CLOSED EVAPORATIVE COOLING OF ROTATING ELECTRICAL MACHINES

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Elements of a theory for centrifugo-axial heat pipes are described along with a calorimetric device for their experimental investigation. Experimental data presented confirms the fundamental correctness of the initial theoretical hypotheses.

At the present time, research is going forward on the use of thermal superconductors — centrifugal and wick heat pipes and two-phase thermosiphons in cooling systems for electrical machines and equipment.

As is well known, thermal superconductors can be made up in the form of independent elements of a cooling system or the properties of thermal superconductors can be imparted to the usual elements or structural parts. In rotating electrical machines, for example, the shaft of the machine may play the part of a thermal superconducting element if it is prepared in the form of a centrifugo-axial heat pipe, i.e., it is made hollow, a small amount of intermediate coolant is placed in it, air is removed, and it is hermetically sealed [1]. When the shaft rotates around its longitudinal axis, the liquid coolant is distributed by centrifugal forces over the internal wells in the form of an annular layer. With heat delivered to one section of the shaft and cooling of another section, continuous closed evaporative cooling of the heated section occurs and heat transfers into the cooled section in vapor form when there is negligibly small internal thermal resistance in the cavity.

Heat transfer in the heat-exchange sections of a centrifugal heat pipe is described by the equations

$$T = t_c - q_1 R_1 \tag{1}$$

and

$$T_1 = t + q_2 R_2. \tag{2}$$

The thermal resistance of the heat-exchange sections is given by

$$R_1 = d_i \left(\frac{1}{\alpha_1 d_{2i}} + \frac{1}{2\lambda} \ln \frac{d_{1i}}{d_i} \right) \tag{3}$$

and

$$R_2 = d_k \left(\frac{1}{\alpha_2 d_{1k}} + \frac{1}{2\lambda_1} \ln \frac{d_k}{d_{2k}} + \frac{1}{2\lambda} \ln \frac{d_{1k}}{d_k} \right). \tag{4}$$

In accordance with the continuity condition, the thermal flux density in the heat-exchange sections is inversely proportional to their area, i.e.,

$$q_2 = q_1 \frac{l_1 d_i}{l_2 d_k}. \tag{5}$$

Studies of two-phase thermosiphons [2] show that the vapor temperature in them is practically constant over the entire length of the vapor space, i.e.,

$$T = T_1. \tag{6}$$

One can assume that this also holds for centrifugo-axial pipes since the pressure differential needed for vapor displacement is negligibly small for moderate loads.

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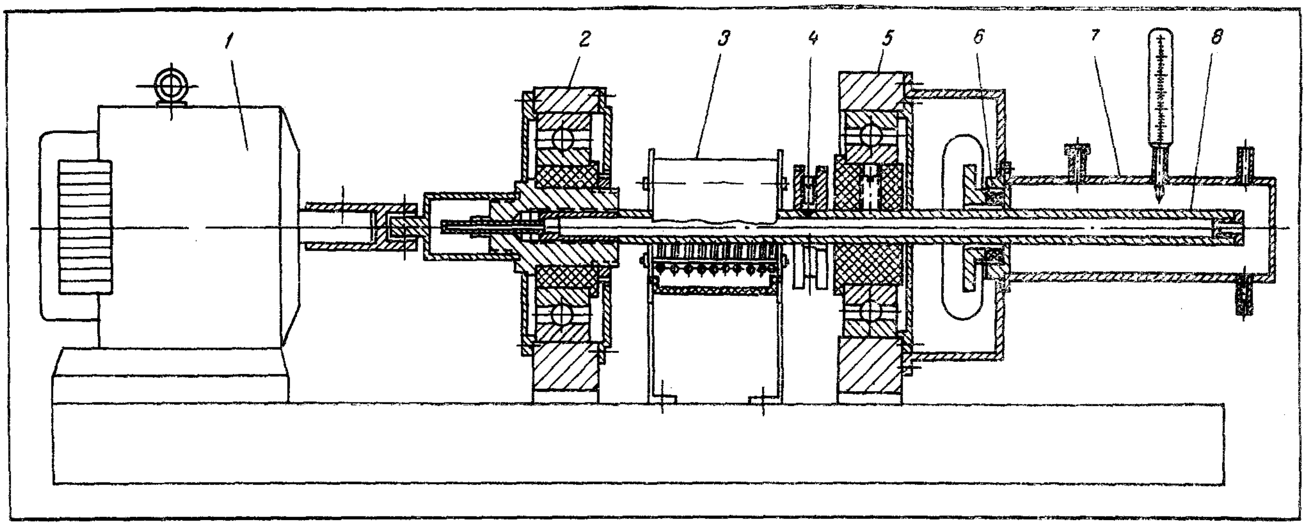

