and then begins to decrease. It is interesting to note that the drop in power with increasing value of π_p proceeds far more slowly for the two-dimensional model than for the complex shear model.

It is evident from an analysis of the foregoing results that, first, wall slip exerts a powerful influence on the performance characteristics of the mechanism and, second, the effect of the side walls in conjunction with fluid slip along the walls is very pronounced and must be taken into consideration in the calculations.

NOTATION

m, n, η_0, b, rheological constants; H, W, depth and width of screw channel; V_0, circumferential velocity of screw ridges (velocity of upper plate in the opposite direction); φ, angle of elevation of screw line; ρ, fluid density; x, y, z, x_i, x_j, Cartesian coordinates; L, screw length; l, length of screw channel; i, j = 1, 2, 3; w_x, x_z, dimensionless fluid velocities; v_x, v_z, v_i, v_j, true fluid particle velocities; v_W, wall slip velocity; ω, vorticity; ψ, stream function; I_2, second (quadratic) invariant of strain-rate tensor; η, effective fluid viscosity; T_W, tangential stress intensity at wall; h, step of computing grid; Q, volumetric flow rate; N, power; A_z, pressure gradient along screw channel; π_Q, π_p, π_N, dimensionless values of flow rate, longitudinal pressure gradient, and power; $\xi = y/H$, dimensionless coordinate; p, pressure; p_0, p_1, fluid pressures at channel entrance and exit; $\bar{\tau} = \eta_0(V_0/H)^n$, characteristic tangential stress; τ_{ij}, components of stress tensor; τ_S, shear stress at which slip is initiated; τ_W, shear stress at wall; β_W, slip ratio.

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THERMAL CONDITIONS OF MAGNETOFLUID SEALS

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Heat production in the working region of magnetofluid seals is theoretically and experimentally evaluated.

One of the most promising sealing techniques for application to rapidly rotating systems is the magnetofluid seal (MFS). The MFS, whose working element is a ferromagnetic fluid (FMF), held in a prescribed position by a magnetic field, has several advantages over the common contact and noncontact seals: MFS operate is a wide range of shaft rotation speeds, have a low friction torque and a long operating life [1].

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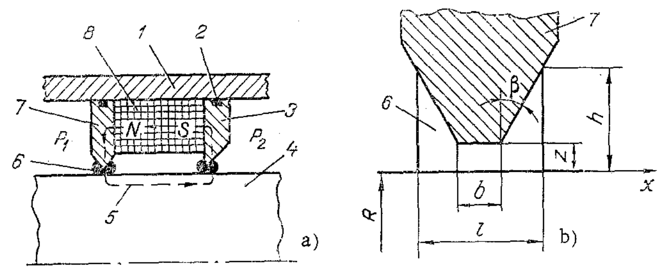

Fig. 1. Diagrams of magnetofluid seal (a) and the working gap (b) for heat-loss calculations: 1) housing; 2) sealing ring; 3, 7) poles of asymmetric and symmetric profile; 4) shaft; 5) direction of magnetic flux; 6) ferromagnetic fluid in sealed working gap; 8) magnet.

The MFS investigations known at present have been devoted mainly to determination of the maximum sustainable pressure differential and optimization of the seal design parameters purely from this viewpoint [1-3]. An increase in the linear velocity of the shaft surface above 10 m/sec makes another problem important – determination of the thermal conditions of MFS [4-6]. This is due to the fact that an increase in the shear rate in the FMF filling the sealed gap leads to an increase in viscous energy dissipation, i.e., to heating of the fluid.

To investigate the thermal conditions of MFS we need to determine, firstly, the absolute heat production in the volume of the ferrofluid and, secondly, the pathways of heat removal from the working region. To analyze these questions we used an elementary seal unit (Fig. 1a), which is the basic component of the diverse existing types of MFS. This unit consists of a nonmagnetic cylindrical housing containing a rotating shaft mounted in bearings and an annular magnet with annular pole pieces. The inside surface of the poles and the shaft surface form the working sealed gap, which is filled with the ferromagnetic fluid. The pressure differential that can be sustained by MFS depends on several factors, mainly the shaft rotation speed, the saturation magnetization of the ferromagnetic fluid, the width of the working gap, the magnetic field strength in the gap, the pole configuration, etc. In the general case the pressure differential that can be sustained by MFS decreases with increase in shaft rotation speed and width of working gap, and with reduction in saturation magnetization of the ferromagnetic fluid and field strength in the gap.

