INTENSIFICATION OF HEAT EXCHANGE IN THE CONDENSATION ZONE OF CENTRIFUGAL HEAT PIPES

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We give the results of the investigation of the intensification of heat exchange in the condensation zone of centrifugal heat pipes with longitudinal grooves.

When the working fluid used in a centrifugal heat pipe (CHP) is a liquid with relatively high heat of vaporization, high thermal conductivity, low viscosity, and high density (water, ammonia), the process of heat transfer is usually limited by the heat exchange on the outside of the CHP. However, in a number of cases, when the material of the hull of the CHP makes it impossible to use such liquids, use is made of working fluids whose physical properties do not provide the necessary level of heat transfer. In this case it is desirable to use devices for intensifying the heat-exchange processes in the heating and cooling zones of the CHP. Among the basic design methods of intensifying the heat exchange in the cooling zone, which makes the main contribution to the thermal resistance of the CHP, we may mention: constructing the cooling and heating zones with different diameters, making the inner surface in the form of a truncated cone, constructing ribs on the inner surface, making grooves on the inner surface, using coaxial inserts to ensure unidirectional motion of the streams of vapor and liquid, and using surface-active substances.

The authors of [1, 2] conducted an experimental investigation of the heat exchange in CHP with a cooling zone in the shape of a cylinder, a truncated cone, a cylinder with helical ribbing, and a cylinder using surface-active substances. In all the designs investigated, the diameter of the cooling zone was smaller than that of the heating zone, so that the thickness of the film of working fluid at the end of the cooling zone was a minimum and the hydrodynamics of the motion of the film in the cooling zone was independent of the flow conditions of the film in the heating zone. It was found that the highest intensity of heat exchange (2-3 times as high as in a smooth-walled cylinder and 1.5-2 times as high as in a truncated cone) is ensured by making the cooling zone in the form of a cylinder with helical ribs. The use of surfactants is effective only for a conical condenser at a low rotation rate.

In [3] we proposed a method for intensifying the heat exchange in the cooling zone of CHP that was simpler from the technological point of view: making longitudinal grooves in the cooling zone. This should lead to a considerable reduction in the thickness of the liquid film covering the condensation surface because the condensate will move not along the axis of the CHP but in the transverse direction into the longitudinal grooves and move in these grooves into the heating zone. Thus we can reduce the length of the condensation surface in the direction of the flow of liquid, the mass flow in the cross section of the film, and therefore the thickness of the film.

To estimate the effectiveness of making longitudinal grooves in the inner surface of the cooling zone of CHP, we shall consider an approximate analysis of the process of condensation of the inner surface of a rotating cylinder with and without grooves. We make assumptions analogous to those of Nusselt for laminar film condensation, since the possibility of using these for CHP was shown in [4]; the heat removal over the entire condensation surface will be considered constant and equal to the average value over the surface, $q = Q/F = \lambda(T_{ap} - T_w)/\delta = \text{const}$, and the thickness of the condensate film will be taken to be much smaller than the radius of curvature of the condensation surface: $\delta \ll R$.

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The last assumption enables us to consider the problem in a manner analogous to condensation on a plane surface normal to the mass forces. The process of condensation for both designs for CHP cooling will be described by the same equations, except for differences in the values of the length of the condensation surface of the heat flux density through it, which we shall consider later. We arrange the coordinate x along the direction of flow of the liquid, and the coordinate y perpendicular to the condensation surface; then the equations of heat transfer and condensate-film motion can be written in the following form:

$$d\left[\int_0^\delta v dy\right] = \frac{q}{r^*\rho}\, dx,$$
$$\frac{\partial P}{\partial x} = \mu \frac{\partial^2 v}{\partial y^2}, \tag{1}$$
$$\frac{\partial P}{\partial y} = -\rho\omega^2 (R-y).$$

The limit and boundary conditions will be written in the form

$$y=0 \quad v=0,$$
$$y=\delta \quad \frac{\partial v}{\partial y}=0, \quad P=P_{\mathrm{v}}, \tag{2}$$
$$x=0 \quad \frac{\partial \delta}{\partial x}=0,$$
$$0<x\leqslant L_{\mathrm{con}} \; q=\mathrm{const},$$
$$L_{\mathrm{con}}\leqslant x<L_{\mathrm{con}}+L_{\mathrm{t}} \quad q=0,$$
$$x=L_{\mathrm{con}}+L_{\mathrm{t}} \quad \delta=\delta_m.$$