Fig. 1. Diagram of calorimetric apparatus for investigation of centrifugo-axial heat pipes: 1) dc motor; 2) front support; 3) electric heater; 4) thermocouple; 5) rear support; 6) chiller packing gland; 7) continuous-flow water chiller; 8) heat pipe.

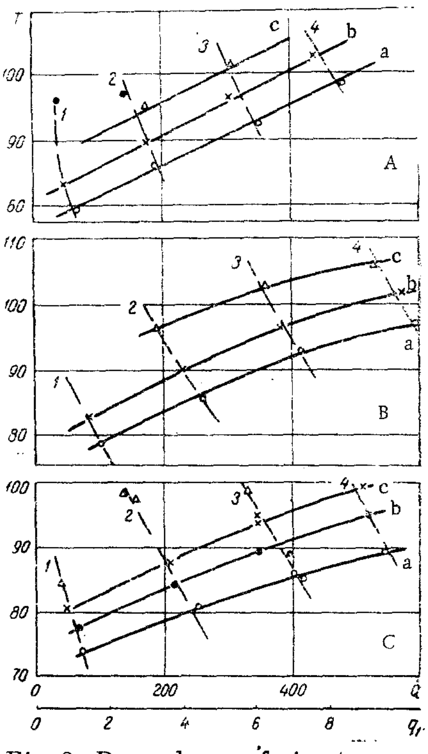

Fig. 2. Dependence of pipe temperature on transmitted thermal power: A) for l_2 = 100 mm and i = 1/860; B) for l_2 = 150 mm and i = 1/860; C) for l_2 = 100 mm and i = 0; water temperature at calorimeter discharge (a) 30, (b) 40, and (c) 50°C; heater power (1) 250, (2) 500, (3) 750, and (4) 1000 W. Q, W; q_1, W/cm^2; T, °C.

Using Eqs. (5) and (6), Eq. (2) takes the form

$$T = t + q_1 \frac{l_1 d_i}{l_2 d_k} R_2. \tag{7}$$

From Eqs. (1) and (7) it is easy to obtain an expression for the determination of the amount of heat transferred by the pipe,

$$Q = \frac{\pi d_i d_k l_1 l_2 (t_c - t)}{R_1 l_2 d_k + R_2 l_1 d_i}. \tag{8}$$

This equation was made the basis for experimental studies carried out on a special calorimetric apparatus (Fig. 1). The heat pipe was prepared from a brass tube with internal diameter d = 15 mm, wall thickness δ = 2 mm, and length l = 500 mm. Distilled water was used as the coolant. Heat was supplied from a fixed electrical heater by radiative heat exchange. The condenser section of the pipe was placed in a continuous-flow water calorimeter. The pipe was rotated by a dc motor. Rate of rotation was determined with a strobotachometer.

A chromel–constant thermocouple was built into the wall of the pipe in the thermally insulated (transport) section between the heater and chiller in order to determine the coolant temperature. The temperature was measured 10-20 sec after shutdown of the pipe with the heater turned on. Since the thermal flux through the wall of the thermally insulated section was insignificant, the readings of this thermocouple were taken to be the average temperature of the vapor in the pipe [3]. Because of temperature separation of the water in the rotating flow, the temperature t of the cooling water at the calorimeter discharge was taken as its mean temperature.

