $R_0 = 0.5 \cdot 10^{-2}$ m; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 5.02 \cdot 10^{-2}$ m; $\alpha_{1,3} = 10^5$ W/m$^3 \cdot$°K; 6) $R_0 = 0.5 \cdot 10^{-2}$ m; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 5.02 \cdot 10^{-2}$ m; $\alpha_{1,3} = 10^5$ W/m$^3 \cdot$°K; $T_0 = -50$°C; $T_1 = -100$°C, T_{max}, °C; V_0, m/sec.

the operating conditions may be much less than 100°C, especially in vacuum magnetofluid sealing units [5].

To obtain recommendations regarding the choice of an optimal variant of the cooling-system construction, calculations are performed for various positions of the cooling channels with respect to the gap. It is found that change in the distance R_3-R_2 in the range $5 \cdot 10^{-3}$ $3 \cdot 10^{-2}$ leads to only 5°C increase in T_{max}. Hence, in this range the position of the cooling channel in the polar endpiece has little influence on the temperature conditions of the sealing layer of fluid. The dependence on the distance R_1-R_0 is found to be very significant. Increase in R_1-R_0 from $5 \cdot 10^{-3}$ to $3 \cdot 10^{-2}$ leads to ≈30°C increase in T_{max}. The difference is explained by the cylindrical geometry. Note that the construction of the cooling system with heat transfer along the axle (e.g., passage of coolant along an axial channel) may prove uniquely efficient for high-velocity magnetofluid sealing units.

The characteristic temperature distribution in the cross section of a one-step sealing unity in steady thermal conditions is shown in Figs. 2 and 3. Calculations show that the time required to heat the fluid to a near-maximum temperature is 1-3 sec after startup of the machine, and the complete stabilization process takes 1-5 min.

Such dynamics of the process indicate the need to switch on the cooling system 1-5 min before the startup of the machine to prevent effervescence of the magnetic fluid in the course of its approach to working conditions.

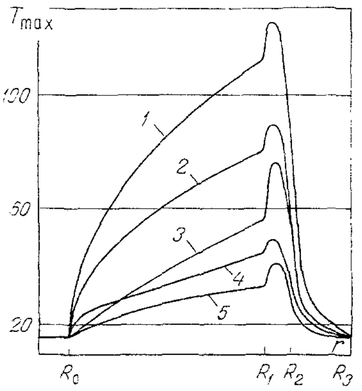

Fig. 2

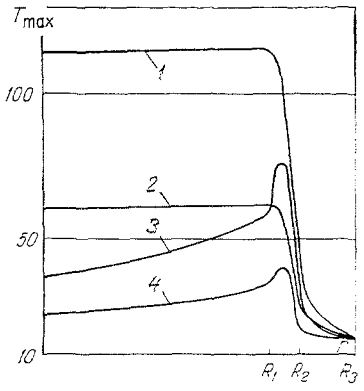

Fig. 3

Fig. 2. Temperature distribution in the cross section of a sealing unit cooled through the polar endpiece and the axle: $R_0 = 0.5 \cdot 10^{-2}$ m; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 5 \cdot 10^{-2}$ m; 1) V_0 = 60 m/sec; $\alpha_{1,3} = 0$; 2) V_0 = 40 m/sec; $\alpha_{1,3} = 0$; 3) V_0 = 40 m/sec; $\alpha_{1,3} = 10^5$ W/m$^3 \cdot$°K; 4) V_0 = 20 m/sec; $\alpha_{1,3} = 0$; 5) V_0 = 20 m/sec; $\alpha_{1,3} = 10^5$ W/m$^3 \cdot$°K. T_{max}, °C; r, m.

Fig. 3. Temperature distribution in the cross section of a sealing unit cooled through the polar endpiece: $R_0 = 0$; $R_1 = 4 \cdot 10^{-2}$ m; $R_2 = 4.02 \cdot 10^{-2}$ m; $R_3 = 5 \cdot 10^{-2}$ m; 1) V_0 = 40 m/sec; $\alpha_{1,3} = 0$; 2) V_0 = 20 m/sec; $\alpha_{1,3} = 0$; 3) V_0 = 40 m/sec; $\alpha_{1,3} = 10^5$ W/m$^3 \cdot$°K; 4) V_0 = 20 m/sec; $\alpha_{1,3} = 10^5$ W/m$^3 \cdot$°K.

NOTATION

V_0, peripheral velocity of axle, m/sec; ρ, density, kg/m^3; c, specific heat, J/kg$\cdot$°K; λ, thermal conductivity, W/m$\cdot$°K; η, viscosity of magnetic fluid, Pa$\cdot$sec; R_0, radius of cooling channel running along axle axis, m; R_1, radius of the sealed axle, m; R_2, internal radius of polar-endpiece tooth, m; R_3, wall radius of cooling channel in polar endpiece, m; T_{max}, maximum temperature; $\delta = R_2 - R_1$, m.

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