Disregarding the variation in the vapor pressure along the x axis, we can represent the solution of (1) for the film thickness with the conditions (2) as

$$\delta(x) = \left[K\left(1+2\frac{L_{\mathrm{t}}}{l_{\mathrm{con}}}\right)+\delta_m^4\right]^{0.25}\left[1-\varkappa\frac{x^2}{l_{\mathrm{con}}^2}\right]^{0.25}, \tag{3}$$

where $K = 6\mu q L_{\mathrm{con}}^2/\rho^2\omega^2 R r^*$, $\varkappa = K/\left[K\left(1+2\frac{L_{\mathrm{t}}}{l_{\mathrm{con}}}\right)+\delta_m^4\right]$.

The minimum thickness δ_m for runoff at the end of the transport zone can be determined from the minimum condition for the total specific energy of the flow in the rotating cylinder [5]. Expressing the volumetric flow rate of the condensate through a unit cross section of the film as a function of the heat-flux density in the condenser, we obtain

$$\delta_m = \left(\frac{q^2 L_{\mathrm{con}}^2}{\omega^2 R \rho^2 r^{*2}}\right)^{1/3}. \tag{4}$$

The value of the average temperature head in the condensation zone is determined as follows:

$$\overline{\Delta T} = \frac{1}{L_{\mathrm{con}}}\int_0^{L_{\mathrm{con}}}\frac{q\delta}{\lambda}\, dx = \frac{q}{\lambda}\left[K\left(1+2\frac{L_{\mathrm{t}}}{L_{\mathrm{con}}}\right)+\delta_m^4\right]^{0.25} I(\varkappa), \tag{5}$$

where $I(\varkappa) = 1-\frac{1}{3\cdot 4}\varkappa-\frac{1\cdot 3}{5\cdot 4\cdot 8}\varkappa^2-\frac{1\cdot 3\cdot 7}{7\cdot 4\cdot 8\cdot 12}\varkappa^3-\frac{1\cdot 3\cdot 7\cdot 11}{9\cdot 4\cdot 8\cdot 12\cdot 16}\varkappa^4\ldots$

The value of the parameter $\varkappa$ may vary between 0 and 1, which corresponds to the values $I(\varkappa)$ 1-0.876.

The intensity of the heat exchange in the condensation zone can be represented in the following form:

$$\bar{\alpha} = \frac{q}{\overline{\Delta T}} = \lambda\left\{\left[K\left(1+2\frac{L_{\mathrm{t}}}{l_{\mathrm{con}}}\right)+\delta_m^4\right] I^4(\varkappa)\right\}^{-0.25} \tag{6}$$

or

$$\overline{\mathrm{Nu}}=\frac{\bar{\alpha}L_{\mathrm{con}}}{\lambda}=\left\{\left[\frac{6\mu q}{\omega^2R\rho^2r^*L^2_{\mathrm{con}}}\left(1+2\,\frac{L_{\mathrm{t}}}{L_{\mathrm{con}}}\right)+\left(\frac{\delta_m}{L_{\mathrm{con}}}\right)^4\right]I^4(\varkappa)\right\}^{-0.25}. \quad (7)$$

Substituting into (5) the values of q, L_{con}, and δ_m, we determine the average temperature heads arising in the transfer of the same heat flux Q in the condensers of a CHP having the form of a smooth-walled cylinder and one having the form of a cylinder with grooves.

For the smooth-walled cylinder $L_{con} = L_{con.\,cyl}$, $L_t = L_{t.\,cyl}$, $q = Q/(2\pi R_{cyl}L_{con.\,cyl})$, $\delta_m = \delta_{m\,cyl} = \left(\frac{Q^2}{4\pi^2\omega^2R^3_{cyl}\rho^2r^{*2}}\right)^{1/3}$; then

$$\overline{\Delta T}_{\mathrm{cyl}}=\frac{Q}{2\pi R_{\mathrm{cyl}}L_{\mathrm{con}}\lambda}\left[\frac{3\mu QL_{\mathrm{con.\,cyl}}}{\pi\rho^2\omega^2R^2_{\mathrm{cyl}}\,r^*}\left(1+2\,\frac{L_{\mathrm{t.cyl}}}{L_{\mathrm{con.\,cyl}}}\right)+\delta^4_{m\,\mathrm{cyl}}\right]^{0.25}I_{\mathrm{cyl}}(\varkappa). \quad (8)$$

