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EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN A CENTRIFUGAL HEAT PIPE WITH AN OPTIMIZED LAYER OF HEAT-TRANSFER AGENT

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This paper describes a technique and gives results of experimental investigations of special features of heat transfer in the evaporator and condenser sections of a centrifugal heat pipe with optimized thickness of the layer of heat transfer agent.

Centrifugal heat pipes of various constructions [1] have found use recently to intensify cooling of rotary electric machinery. Internal heat transfer is somewhat higher in conical heat pipes than in cylindrical ones. However, it is a considerable technical problem to make long conical heat pipes ($l/d > 5$), especially with mass discharge. One way to intensify the heat transfer in cylindrical pipes is to use a condenser section of smaller diameter than that of the transfer and evaporator sections [2]. In a pipe of this construction the thickness of condensate layer increases in a stepwise manner towards the closed end of this section, and is therefore considerably less than in the simple cylindrical pipe, where it increases over the whole pipe length.

Another efficient way to intensify the heat transfer in a heat pipe is a device at the closed end of the evaporator section of the annular channel, for pouring away the excess condensate. Calculations presented in [3] have shown, for example, that if we fill a cylindrical heat pipe with heat-transfer agent for a design angular velocity of 70 rad/sec, and then increase the speed to 300 rad/sec, then up to 70% of the condensate is excess, and will lower the heat-transfer rate at low heat fluxes. If we provide an annular groove at the closed end of the evaporator section, the excess condensate will automatically flow into the groove, the result being that for any rotational speed the evaporator section will contain the minimal necessary (i.e., optimal) condensate layer.

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Experimental investigations of heat transfer in a heat pipe with a step and a groove (with the optimal layer of agent) were conducted on a calorimetric equipment with convective supply and removal of heat, by locating the heat-transfer sections in counterflow water heat exchangers. The diameter of the evaporator section was 36 mm, and of the condenser 28 mm. At the closed end of this section we made an annular groove of width 15 mm and depth 9 mm (of 19-cm^3 volume). On the outer surface the heat-transfer sections had an annular fin to increase the heat flux density. In the pipe were embedded 14 thermocouples, located in the vapor space and caulked into the body of the heat pipe. The thermocouple readings were picked off via brushes and were measured on a potentiometer.

Before the experiments the pipe was washed with acetone and alcohol, was evacuated down to 10^{-3} torr, and was then charged with heat-transfer agent via the stem in the end cover. The heat pipe was rotated by means of a dc motor. During the experiment the angular velocity could be varied from 7 to 200 rad/sec, and the heat-transfer rate in the evaporator section could be varied from $1.5 \cdot 10^4$ to $2 \cdot 10^5$ W/m^2. The working liquids used were water and acetone. The vapor temperature in the tube with water varied from 46 to 75°C (vapor pressure from 0.1 to 0.4 bar), and in the pipe with acetone, from 60 to 80°C (P = 1.15–2.15 bar). The required vapor temperature and heat flux density were achieved smoothly by controlling the temperature of the water circulating in the heat exchangers. The heat-transfer coefficients were calculated from the experimental data by conventional methods, allowing for leakage of heat in the pipe body, and heat release in the seals of the exchangers.

The investigations were performed into two phases.

We first performed control experiments with excess agent in the pipe, for which the heat pipe was charged with 104 cm^3 of agent (water or acetone). The average thickness of condensate layer in the evaporator section was 3.5 mm. The experiments showed that with bubble boiling in the layer the heat-transfer intensity depends on the heat flux density to the power n = 0.7, as it does with large-volume boiling. With an increase in rotation rate and a decrease of vapor pressure, the heat-transfer intensity is reduced. For example, for a heat flux of $6 \cdot 10^4$ W/m^2 an increase of angular velocity by a factor of 2.7 leads to heat transfer being reduced from $7 \cdot 10^3$ to $5 \cdot 10^3$ W/m^2·deg, i.e., by almost 30%. Evidently, the decrease of density of active vapor-forming centers and of breakaway diameters of bubbles is not compensated for by an increased separation frequency, and this leads to reduced α. For an angular velocity of 192 rad/sec (68 g loading) and a water vapor pressure of 0.1 bar, the transition from surface evaporation to bubble boiling occurs at a heat flux of $5.5 \cdot 10^4$ W/m^2, which is greater by a factor of 3-5 than the transition heat flux for large-volume zero-acceleration boiling.