We estimate the heat loss in the working gap formed by a pole with a symmetric cross section (Fig. 1b). For simplicity we assume that the free boundaries of the ferromagnetic fluid filling the working gap between the pole and shaft are vertical. It is known that the heat loss per unit volume due to viscous dissipation is given by the expression [7]

$$\dot{E}_V = \eta \left(\frac{\partial v}{\partial r} \right)^2 . \tag{1}$$

We assume that the fluid flow is laminar, the velocity has only an azimuthal component, and the velocity drop between the rotating shaft and the stationary pole is considerable. In this case planes x = const, perpendicular to the shaft axis, can be regarded as planes of constant shear rate $\tau = \partial v/\partial r$, and in view of the small height h of the liquid layer in comparison with the shaft radius R we can put for each plane $\tau \approx \Delta v/\Delta r = \omega R/r(x)$, where ω is the shaft rotation speed; r(x) is the distance from the shaft surface to the pole in the particular x plane. Then the total heat release in the volume of the liquid is

$$N = \int_V \dot{E}_V \, dV, \tag{2}$$

where V is the total volume of ferrofluid between the pole and shaft. When the above assumptions are made, this integral has the form

$$N = 2\pi R^3 \omega^2 \eta \left[\frac{b}{z} + \operatorname{tg}\beta \ln \frac{h}{z} \right] . \tag{3}$$

Since there is a simple relation between h and V when h, z ≪ R, we retain as arguments in (3) only the design parameters b, z, and β, and also the viscosity of the fluid η and its volume V:

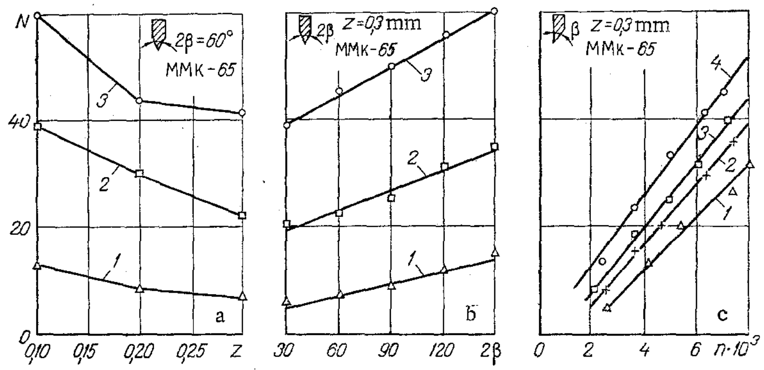

Fig. 2. Heat loss N (W) as function of: a) shaft rotation speed n (rpm), gap width z (mm) (1 – 2000; 2 – 5000, 3 – 7000 rpm); b) pole-tip angle 2β and shaft rotation speed n (1 – 2000, 2 – 5000, 3 – 8000 rpm); c) shaft rotation speed n and pole-tip angle β: 1) 15°; 2) 30°; 3) 45°; 4) 60°.

$$N = 2\pi R^3\omega^2\eta\left[\frac{b}{z} + \operatorname{tg}\beta\ln\left(1 + \frac{V - 2\pi Rbz}{2\pi Rz^2\operatorname{tg}\beta}\right)\right]. \quad (4)$$

The obtained expression indicates the nature of the relation between the heat loss and the design parameters: the loss increases with increase in pole-tip width b, with reduction of the gap between the pole and shaft z, and with increase in the pole-tip angle β; when β changes from 0 to 90° the second term in the square brackets in (4) changes from 0 to $(V-2\pi Rbz)(2\pi Rz^2)^{-1}$, increasing almost linearly with change in β from 10 to 70°. In addition, the heat production in the gap increases with increase in viscosity η of the fluid, the shaft radius R, and its angular velocity ω.

For instance, for an MFS with parameters R = 20 mm, b = 0.5 mm, z = 0.3 mm, n = 8000 rpm, filled with ferrofluid of volume 0.1 cm^3 with viscosity 4 P, expression (4) gives a heat loss of 40 W.