The purpose of the experiments was to check the basic correctness of the initial hypotheses assumed in the derivation of Eq. (8). If the right and left sides of Eqs. (1) and (7) are added, one can obtain an expression for the determination of the vapor temperature in the pipe,

$$T = \frac{1}{2}\left[t_c + t - q_1\left(R_1 - \frac{l_1 d_i}{l_2 d_k} R_2\right)\right].$$

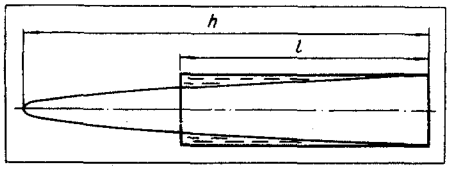

Fig. 3

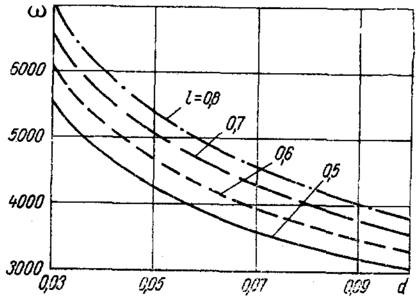

Fig. 4

Fig. 3. Diagram of fluid distribution for vertical axis of rotation of pipe.

Fig. 4. Dependence on pipe length l of required rate of rotation of vertical pipe with diameter d for an average thickness $\delta = 0.5$ of wall layer of fluid. ω, rpm; d, m

This equation indicates that with other conditions kept constant, the vapor temperature in the pipe should increase by $\frac{1}{2}\Delta t$ when there is an increase in the temperature of the cooling water by an amount Δt, i.e., when

$$t_c - q_1\left(R_1 - \frac{l_1 d_i}{l_2 d_k} R_2\right) = \text{const} \tag{9}$$

$$\Delta T = \frac{1}{2}\Delta t. \tag{10}$$

Favorable conditions are created for the satisfaction of Eq. (9) in centrifugo-axial pipes with radiative input of heat and small variations in the temperature t. The degree of blackness of the centrifugo-axial pipe and its location with respect to the heater was not changed during the tests. Consequently, with the constant thermal flux q_1 = const, the temperature t_c was also practically unchanged. A constant rate of rotation and considerable thickness of the wall layer of fluid ensured constant thermal resistance in the heat-exchange sections.

In line with what has been said, observations were made at a constant rate of rotation of the pipe, n = 500 rpm, and for a constant amount of water in the pipe, w = 10 cm^3 (average thickness of wall layer δ = 0.45 mm), and consisted of measurements of the pipe temperature T with changes in the power supplied to the heater, which was varied over the steps 250, 500, 750, and 1000 W, and of measurements of the water temperature t at the calorimeter discharge (it was varied over the basic steps 30, 40, and 50°C). Figures 2A and 2B show the results of measurements for a tilt i = 1/860 of the axis of rotation toward the heater and condenser-section lengths l_2 = 100 mm (A) and l_2 = 150 mm (B). Figure 2C shows the results of measurements for a strictly horizontal axis of rotation and a condenser-section length l_2 = 100 mm. The length of the evaporator section was kept constant and was 100 mm.

The curves make it clear that in all cases with constant thermal flux (q_1 = const), a rise in the temperature of the water in the calorimeter by 10°C leads to an increase of the temperature T in the pipe by 5-6°C.

Significant deviations from this behavior were only observed in particular cases where the pipe temperature approached 100°C. They are evidently explained by deterioration of heat-exchange conditions in the calorimeter because of separation of gas bubbles from the cooling water which were concentrated on the surface of the pipe through the action of centrifugal forces. Evidence of this was the vigorous liberation of gas bubbles when there was a reduction in the rate of rotation of the pipe at that time.