As a whole these experiments confirmed the previously known laws of heat transfer in the evaporator section of cylindrical heat pipes [4], and showed that the construction of the calorimeter equipment and the measurement method used yield satisfactory results.

We studied special features of heat transfer in the pipe with optimal layer with the pipe charged with 23 cm^3 of agent. This amount of agent was enough to create the optimal layer over the entire above range of change of rotational velocity and heat flux.

In a water heat pipe at constant 0.31-bar pressure we noted that, with increase in the angular velocity, the heat flux at which there occurs transition from surface evaporation to boiling increases, and simultaneously the relation $\alpha = f(q)$ becomes steeper (the exponent n of q increases from 0.65 to 1.05). The curves converge at a heat flux of 10^5 W/m^2. A similar dependence was observed earlier in a coaxial centrifugal heat pipe [5]. This behavior of the relation $\alpha = f(q)$ means that in the range of heat loading from transition to 10^5 W/m^2 the heat transfer decreases with increase of rotational velocity, and here the less is the heat flux level the stronger is its influence. The last dependence can be seen clearly in Fig. 1, which shows data on the influence of rotational velocity on the heat transfer in the evaporator section at a constant vapor pressure of 0.33 bar. For example, at $q = 1.1 \cdot 10^5$ W/m^2 a change of angular velocity in the range from 60 to 192 rad/sec has practically no influence on the heat transfer, while at $q = 6 \cdot 10^4$ W/m^2 a change in velocity in the same range leads to a heat-transfer reduction from $5.8 \cdot 10^3$ to $4.8 \cdot 10^3$ W/m·deg, i.e., by 17%.

With an increase of rotational speed there is a decrease in the mobility of the layer of heat-transfer agent (see below) and a decrease in the convective component of heat transfer. It is known that the less is the heat flux, the larger is the fraction of heat transferred by convection. Therefore, the influence of rotation frequency increases as the heat flux diminishes.

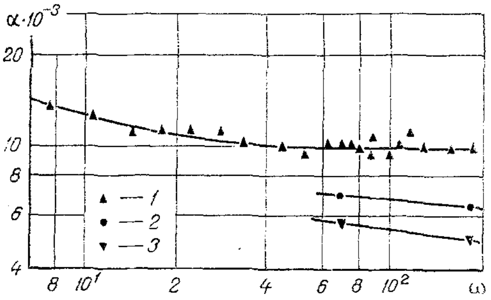

Fig. 1

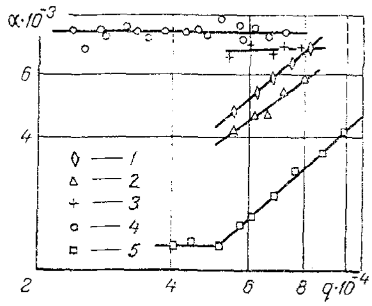

Fig. 2

Fig. 1. The influence of angular rotational velocity on the heat transfer in the evaporator section, at a heat flux density of: 1) $1.1 \cdot 10^5$ W/m^2; 2) $8 \cdot 10^4$; 3) $6 \cdot 10^4$. ω, rad/sec.

Fig. 2. The heat transfer in the evaporator section as a function of the heat flux density at a rotational velocity of 192 rad/sec and a vapor pressure: the amount of water is 23 cm^3: 1) 0.37 bar; 2) 0.29; 3) 0.20; 4) 0.10; with amount of water 104 cm^3: 5) 0.10 bar.

