automation system. It is shown as a result of the investigation that the evaporator temperature was minimal during passage of the trailing edge of the regulated heat pulse, i.e., upon disconnection of the heater Q_1.

NOTATION

Q, heat flux; T, temperature; s, area; λ, coefficient of heat conduction; α, heat transfer coefficient; δ, thickness; r, radius; R', thermal resistivity; σ, temperature checking factor; P, pressure; θ, heat pipe slope; τ, time. Subscripts: hp, heat pipe; v, vapor; e, evaporator; c, condenser; s, heat sink; wa, wall; w, wick; df, destabilizing factor; g, gas.

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HEAT TRANSFER IN THE EVAPORATOR SECTION OF A ROTATING HEAT PIPE AT LOW ROTATIONAL SPEEDS

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The heat transfer in the evaporator section of a rotating heat pipe, with and without a wick, is investigated experimentally at low rotational speeds and low heat-flux densities.

Rotating heat pipes are used at the present time for the cooling of electrical machines subjected to heavy rotational-speed control demands (high-torque dc motors, frequency-regulated asynchronous motors, etc.). At high speeds of rotation, the condensate is uniformly distributed by centrifugal forces around the periphery of the heat pipe, and steady-state cooling prevails [1-4]. Periodically, however, the rotational speed of regulated motors approaches zero, whereupon the inertial overloads prove inadequate for uniform distribution of the condensate around the periphery. In this case, the condensate collects in the lower part of the pipe as a pool, and the wall-cooling regime becomes nonsteady. Accordingly, the following are the most timely problems in connection with the development of heat pipes for regulated electrical machines: determination of the rotational speed at which transition takes place from steady to nonsteady cooling of the pipe walls; assessment of the nature of the heat transfer in this transition regime and of the practicality of installing a wick in the pipe to improve the heat transfer at low rotational speeds.

We have investigated these problems experimentally on a special calorimetric arrangement with radiative heat input to the heat pipe and convective heat withdrawal from its condenser section by placement of the pipe in a flow-through water heat exchanger. The construction of the heat pipe and experimental apparatus was designed to create heat-transfer conditions such as occur in heat pipes used in regulated electrical machines. The vapor temperature in the pipe could be varied between 60 and 120°C, the heat-flux density in the evaporator section from $0.4 \cdot 10^4$ to $2 \cdot 10^4$ W/m², and the angular speed from 0.05 to 6 rad/sec. The evaporator section of the pipe had an inside diameter of 100 mm, length of 150 mm, and wall thickness of 10 mm; the condenser section had a diameter of 30 mm and length of 100 mm. The readings of

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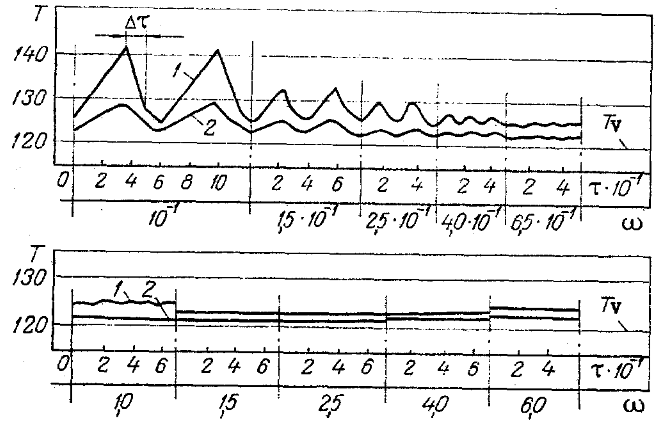

Fig. 1. Variation of local wall temperature of heat pipe T, °C, with rotational speed ω, rad/sec, and time $\tau\cdot 10^{-1}$, sec. 1) $q = 1.8\cdot 10^4$ W/m²; 2) $0.4\cdot 10^4$ W/m².

thermocouples placed in the vapor space and embedded in the pipe wall at a distance of 3 mm from its inner surface were collected by means of an RAT-2 slip ring and recorded on a KSP-4 multiple-contact recording potentiometer. A volume of 75 cm³ of water or acetone was placed in the pipe.