Thus the experimental data confirms the fundamental correctness of the physical model used in the derivation of Eq. (8). In these experiments, the temperature of the outer surface of the pipe in the evaporator section was not measured and it is therefore impossible to arrive at a value for the heat-transfer coefficients. Rough heat calculations have an error of 20-30% and higher, and furnish justification for assuming that a certain amount of noncondensible gases was present in the test pipe. These gases are of higher density than the vapor and, entering the centrifugal force field, are displaced from the axis of rotation toward the periphery and are distributed at the phase interface. As is well known [4], the amount of additional thermal resistance created by these gases depends on their concentration and velocity. Therefore,

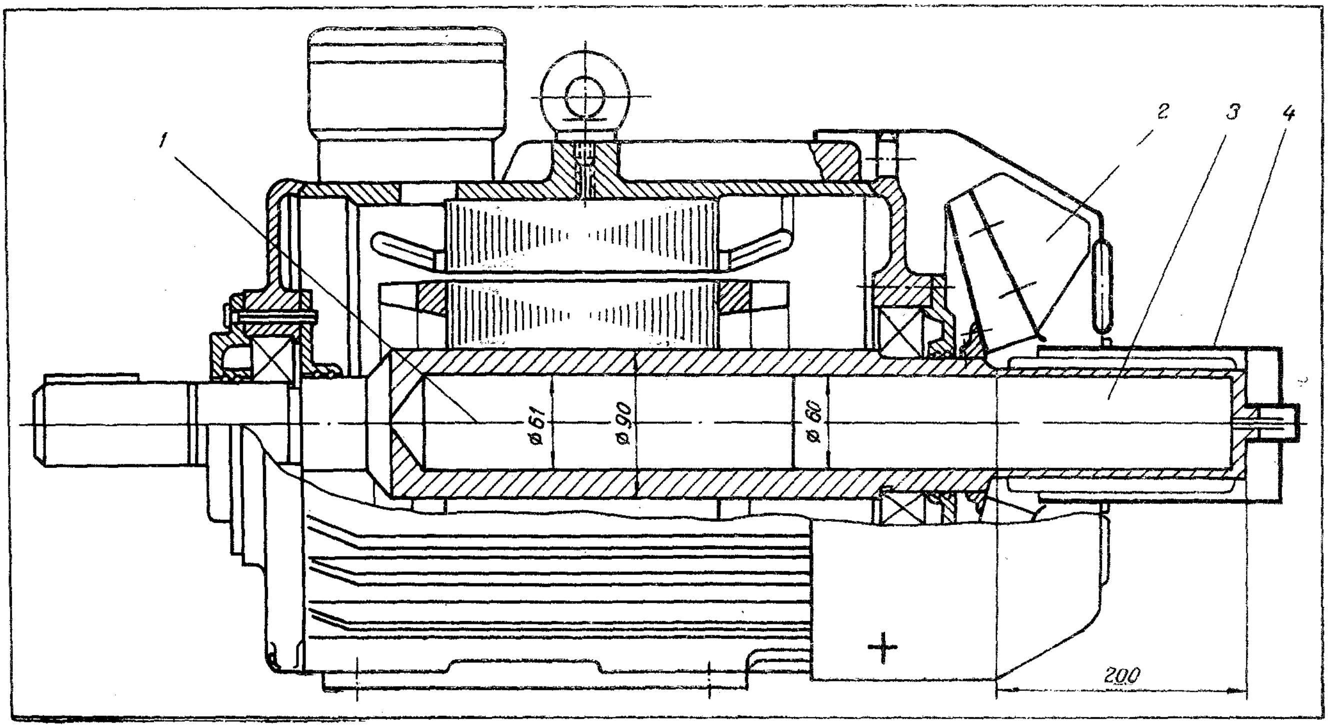

Fig. 5. 4AC-160-M 4-asynchronous motor with heat pipe in shaft: 1) evaporator section of pipe; 2) two-bladed fan; 3) condenser section of pipe; 4) flow-directing pipe.

this resistance decreases as the thermal load increases. It is also possible that the area of the condenser section increases as the thermal loading increases, being filled with a turbulent flow, and the thermal resistance of the condensate layer in this section is reduced because of this.

A comparison of the curves in Figs. 2A and 2B indicates that with other conditions held constant, an increase in the length of the condenser section from 100 to 150 mm leads to a reduction of about 10°C in the vapor temperature in the pipe and to an increase in the amount of transmitted heat by 8-12%. For a tilted pipe, the increase in the length of the condenser section is accompanied by an increase in the average thickness of the wall layer of fluid in this section and a corresponding increase in its thermal resistance, which somewhat reduces the effect of condenser-section length on the amount of heat transmitted by the pipe.