For the experimental investigations of the thermal conditions in MFS we used the same elementary seal unit (Fig. 1). The friction torque was determined with the aid of a calibrated spring connecting the shaft of an electric motor to the MFS shaft, from the angle of rotation of which we determined the friction torque M in the MFS and the heat loss $N = M\omega$. To measure the angle of twist of the spring we used a device consisting of inductance coils attached to the MFS housing and the electric motor, whose relative position could be altered, and permanent magnets attached to the shafts of the motor and MFS. When the magnets moved past the inductance coils they induced an emf in the coils, which was measured by an oscillograph and a frequency meter. From the pulse frequency we obtained the shaft rotation speed, and from the phase difference of the current pulses in the coils we determined the angle of twist of the calibrated spring, the friction torque M, and the heat loss N. During the heat-loss measurements we also made thermocouple measurements of the temperature near the seal working gap, for which a copper–constantan thermocouple of diameter 0.18 mm was embedded at a distance of 0.4 mm from the beveled edge of the pole. For measurement of the heat flux through the pole a second thermocouple was embedded at a distance of 1.5 mm from the first. In the experiments we used symmetric and asymmetric poles and two ferrofluids prepared from mineral oil and magnetite with a saturation magnetization of 65 and 37 kA/m, which we will henceforth designate MMk-65 and MMk-37, respectively.

We conducted a series of experiments to determine the heat production in the volume of the FMF under the pole and its variation in relation to the geometric parameters (the pole-tip angle β and the gap width z) and the shaft rotation speed n. The range of pole-tip angles was β = 15–75° for symmetric and asymmetric poles, the gap width z was 0.1, 0.2, and 0.3 mm, and the shaft rotation speed n varied from 2000 to 8000 rpm. The MFS shaft diameter was 40 mm, the pole-tip width b = 0.5 mm, and the working gap was filled with 0.16–0.18 cm^3 of ferrofluid.

Figure 2 shows typical plots of heat production in the working gap against shaft rotation speed n, pole-tip angle β, and gap width z. As was to be expected from the obtained relation (4), the heat production increased with reduction in gap width z and increase in the pole-tip angle β. When we come to compare the relation between the heat loss N and the shaft rotation speed n, shown in Fig. 2c, with expression (4) they appear at first sight to contradict one another: expression (4) indicates a square-law variation, whereas the experiment shows a linear relation. In the interpretation of the experimental data, however, it should be borne in mind that with increase in n the temperature in the gap also increases, which leads to a reduction of the viscosity η of the ferromagnetic fluid, i.e., the degree of increase of N decreases. Temperature measurements made for a symmetric pole with gap width z = 0.1, 0.2, and 0.3 mm showed that when the device was in operation the temperature of the magnetic fluid in the gap increased and within an hour reached a value which subsequently remained practically constant. This steady temperature depended on the gap width and on the shaft rotation speed; the obtained data can be approximated by the relation (for the ferromagnetic fluid MMk-65)

$$T = 20^{\circ} + (5{,}6 - 10z)\, n/1000, \tag{5}$$

where z is in mm, and n in rpm. We can also approximate the temperature dependence of the viscosity of the fluids MMk-65 and MMk-37 at temperatures 25-55°C by the relation

$$\eta = \frac{F_0}{T - 11^{\circ}}, \tag{6}$$

where F_0 is an empirical constant that depends on the type of fluid. In this case a direct comparison of experiment and theory is hindered by the uncertainty of the value of F_0. Viscosity measurements by the Höppler method gave a value η = 60 P (at T = 20°C) in a field H = 100 kA/m and a shear rate $\tau = 1\ \text{sec}^{-1}$. When the shear rate was increased, however, the viscosity of the ferromagnetic fluid decreased sharply, which is probably due to breakup of the structures (chains, clusters) formed in the fluid in the magnetic field and reduction of the rotational viscosity, since shear flow reduces the orienting effect of the magnetic field on the ferromagnetic particles in the fluid. The viscosity decreased to a limit given by the Einstein formula $\eta = \eta_0 \cdot (1 + 5\alpha/2)$ (where η_0 is the viscosity of the carrier fluid; α is the volume concentration of the ferrophase), and for the fluids MMk-65 and MMk-37 had values of 60 cP and 50 cP, respectively. Thus, the viscosity of the ferrofluid changes in a wide range with increase in shear rate and, hence, the value of F_0 can be determined only by direct measurement. Expressions (4) and also (5) and (6), however, indicate the kind of empirical relation that approximately predicts the experimental data:

$$N = \frac{A\,(B + 2\beta/180)\,(n/1000)^2}{z\,[C + (D - Ez)\, n/1000]}, \tag{7}$$

where A, B, C, D, and E are empirical constants. Here D and E are found from the plots of temperature against the gap width z and shaft rotation speed n; A, B, and C are determined from the relations $N(\beta)$, N(n), and N(z). The experimental data for MMk-65, treated as described, produced the following values of the empirical constants: a) symmetric pole – A = 5.8, B = 0.36, C= 1.2; b) asymmetric pole – A = 3.7, B = 0.75, C = 0.5. In both cases D = 5.6, E = 10.

It is of interest to determine the pathways of heat removal from the heat production zone. The ferrofluid is in contact with the shaft, pole, and also with the sealed and external medium, in our case air. Since direct heat removal to the atmosphere is slight, the main heat removal is effected through the shaft and pole. The thermocouple measurements showed that the two thermocouples on the pole at a distance of 1.5 mm from one another indicated a temperature difference ΔT = 9°C with N = 56 W at a shaft rotation speed of 6000 rpm. Such a difference corresponds to a heat flux $q = \lambda \Delta T/\Delta x = 280\ \text{kW/m}^2$. The area of contact of the fluid with the pole can be taken as $2\pi R l$, where l is the width of the strip of contact with the pole, equal to 1.5 mm. The heat removal is then N = 53 W. This indicates that the main heat flux from the fluid in the experimental conditions passes through the pole to the housing and from the latter to the surrounding air.

NOTATION

$\dot{E}_V$, specific heat loss due to viscous dissipation; η, viscosity; v, velocity; τ, shear rate; ω, angular speed; n, shaft rotation speed; r, radial coordinate; R, shaft radius, mm; h, l, height and width of FMF in gap, mm; z, width of gap between pole and shaft, mm; β, pole-tip angle; b, pole-tip width, mm; V, volume of FMF in seal; N, heat loss; M, friction torque; A, B, C, D, E, empirical constants; T, temperature, °C; λ, thermal conductivity; q, heat flux.

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POSITIVE COLUMN OF AN ELECTRIC ARC WITH A GIVEN POWER DENSITY DISTRIBUTION OF INTERNAL HEAT SOURCES

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An analytical solution is obtained for the nonlinear boundary-value problem describing heating of gases in the positive column of an arc with current and gas consumption varying along a channel for a given power density distribution of the internal heat sources.

To assure optimal reaction conditions in plasmachemical reactors it is often necessary to distribute the power density of the internal heat sources in a definite manner along the channel length. An analogous situation holds in processes for treating different powders and materials in an arc plasma. The most natural way to solve this problem is to change the current intensity, the size of the arc chamber, and the gas consumption along the length of the positive column. Such plasmatrons can be called electric arc heaters with distributed parameters, which are of interest both from the viewpoint of raising the plasmatron resources, and from the optimal distribution of heat fluxes [1, 2].

The power density distribution of the heat sources can be given, in many cases of practical importance, by the conditions of the technological process. However, the current distribution would be unknown in advance; hence, there is a necessity to solve the problem for a given power density distribution. Such a problem is also urgent from the viewpoint of designing plasmachemical reactors and a number of other electric arc devices.

The system of equations

$$\frac{h_s \rho u}{l}\frac{\partial S}{\partial z} + \frac{h_s \rho v}{R}\frac{\partial S}{\partial r} = \frac{1}{R^2}\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial S}{\partial r}\right) + \sigma_s E^2 S - \varepsilon_s S, \tag{1}$$

$$\frac{1}{l}\frac{\partial}{\partial z}(\rho u) + \frac{1}{R}\frac{1}{r}\frac{\partial}{\partial r}(r\rho v) = 0, \tag{2}$$

$$I(z) = 2\pi R^2 \sigma_s E(z)\int_0^{\zeta} S r dr, \tag{3}$$

describing the heating of gases in a positive arc column with gas consumption and current distributed along the channel length [3, 4], is solved under the boundary conditions

$$S(r, 0) = \varphi(r/\zeta), \quad S_r(0, z) = 0, \quad S(\zeta, z) = 0, \quad v(0, z) = 0. \tag{4}$$

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