Figure 1 gives the time variation of the local wall temperature of the pipe containing water for various speeds of rotation. It is seen in the figure that at very low speeds, upon emergence of the investigated point from the pool, the linear rise of the local wall temperature is superseded by an abrupt drop, which gradually slows down. At the start of a new revolution, the wall temperature once again begins to grow, and the cycle repeats itself. The linear growth of the temperature in the first period indicates that heat transfer from the wall to the vapor is negligible in this period. The abrupt drop of the local wall temperature in the second period corresponds to submersion of the investigated section into the pool. The average heat-transfer coefficient for this period can be approximately determined from the expression

$$\bar{\alpha} = \frac{q + \bar{q}}{\Delta T},$$

in which

$$\bar{q} = \delta_W \rho c \, \frac{T_1 - T_2}{\Delta\tau} \text{ and } \Delta T = \frac{T_1 + T_2}{2} - T_V.$$

Here $\bar{q}$ accounts for the heat flux associated with removal by the wall of the heat accumulated in the first period.

Calculations according to this expression for $q = 1.8\cdot 10^4$ W/m² show that the heat transfer from the surface of the pool in the period of rapid descent of the local wall temperature amounts to $\bar{\alpha} = 5\cdot 10^3$ to $6\cdot 10^3$ W/m²·deg. At the instant of entry of the overheated wall into the pool, this coefficient is obviously 1.5 to 2 times the average value. The high value of α indicates that the heat transfer in the pool (or at least from part of its surface) takes place in the nucleate boiling regime. The amplitude of the wall-temperature fluctuations for $q = 1.8\cdot 10^4$ W/m² and $\omega = 0.05$ rad/sec attains 25°C and rapidly diminishes as the rotational speed is increased. For $\omega > 1$ rad/sec, the temperature stabilizes; the fluctuation amplitude for $q = 0.4\cdot 10^4$ W/m² (curve 2) is smaller, and stabilization sets in at a lower speed, than for $q = 1.8\cdot 10^4$ W/m². In both cases, the average temperature difference between the wall and the vapor reaches a minimum soon after stabilization of the wall temperature, indicating concomitant variations of the average value of the heat-transfer coefficient α over the perimeter of the pipe (Fig. 2).

It is evident from Fig. 2 that in the heat pipe containing water (curves 1-3) the coefficient α depends rather strongly on the heat-flux density at low rotational speeds, as is typical of nucleate boiling. Since the pool takes up roughly one fourth of the pipe surface, the actual value of the heat-transfer coefficient in the pool is 3 to 3.6 times the perimeter-average heat-transfer coefficient and amounts to $5\cdot 10^3$ to $6\cdot 10^3$ W/m²·deg. This result

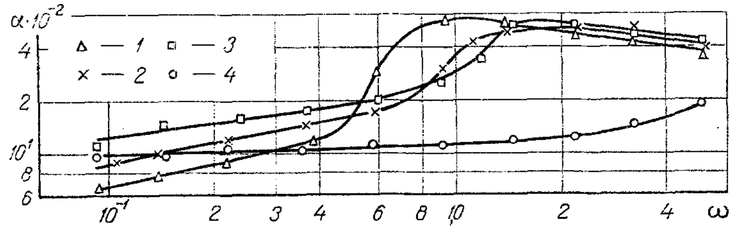

Fig. 2. Perimeter-average heat-transfer coefficient $\alpha \cdot 10^{-2}$, $W/m^2 \cdot deg$, versus rotational speed ω, rad/sec, of heat pipe. Heat-transfer medium: 1-3) water, $T_V = 120°C$: 1) $q = 0.4 \cdot 10^4$ W/m^2; 2) $1.0 \cdot 10^4$; 3) $1.8 \cdot 10^4$; 4) acetone, $T_V = 100°C$, $q = 1.4 \cdot 10^4$ W/m^2.

confirms the fact that the heat transfer in the pool during this period takes place mainly in the nucleate boiling regime.

At low rotational speeds, a thin film forms on the inner surface of the pipe due to the forces of surface tension (wetting), evaporating rapidly upon emergence from the pool [5]. With an increase in the speed of rotation, the thickness of this film increases, so that the surface wetted by the condensate and removing heat increases, and the perimeter-average heat-transfer coefficient increases accordingly. For $\omega = 0.8$ to 1.5 rad/sec, the coefficient α passes through a maximum; at the maximum point, the heat transfer attains $5.5 \cdot 10^3$ to $6.0 \cdot 10^3$ $W/m^2 \cdot deg$ and is practically independent of the heat-flux density. The weak dependence of α on q is also typical of the section situated to the right of the maximum point, where the heat transfer decreases slowly with increasing ω. In these periods, obviously, heat transfer takes place in the regime of evaporation from the surface, typified by a weak dependence $\alpha = f(q)$. The heat-transfer maximum occurs at a rotational speed ($\omega_m = 0.8$ to 1.5 rad/sec) such that the liquid film emerging from one rim of the pool becomes distributed during rotation to its other rim and the average thickness of the film is a minimum, while the wall cooling regime becomes steady. With a further increase in the rotational speed, the average thickness of the film increases, causing the average heat transfer to decrease, which it continues to do so long as all the condensate is distributed over the wall in a uniform annular layer by the centrifugal forces [6]. Characteristically, with a decrease in the heat-flux density, the rotational speed ω_m at which maximum heat transfer sets in also decreases.