For a condenser with n longitudinal grooves of width a,

$$L_{\mathrm{con}}=\frac{2\pi R_{\mathrm{cg}}-na}{2n},\quad L_{\mathrm{t}}=0,\quad q=\frac{Q}{(2\pi R_{\mathrm{cg}}-na)\,L_{\mathrm{con.\,cyl}}},$$

$$\delta_m=\delta_{m\,\mathrm{cg}}=[Q^2/(4n^2\omega^2R_{\mathrm{cg}}L^2_{\mathrm{con.\,cyl}}\,\rho^2r^{*2})]^{1/3},$$

$$\overline{\Delta T}_{\mathrm{cg}}=\frac{Q}{(2\pi R_{\mathrm{cg}}-na)\,\lambda L_{\mathrm{con.\,cyl}}}\left[\frac{3\mu Q\,(2\pi R_{\mathrm{cg}}-na)}{2n^2\rho^2\omega^2R_{\mathrm{cg}}\,L_{\mathrm{con.\,cyl}}\,r^*}+\delta^4_{m\,\mathrm{cg}}\right]^{0.25}I_{\mathrm{cg}}(\varkappa). \quad (9)$$

An analysis of the profiles of the thickness of the condensate film along the cooling zone, carried out according to (3), shows that when the temperature drops according to (8) and (9) are compared, the second term in brackets (δ_m) can be disregarded; then we have approximately

$$\frac{\overline{\Delta T}_{\mathrm{cyl}}}{\overline{\Delta T}_{\mathrm{cg}}}=[n^2(2\pi R_{\mathrm{cg}}-na)^3]^{0.25}\left[\frac{R_{\mathrm{cg}}}{8\pi^5R^6_{\mathrm{cyl}}}\right]^{0.25}L^{0.5}_{\mathrm{con.\,cyl}}\,\frac{I_{\mathrm{cyl}}(\varkappa)}{I_{\mathrm{cg}}(\varkappa)}. \quad (10)$$

The optimal number of grooves is determined by equating to zero the first derivative of (10): $n_{ov} = 0.8\pi R_{cg}/a$.

Figure 1 shows the variation of $\overline{\Delta T}_{cyl}/\overline{\Delta T}_{cg} = f(n)$ for CHP condensers with a diameter of 0.03 m, on whose surface grooves of length a = 0.001 m have been made, for different cooling-zone lengths. CHP with longitudinal grooves can transfer the same heat flux as cylindrical ones for a temperature drop 1/6 to 1/4 as large if the geometric parameters are the same.

For experimental verification of the intensification of the heat exchange in the cooling zone of the CHP when longitudinal grooves have been made on the inner surface, we constructed a CHP with removable cooling sections. The heat pipe was attached to centers set in two bearing supports and was set into rotation by means of an electric motor with adjustable rpm.

Hollow-cylinder condensers were constructed from copper with identical outer dimensions: diameter 0.04 m, length 0.19 m, cooling-zone length 0.095-0.14 m. The inner surface of the condenser was: 1) a cylinder 0.035 m in diameter; 2) a truncated cone with small diameter 0.03 m and angle of taper 1°; 3) a cylinder 0.03 m in diameter with six grooves 0.001 m wide on its surface at equal intervals, the depth of the grooves increasing from 0 to 0.003 m in the direction of the evaporator.

The condensers were set up, hermetically sealed, in a conical opening in the guiding flange. Jets of water were used to cool the condenser. The sprinkling tubes and the condenser section of the CHP were placed in a hermetically sealed and thermally insulated collector for the cooling liquid.

The measurement of the temperature was carried out by means of copper–constantan thermocouples with wires 0.2 mm in diameter with cotton insulation. The temperatures of the condenser wall and the evaporator were measured by means of six and three thermocouples, respectively, caulked into the wall at a distance of 1 mm from the inner surface, and the vapor temperature in the inner cavity of the CHP was measured with two thermocouples placed along the pipe axis at the centers of the evaporating and condensing sections. The cold joint of each thermocouple was placed in a rotating thermally insulated oil-filled capsule whose temperature was measured with a semiconductor thermoresistor. The copper wires of the thermo-

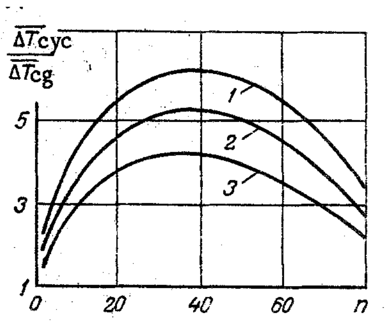


Fig. 1. Variation of $\overline{\Delta T}_{cyl}$/$\overline{\Delta T}_{cg}$ in the condenser of a CHP as a function of the number of grooves on the condensation surface: 1) L_{con} = 0.14 m; 2) 0.095 m; 3) 0.06 m.