Figure 2 shows results of the investigation of the influence of vapor pressure on heat transfer in the evaporator section of a heat pipe with an optimized water layer. For pressure variation in the 0.37-0.29-bar range the usual dependence was observed; with reduced pressure the heat transfer was reduced, and the experimental data lie on a parallel curve. At a pressure of 0.2 bar and less the heat transfer increases to $73 \cdot 10^4$ W/m^2.deg and becomes independent of the heat flux density. For comparison Fig. 2 shows data on the influence of heat flux density on the heat transfer at a pressure of 0.1 bar and with excess heat-transfer agent (layer thickness 3.5 mm). In this case we observed the usual relation $\alpha = f(q)$, although the general heat-transfer level was quite low. These data show that the above noted independence of the heat transfer with respect to heat flux density arises from a combination of low condensate film thickness and low saturation pressure and is associated with the onset of bubble boiling over the entire range of heat flux density examined.

Interesting results were obtained in an investigation of the influence of rotational velocity on heat transfer in a cylindrical condensate section of finite length. The experiments were performed at a constant heat flux density in the condenser section of $7.3 \cdot 10^4$ W/m^2 and a constant vapor pressure of 0.33 bar. The condenser section diameter, as was mentioned above, was 28 mm, and it was less than the diameter of the evaporator section.

It can be seen in Fig. 3 that in the range of variation of angular velocity from 63 to 116 rad/sec the heat transfer intensity varies sharply, and we observe a peculiar hysteresis in the heat transfer. In the case when the experiments were conducted with a gradual increase in the rotational velocity, a sharp jump in heat transfer intensity is observed at higher rotational velocities (116 rad/sec) than in the case when the experiments are conducted with a gradual decrease in the rotational velocity (63 rad/sec).

The hysteresis can be explained by the wavelike nature of motion of liquid in the layer at the reduced rotational velocities, when the force of earth gravity begins to appreciably affect the liquid motion. The wavelike nature of the liquid motion in a rotating heat pipe at reduced rotational velocities has been observed visually earlier and was described in [6, 7]. A sudden appearance and disappearance of waves was observed at specific rotational velocities. The forming of waves stems from the fact that the pressure in the rotating liquid layer constantly changes due to the earth gravity acceleration, reaching a maximum when a given liquid particle is located in the lower part of the pipe, and a minimum when it is located in the upper part. Therefore, a liquid particle, in going from the upper position to the lower and back again tends to move to the reduced pressure zone and performs oscillations at each turn: in the upward motion it moves toward the pipe wall and in the downward motion it moves away from the wall.

The possibility of waves arising is governed, evidently, by the thermophysical properties and the layer thickness of the heat transfer agent, and also by the ratio of the earth gravity force and the centrifugal acceleration and liquid mass inertia forces. The generation of waves leads to the formation of a convective heat-transfer component in the layer and to an

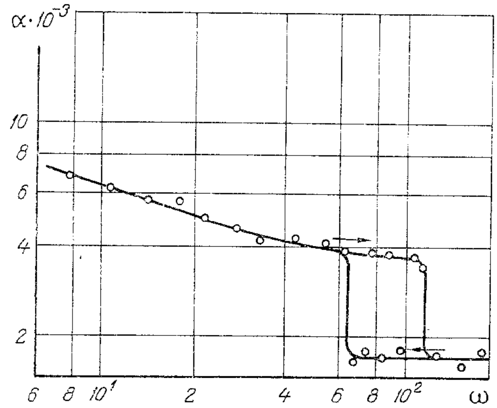

Fig. 3

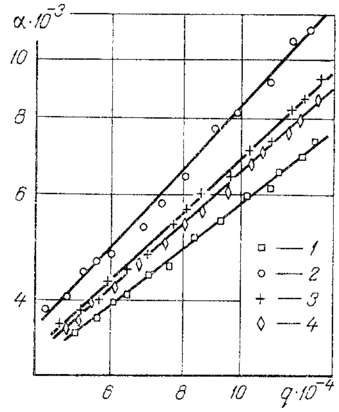

Fig. 4

Fig. 3. The influence of rotational velocity on the heat transfer of a cylindrical condensate section of finite length.