The influence of the principal factors on the value of ω_m can be estimated on the basis of the following considerations (Fig. 3). The component of the force of gravity under whose action the liquid flows along the wall is a maximum and directed along the tangent at point A on the inner surface of the pipe wall. The thickness of the layer at this point can be determined according to the Landau–Levich solution for the thickness of the liquid film left on the surface of a plate withdrawn vertically from a pool of liquid [7]. Under conditions of low withdrawal rates ($v \ll \sigma/\eta$) and a sufficient distance of the investigated point from the surface of the liquid pool, the Landau–Levich equation can be written in the form

$$\delta_A = 0.93 \frac{(\omega R \eta)^{2/3}}{\sigma^{1/6} \sqrt{\rho_1 g}} . \qquad (1)$$

During the time in which the investigated zone passes from point A through point B to point C at reentry into the pool, the liquid film completely evaporates at the speed ω_m. All

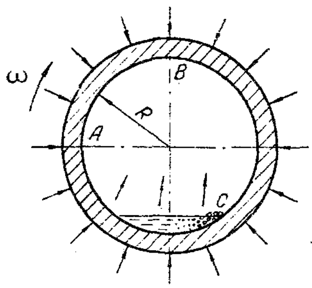

Fig. 3. Schematic distribution of heat-transfer medium for $\omega \ll \omega_m$.

TABLE 1. Comparison of Experimental and Calculated Values of ω_m

Parameter	Heat-transfer medium			
	water			acetone
$q\cdot 10^{-4}$, W/m^2	1,8	1,0	0,4	1,4
ω_m, rad/sec	2,6	1,83	1,06	16,2
ω'_m, rad/sec	1,7—2,5	1,2—2,5	0,8—1,2	$\gg$6,0

the heat admitted externally per unit time in the section ABC is used up in evaporation of the liquid entering this section, i.e.,

$$G\rho_1 r = q2\pi R l_e C, \quad C = \frac{l}{2\pi R} .$$

In turn, $G = \delta_A \omega R l_e$. Then $\delta_A \omega R l_e \rho r = q2\pi R l_e C$, and

$$\delta_A = \frac{2\pi q C}{\omega r \rho} . \tag{2}$$

Solving Eqs. (1) and (2) simultaneously for ω and taking C = 0.7 (complete wetting of the circumference), we obtain an expression for the rotational speed at which the heat transfer is a maximum:

$$\omega_m = 2.54 K q^{3/5} , \tag{3}$$

$$K = \left(\frac{1}{r}\right)^{3/5} \left(\frac{g}{\rho_1}\right)^{3/10} \frac{\sigma^{1/10}}{(R\eta)^{2/5}} .$$

The combination factor $K = 2.9\cdot 10^{-3}$ for water at T = 120°C and $K = 2.34\cdot 10^{-2}$ for acetone at T = 100°C. The calculated values of ω_m are compared with the experimental values ω'_m in Table 1.

It is evident from the given data that the calculated values of ω_m fall within the range of experimental values of ω'_m for the most part. A certain discrepancy between the experimental and calculated values of ω_m for elevated values of the heat flux and the maximum-transfer speed ω_m can be attributed to the thermal inertia of the wall in which the thermocouples are embedded, as well as to the influence of inertial acceleration, which is not included in Eq. (3). It is obvious that this equation is applicable to a limited range of parameters of the heat pipe and heat-transfer medium. Typically, in the pipe containing acetone (Fig. 2, curve 4), the maximum heat transfer is attained at much larger values of ω_m than in the water pipe, although for $\omega < \omega_m$ the heat-transfer rate is of the same order for both.