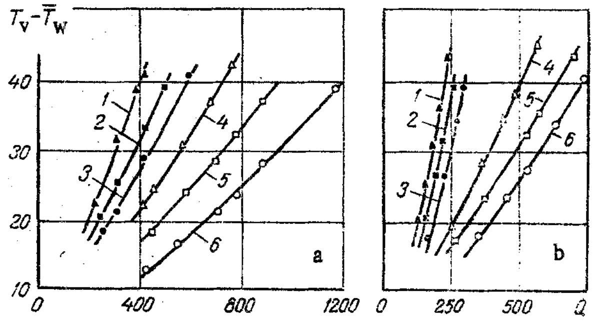


Fig. 2. Average temperature drop between the vapor and the condenser wall as a function of the heat flux: a) working fluid is acetone; b) Freon-113; 1, 2, 3) condenser No. 1, L_{con} = 0.14 m; 4, 5, 6) condenser No. 3, L_{con} = 0.14 m; rotation rate 1000 rpm, 1600 rpm, and 2000 rpm, respectively. Q in watts.

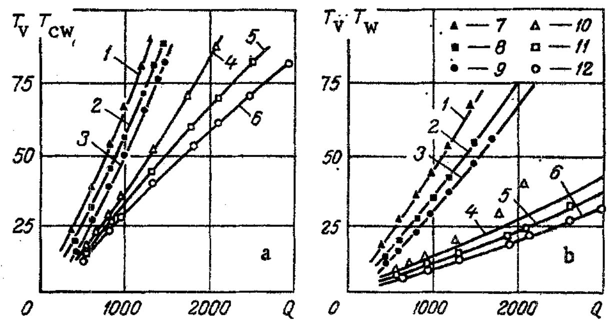
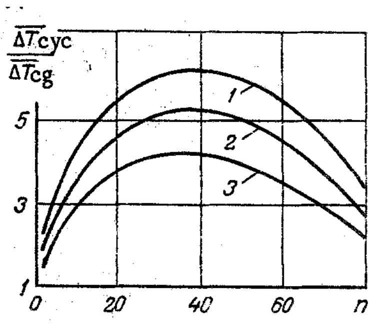


Fig. 3. Temperature drop as a function of the heat flux through the condenser surface (working fluid is distilled water, L_{con} = 0.095 m): a) between the vapor and the cooling water (1, 2, 3: condenser No. 1; 4, 5, 6: condenser No. 3); b) between the vapor and the condenser wall (condenser No. 1 — 1, 2, 3: calculation by (8), 7, 8, 9: experiment; condenser No. 3 — 4, 5, 6: calculation by (9), 10, 11, 12: experiment; rotation rate 1000 rpm, 1500 rpm, and 2000 rpm, respectively).



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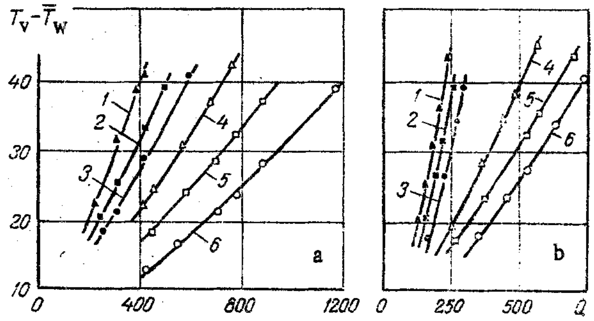

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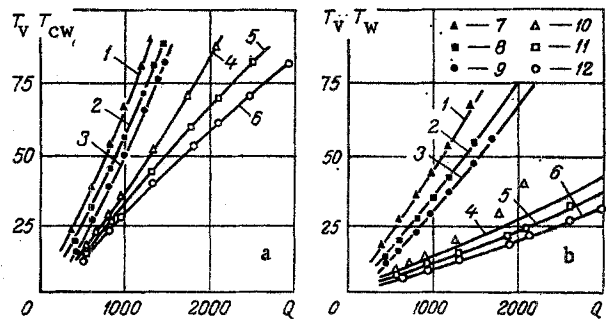

Fig. 3. Temperature drop as a function of the heat flux through the condenser surface (working fluid is distilled water, L_{con} = 0.095 m): a) between the vapor and the cooling water (1, 2, 3: condenser No. 1; 4, 5, 6: condenser No. 3); b) between the vapor and the condenser wall (condenser No. 1 — 1, 2, 3: calculation by (8), 7, 8, 9: experiment; condenser No. 3 — 4, 5, 6: calculation by (9), 10, 11, 12: experiment; rotation rate 1000 rpm, 1500 rpm, and 2000 rpm, respectively).