Fig. 4. Heat transfer as a function of heat flux density at a pressure of 0.31 bar, a rotational velocity of 192 rad/sec, and amount of heat-transfer agent (water) of: 1) 12 cm^3; 2) 25 cm^3; 3) 60 cm^3; 4) 104 cm^3.

increase in its effective heat conduction. At an angular velocity of 63 rad/sec, where the loading $\eta \leqslant 5.7$ g and the centrifugal forces are not yet large, the waves arise independently of the direction of change of the angular velocity of the heat pipe. For $\omega > 116$ rad/sec ($\eta > 19q$) the centrifugal acceleration forces overwhelm the gravity force and no wave arise. In the intermediate region (63 rad/sec $< \omega <$ 116 rad/sec) the liquid mass inertia forces govern the regime of motion and one observes a tendency for the liquid to keep the state of motion from which it came to the hysteresis region.

When the rotational velocity is reduced below the hysteresis range one observes an increase in heat-transfer intensity, which can be explained by a gradual decrease of the average layer thickness along the perimeter, resulting from accumulation of condensate in the lower half of the pipe.

When acetone is used as the heat-transfer agent the heat-transfer hysteresis appears in the angular velocity range 56-83 rad/sec (loading of 4.4-9.7 g). Here for rotational velocities close to 56 and 83 rad/sec, considerable oscillations were observed in the acetone vapor temperature (up to 3-4°C), with an oscillation period of 5-10 sec. The general heat-transfer level was very low (about $3 \cdot 10^2$ W/m^2).

The phenomenon of hysteresis has been observed earlier in drying drums during condensation of water vapor [8]. The hysteresis occurred in the range of loading 6.5-11.3 g, which corresponds roughly with our data. The author of [8] explains the jump in heat transfer by breakdown of the annular layer of condensate.

The wave formation at low rotational velocities, observed in the condensate section, evidently occurs also in the evaporator section. However, in developed boiling, especially if it occurs in a thin layer, the contribution of convection to the heat transfer is appreciably less than in condensation, and this phenomenon shows up quite markedly. It is possible that the increase of mobility of the layer of heat-transfer agent for reduced rotational velocity is one cause of the corresponding increase of heat transfer in the evaporator section at reduced frequency of rotation (see Fig. 1).

Figure 4 shows results of the experiments to determine the influence of the amount of heat-transfer agent (water) on the intensity of heat transfer in the evaporator section in the boiling regime. The tests were conducted at a constant rotational velocity of 192 rad/sec and a constant pressure of 0.31 bar. According to calculations using the technique described in [3], the minimal required amount of heat-transfer agent for the conditions of the experiments varies from 15 cm^3 at $q = 5 \cdot 10^4$ W/m^3 to 23 cm^3 at $q = 2 \cdot 10^5$ W/m^2. The volume of the annular groove in the end face of the evaporator section, as was mentioned, was 19 cm^3. Evidently, with a charge of 12 cm^3 of water the pipe operates with a defect of heat-transfer agent, when part of the surface of the evaporator section becomes dried out and plays no part in the heat transfer. For a charge of 25 cm^3 the heat pipe operates at the optimal layer thickness in the heat-exchanger sections (the excess pours off into the groove) over the entire heat flux range. For w = 60 cm^3 in the pipe the excess heat-transfer agent and the layer thickness in the evaporator section is 1-1.5 mm. For w = 104 cm^3 this thickness reaches 2.5-3.5 mm.

It can be seen from the data presented in Fig. 4 that the maximum heat-transfer intensity occurs for the optimal layer thickness of heat-transfer agent (w = 25 cm^3). This is 25-35% less than for a deficiency of agent (w = 12 cm^3). With an excess of heat-transfer agent (w = 60 cm^3 and w = 104 cm^3) the heat-transfer intensity is 15-20% less than with the optimal charge.

An analogous relation holds for heat transfer in the surface evaporation regime. A step in the condenser section also produces an appreciable intensification of heat transfer in this regime, especially for low and moderate rotational velocities. Evidently, centrifugal heat pipes with a step and a groove (with an optimized layer) can conveniently be used in electrical machines with controlled rotational velocities.

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