Visual observations of the heat-transfer process were carried out with the end wall of the evaporator section removed. A ring was inserted in its place to prevent water from escaping. In all operating regimes, a thin film of liquid was observed on the wall behind the pool, its area increasing with the speed ω. Heat transfer took place across this film by evaporation from the surface. For a heat-flux density $q \geqslant 1\cdot 10^4$ W/m^2 and a rotational speed $\omega \leqslant 0.15$ to 0.2 rad/sec, fully developed nucleate boiling of the liquid took place at the entry site of the heated wall into the pool, the area of the pool surface occupied by the zone of fully developed nucleate boiling increasing with increasing value of q and decreasing value of ω. When the rotational speed was increased to ω_m such that the liquid layer covered the entire periphery of the pipe, boiling ceased in the pool for any values of q.

It was observed that effervescence of the rim of the pool and dryout of the wall with a reduction in the rotational speed occurs primarily in the middle part of the evaporator section. Clearly, the heat flux in the middle part, where the thermocouples were placed, greatly exceeds the average heat flux. This fact, along with the thermal inertia of the wall in which the thermocouples are embedded, obviously will have a considerable smoothing effect on the heat-transfer peak at the speed ω_m.

For the investigation of the heat-transfer characteristics of rotating heat pipes with wicks, we used a grid of serge netting with mesh dimensions 0.2 × 0.15 mm and a thickness of approximately 0.5 mm, made of stainless steel, as the porous structure. The washed and

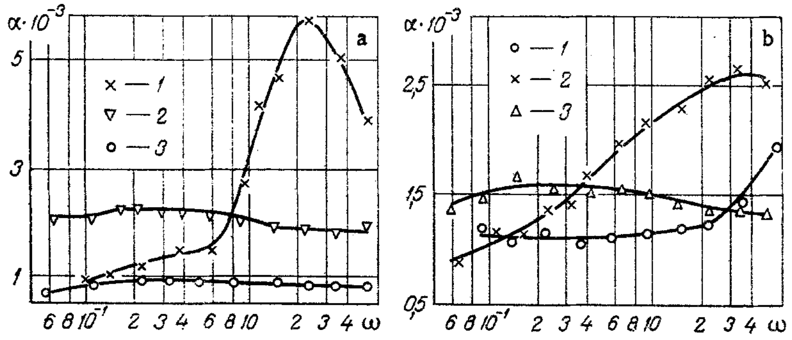

Fig. 4. Perimeter-average heat-transfer coefficient $\alpha \cdot 10^{-3}$, $W/m^2 \cdot deg$, versus rotational speed ω, rad/sec. a) Heat pipe containing water, $q = 1.0 \cdot 10^4\ W/m^2$: 1) wickless; 2) with one grid; 3) with two grids; b) acetone heat pipe: 1) wickless, $q = 1.4 \cdot 10^4\ W/m^2$; 2) with one grid, $q = 1.0 \cdot 10^4\ W/m^2$; 3) with two grids, $q = 0.9 \cdot 10^4\ W/m^2$.

annealed grid and mounting were pressed against the walls of the evaporator section by means of a coil spring with a pitch of 10 to 12 mm, formed from wire with a diameter of 2 mm. Preliminary calculations according to [8] showed that this grid can lift acetone to a height of 4 or 5 cm, and water to a height equal to the pipe diameter, by virtue of capillary forces, but for $q \geq 1 \cdot 10^4\ W/m^2$ one layer of the given grid is not capable of ensuring delivery of the required quantity of heat-transfer medium into the upper part of the braked pipe.

Observations showed that for $q \simeq 1.3 \cdot 10^4\ W/m^2$ the placement of the grid in a heat pipe containing water produces an appreciable reduction in the amplitude of the fluctuations of the local wall temperature. Steady-state cooling of the wall is clearly observed in one layer during one revolution of the pipe with the grid. Obviously, even at the lowest rotational speeds with one grid, the capillary forces ensure wetting of the wall over a sizable portion of the periphery of the pipe. Moreover, the average temperature difference between the wall and the vapor in a pipe with one grid is considerably smaller in this regime than in the pipe without the grid or in the pipe with two grids. The pipe with two grid layers is characterized by a large amplitude of the fluctuations of the local wall temperature at a high overall temperature drop. The capillary forces in the grid of serge netting is probably not sufficient to raise the heat-transfer medium to the height of one pipe diameter under conditions of vigorous evaporation.

The perimeter-average heat-transfer coefficient (Fig. 4a) for two grids is half the value for one grid, and the absolute value of this coefficient corresponds to the thickness of the liquid layer in the wick. In both cases, a very mildly sloping heat-transfer maximum is observed in the interval of rotational speeds 0.2 to 0.3 rad/sec, roughly coinciding with the speed at which the wall-temperature fluctuations die out. The smoothing of the heat-transfer peak in the pipe with the grid is obviously much greater than in the pipe without the grid, because the capillary forces promote admission of the heat-transfer medium to the dried-out part from the end zones of the pipe, where the heat-flux density is smaller.