A comparison of the curves in Figs. 2A and 2C shows that imparting a strictly horizontal position to the pipe gives rise to an increase in transferred power by 20-25% for a small reduction in pipe temperature. More detailed studies showed that with an increase in the tilt angle of the axis of rotation toward the evaporator section from 0 to 90°, the amount of heat transmitted by the pipe increases linearly. Furthermore, the effect of the tilt angle of the axis of rotation depends on pipe diameter, thermal flux density in the heat exchange sections, the amount of coolant, and a number of other factors.

When the axis of rotation is tilted toward the condenser, the power transmitted by the pipe is reduced to practically nothing and rapid overheating of the evaporator section occurs, i.e., the pipe loses the properties of a thermal superconductor. Obviously, the assumed rate of rotation ($n = 500$ rpm) was insufficient for distribution of the fluid over the entire length of the pipe. For $n = 500$ mm, the centrifugal forces exceed the gravitational force by a factor of 2 at most. This is responsible for the wave nature of fluid motion.

When using closed evaporative cooling in electrical machines, there is great practical interest in what the conditions are under which a centrifugo-axial pipe can operate efficiently for any spatial configuration of the heat-exchange sections. To answer this, the limiting case where the axis of rotation of the pipe is vertical was considered with the input of heat in the upper portion. It is well known that the surface of a fluid rotating together with a vertical pipe is a paraboloid of revolution. Calculations show that for the height of the paraboloid of revolution to be $l = 500$ mm in a pipe with a diameter $d = 15$ mm, it is necessary to rotate the pipe at a rate of 5300 rpm. In this case, however, the thickness of the wall layer of the fluid

TABLE 1. Thermal Test Results for 4AC-160-M4 Motor

Temperature rise of operating motor parts, °C	Without heat pipe	With heat pipe in shaft			
		$w=60$ cm^2	$\Delta\theta$	$w=30$ cm^2	$\Delta\theta$
Stator winding, average	112,8	92,3	—20,5	87,3	—25,5
Slotted part of rotor	169,7	97,2	—72,5	86,5	—83,2
At point of rotor fitting on shaft	174,5	89,2	—85,3	80,0	—94,5
Back of stator	62,7	48,5	—14,2	46,5	—16,2
Slotted portion of the winding	94,0	69,5	—24,5	67,0	—27,0
Frontal portion of winding on fan side	112,2	89,5	—22,7	87,5	—24,7
Frontal portion of winding on drive side	123,7	96,2	—27,5	92,5	—31,2

in the lower portion of the pipe will reach the size of its radius, and the use of the pipe for transmission of heat loses practical meaning because of the low thermal conductivity of low-temperature coolants. It is obvious there is practical interest in the case where the surface of the paraboloid of revolution intersects the bottom of the container and the liquid is distributed over the walls in the form of a thin layer (Fig. 3).

The angular velocity required for distribution of a given volume w of fluid over the entire length of a pipe for the case where the height h of the paraboloid of revolution is greater than the length of the pipe is given by

$$\omega = l\sqrt{\frac{\pi g}{w}}.$$

Figure 4 shows the dependence of the required rate of rotation on pipe diameter for various lengths and an average thickness $\delta = 0.5$ mm of the wall layer of fluid (maximum thickness of layer in the lower portion is $\delta_{max} = 1.0$ mm). As is apparent, pipes which are typical of electrical machines of medium power ($l = 0.5$–0.8 m, $d = 0.03$–0.06 m) can operate with any spatial arrangement of the heat-exchange sections at rates of rotation above 4000-7000 rpm or in the absence of gravitational forces. In ordinary electrical machines (with $n < 3000$ rpm), one can look toward the use of centrifugo-axial pipes for a horizontal shaft and also for the input of heat from below.

The correct choice of the amount of coolant place in a centrifugo-axial pipe is an important condition for its efficient operation.

In the ideal case, the necessary amount of coolant can be determined by hydraulic calculations based on the following hypotheses. With evaporation of the coolant at one end of the pipe and condensation at the other, there arises in the fluid layer a difference in levels between the evaporation and condensation sections. Since the annular layer of fluid is in a centrifugal force field, the pressure differential resulting from the difference in levels is proportional to the square of the angular velocity and is found from the expression

$$\Delta p = \frac{\rho}{8}\,\omega^2 (d_{2k}^2 - d^2). \tag{11}$$

This pressure differential ensures automatic return of condensate to the evaporator section and is expended in overcoming the resistance of fluid friction with respect to pipe wells and to the opposing flow of vapor and also in the creation of a velocity head for the vapor and fluid. The inertial components are negligibly small for laminar flow of vapor and fluid. One can show that the pressure differential required in this case in order to overcome the resistance of fluid friction with respect to the walls of the pipe and to the opposing flow of vapor is determined by the equation

$$\frac{dp}{dl} = \frac{6\mu}{\pi\delta^2}\left[\frac{Q}{\delta(d-\delta)} + \frac{64}{3}\cdot\frac{Q_v\delta^2}{(d-2\delta)^4}\cdot\frac{\mu_v}{\mu}\right]. \tag{12}$$

If Eqs. (11) and (12) are solved simultaneously with respect to δ, the necessary amount of coolant can be determined. As is obvious, the amount of coolant depends on the rate of rotation, coolant density and viscosity, amount of heat transmitted, the length and diameter of the pipe, and many other factors.

Under actual conditions, the required amount of coolant also depends on the accuracy of preparation and the quality of the machining of the internal cavity of the heat pipe in addition to the factors mentioned. The slightest misalignment of the internal cavity of the pipe with respect to the axis of rotation leads to an increase in the thickness of the fluid layer in some sections and bareness in others. On the one hand, coarse machining of the surface gives rise to an increase in hydraulic resistance to axial displacement of the fluid; on the other hand, transverse grooves, acting like the capillary structure in arterial wick heat pipes [5], facilitate distribution of fluid over the walls of the pipe. These factors are apparently of decisive significance in actual heat pipes.

For a preliminary evaluation of the required amount of coolant, two heat pipes 15 mm in diameter and 500 mm long were tested to burnout at 500 rpm. The heat pipe in which the thickness of the distilled water layer was 0.2 mm lost the properties of a thermal superconductor at a thermal flux density of 3 W/cm^2 in the evaporator section. With a layer 0.45 mm thick, the maximum thermal flux density in the second pipe was 12 W/cm^2. These limiting values refer to seamless brass pipe not subjected to any additional machining except straightening. Until there is a more detailed study of the problem, one can recommend an amount of coolant based on a requirement for the provision of a wall layer of fluid 0.3 mm thick.

In using centrifugo-axial pipes for cooling electrical machines, there is great practical interest in the way in which rate of rotation effects heat transmission along the pipe. The extent of the effect of rate of rotation on the amount of heat transmitted by the pipe depends on the spatial position of the axis of rotation, pipe geometry, the amount of coolant, etc. Our studies have shown that with a tilt of $i = 1/860$ in the pipe axis toward the heater, an increase in rate from 500 to 2000 rpm leads to an increase of 3-5% in the amount of heat transmitted. According to calculations [4], the coefficient of heat transfer from the pipe wall to the cooling medium increases by a factor of 2.5 in this case. Simultaneously, however, equalization of the thickness of the wall layer of fluid occurs over the length of the pipe, the thickness of the layer increases in the condenser section, and the total thermal resistance of the pipe is increased. The effects of these factors is mutually compensated, which also ensures a slight effect of the rate of rotation on heat transmitted under these conditions.

At low rates of rotation (of the order of 50-100 rpm) and with a pipe arranged horizontally, the main mass of the fluid is obviously located in the lower portion of the pipe and the fluid is distributed over the internal wall in the form of a thin film because of surface-tension forces (wetting). If the film does not evaporate during a single revolution, heat transmission in the heat-exchange sections may even increase somewhat at these rates.