In an acetone pipe with one grid (Fig. 4b), nonsteady cooling of the wall is observed over a large part of the investigated range of rotational speeds, and a sharp heat-transfer peak occurs at ω_m = 3-4 rad/sec. Characteristically, even for $\omega > 0.8$ rad/sec the heat-transfer rate in the acetone pipe with one grid is higher than in the same pipe with water. The heat transfer in a pipe with two grid layers is also approximately 1.5 times greater than in the corresponding pipe with water. Clearly, heat transfer takes place in the nucleate boiling regime in an acetone pipe in the liquid layer confined by a grid at the wall of the pipe, and part of the heat-transfer medium is ejected from the wick in the droplet state during boiling. A mildly sloping heat-transfer maximum in the acetone pipe with two grids is observed for roughly the same values of ω as in the water pipe with grids. It is seen that the use of a single-layer wick provides more intense heat transfer in the vicinity of the maximum than a two-layer wick, but the two-layer wick decreases the value of ω_m by 1/40 to 1/60, while the single-layer wick decreases it only by 1/3 to 1/4 in comparison with the wickless pipe. Clearly, the thicker the wick, the greater will be its useful range with respect to speed of rotation, but the lower will be the heat-transfer rate for $\omega < \omega_m$ and vice versa.

Consequently, the net effectiveness of using wicks in such heat pipes is comparatively slight.

The foregoing discussion indicates that the problem of whether to recommend the use of wicks in rotating heat pipes and of their construction must be solved as a function of the specific conditions of their application with regard for the minimum required rotational speed of the pipe, the properties of the heat-transfer medium, and several other factors. Thus, under the conditions of regulated electrical machines, the quantity of heat-transfer medium needed to handle the shafts and heat pipes is determined (with allowance for possible axial misalignment of the pipe cavity and shaft) from the condition of ensuring an average thickness of the condensate layer in the evaporator section of 0.3 to 0.5 mm. For evaporation from the surface, this corresponds to a heat transfer of $1 \cdot 10^3$ to $2 \cdot 10^3$ $W/m^2 \cdot deg$. It is evident from Figs. 2 and 4 that the same average heat-transfer level is observed in wickless pipes containing water and acetone for $0.3 < \omega < \omega_m$, in which case the pipe operates in the nonsteady state. However, whereas the wick only slightly increases the heat transfer at these speeds in the water heat pipe, a single-layer wick in the acetone pipe increases the heat transfer by a factor of 1.5 to 2.0 in the interval of rotational speeds from 0.4 to 4 rad/sec. We infer on this basis that it is unnecessary to use a wick in regulated electrical machines with heat pipes having a diameter of about 100 mm for $q < 2 \cdot 10^4$ W/m^2 and a minimum speed of 0.3 rad/sec with water as the heat-transfer medium; if a heat-transfer medium of the acetone type is used, it is advisable to equip the pipe with a wick consisting of one layer of a grid of serge netting. If the required minimum speed under the same conditions is, say, 0.05 rad/sec, then it follows from Figs. 4a and 4b that a single-layer wick should be used in a water pipe, and a two-layer wick in an acetone pipe.

NOTATION

α, heat-transfer coefficient, averaged over perimeter of heat pipe; $\bar{\alpha}$, heat-transfer coefficient, averaged over period of rapid descent of heat-pipe wall temperature; q, heat-flux density supplied by radiation; $\bar{q}$, average heat flux due to heat capacity of wall material; δ_w, thickness of pipe wall; ρ, density of wall material; T_1, T_2, maximum and minimum wall temperatures in the period analyzed; $\Delta\tau$, period; ΔT, temperature difference between average wall temperature over period $\Delta\tau$ and vapor temperature T_v; δ_A, thickness of condensate film at point A (Fig. 3); ω, angular speed of rotation of heat pipe; R, radius of evaporator section of pipe; η, dynamic viscosity coefficient of condensate; σ, coefficient of surface tension; ρ_1, density of condensate; g, acceleration of gravity; r, latent heat of vaporization of condensate; l_e, part of pipe circumference coated with evaporative film; ω_m, angular speed of rotation of pipe when maximum heat transfer is observed; c, specific heat of pipe wall material.

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