As far as distribution of the fluid along the pipe is concerned, it is well known that with laminar flow of a fluid between two cylinders and maximum eccentricity, the loss of pressure in longitudinal displacement of the fluid is 2.5 times less than for coaxial arrangement of the cylinders. A set of experiments at the rates of 400, 300, 200, 100, and 50 rpm, which was performed with a tilt of $i = 1/860$ of the axis of rotation toward the heater, showed that deviation from the results obtained at $n = 500$ rpm was within the limits of 3-5%.

During the study, a maximum radial thermal flux density in the evaporator section of 12.6 W/cm^2 was recorded. The thermal power flux in the axial direction reached 600 W or 340 W per cm^2 of pipe cross section. Rough calculations demonstrated that in commercial electrical machines with a rotor temperature rise to 120-150°C, the radial thermal flux density in a pipe realized within the shaft of the machine may reach 2-4 W/cm^2 and the total thermal power of the pipe may reach several hundred watts with sufficiently intense heat removal.

Based on the results of the study, a 4AC-160-M4 three-phase asynchronous motor with a nominal power $P_2 = 17$ kW was assembled and tested with closed evaporative cooling of the rotor (Fig. 5). The shaft diameter in this motor was increased by a factor of 1.5 as compared with commercial motors, and axial channels in the rotor were eliminated. The pipe diameter in the motor was taken to be 60 mm in the condenser section, which extended outside the housing, and 61 mm in the evaporator section. The extended end of the shaft was lengthened by 150 mm and provided with small longitudinal ribs. The construction of the fan provided a blast of air around the extended portion of the shaft at a velocity of 15-20 m/sec. Distilled water was used as the intermediate coolant. Some results of motor tests at $P_2 = 18.9$ kW are shown in Table 1.

Analysis of the data presented shows that the use of a centrifugal heat pipe made it possible to reduce the average temperature rise of the stator winding by 20-25°C (18-22%) and of the rotor by 72-83°C (43-49%). Reduction in the amount of intermediate coolant from 60 cm^3 to 30 cm^3 led to an additional reduction in the temperature of the stator winding by 5°C and of the slotted portion of the rotor by 11°C. Calculations showed that up to 70% of the heat deposited in the rotor (400-500 W) is removed through the shaft in the rotating heat exchanger for a motor with a heat pipe. In addition, use of a heat pipe reduced the nonuniformity of the temperature distribution over the operating portions of the motor. Through the use of a heat pipe, the overall power of a 4AC-160-M4 motor can be increased by 20-25% without changing the heat-resistance class of coil insulation.

Thus we have experimentally demonstrated the feasibility of using centrifugal heat pipes for the intensification of cooling in rotating electrical machines of medium power. However, one should keep in mind that it is apparently not feasible to use heat pipes in machines with power above 30-40 kW since the rotor temperature and shaft diameter change little with increase in machine power.

NOTATION

T, vapor temperature in evaporator (temperature of tube); T_1, vapor temperature in condenser; t_C, temperature of external tube surface in evaporator; t, cooling fluid temperature; q_1, q_2, heat flux density in evaporator and condenser; R_1, R_2, thermal resistance of evaporator and condenser; d, d_1, internal and external diameters of heat pipe; d_2, mean diameter of internal surface of annular liquid layer; λ, pipe wall material thermal conductivity; λ_1,thermal conductivity of liquid heat transfer agent; α_1, heat transfer coefficient in liquid evaporation; α_2, heat transfer coefficient from external pipe wall in condenser to cooling medium; l_1, l_2, length of evaporator and condenser sections; Q, amount of heat transferred by pipe; δ_1, thickness of pipe wall; l, pipe length; ΔT, vapor temperature increment in pipe; Δt, cooling medium temperature increment; n, velocity of pipe rotation; δ, mean thickness of wall liquid layer; N_H, power supplied to heater; i, pipe axis inclination to horizon; w, liquid volume in tube; ω, angular velocity of pipe rotation; β; density of liquid heat transfer agent; Δp, liquid pressure drop; μ_V, dynamic viscosity of liquid and vapor; Q, Q_V, liquid and vapor flow rates; $\Delta\theta$, difference in temperature excess of active engine parts.

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