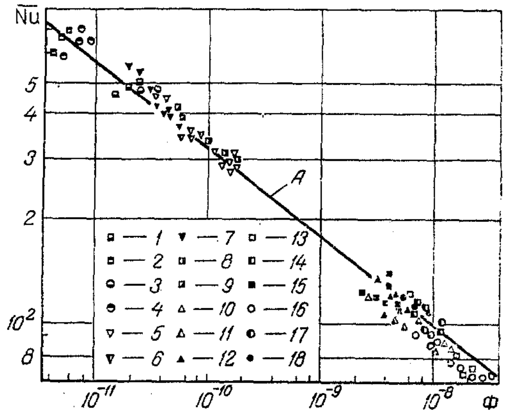

Fig. 4. Heat transfer in the condensation zone of the CHP: A) calculation by (7); condenser No. 1: if the working fluid is acetone 1) 1000 rpm; 2) 2000 rpm; if Freon-113 3) 1000 rpm; 4) 2000 rpm; condenser No. 3: if water 5) 1000 rpm; 6) 1500 rpm; 7) 2000 rpm; if acetone 8) 1000 rpm; 9) 2000 rpm; condenser No. 3: if water 10) 1000 rpm; 11) 1500 rpm; 12) 2000 rpm: if acetone 13) 1000 rpm; 14) 1500 rpm; 15) 2000 rpm; if Freon-113 16) 1000 rpm; 17) 1600 rpm; 18) 2000 rpm. $\Phi = \frac{\mu}{\rho^2 r^*} \frac{6}{\omega^2 R L^2_{con}} \left(1 + 2\frac{L_t}{L_{con}}\right) q l^{4}(\varkappa)$.

couples passing through the drive shaft were led out to slip rings and brushes. We used a zero-flow-rate pneumatic pressure device which pressed the brushes against the slip rings only while the measurements were being made, a cooling system was included to protect the brushes from wear, and we used brushes and slip rings made of specially selected materials, so that we had much lower contact emf (±0.01 mV) and much smaller oscillations of the transient resistance (±0.01 Ω) for a rotation rate of up to 3000 rpm.

The temperature of the cooling water was measured with thermocouples in the mixing chambers, situated at the inlet of the sprinkling tube and the outlet of the liquid collector.

All the thermocouples were calibrated with an accuracy of ±0.1°C. The thermocouple readings were recorded by an F-30K digital measuring system. The cooling-water flow rate was determined by means of a rotameter whose maximum error was ±1%. The rate of rotation was measured by a TM-2P magnetic-induction tachometer with an error of no more than 1%, and also independently measured by a Ch3-33 electronic-count frequency meter with a photoelectric converter for the rotation rate.

Before carrying out the investigation, we subjected the inner surface of the CHP to chemical cleaning followed by washing with distilled water and ethyl alcohol in order to wet the surface with the coolant. The heat pipe was evacuated to a pressure of $5 \cdot 10^{-2}$ mm Hg, and its hermetic seal was checked. As the working fluid, we used acetone, Freon-113, and distilled water, all of which were degassed and fed into the pipe in batches of 50 ml. The thermocouple readings, the flow rate of the cooling liquid, and the rate of rotation were recorded when the heat pipe was in a stationary regime of operation. The heat flux in the condensing section was determined on the basis of the heat balance of the cooling water, with a correction for the thermal effects of friction in the packings and the viscous dissipation of energy in the cooling water during rotation. Each condenser section was investigated while rotating at rates of 1000 to 2000 rpm. The temperature of the cooling water was kept at 10 ± 1°C.

Our experimental investigation of heat exchange in CHP with different condenser sections showed that an increase in the frequency of rotation can lead to a notable intensification of the heat exchange for all condenser designs and all working fluids; this was due to the increase in the centrifugal force and the corresponding reduction in the thickness of the liquid film on the condensing surface.

In Fig. 2 we show the results of test conducted on condensers No. 1 and No. 3 for a cooling-zone length of 0.14 m with acetone and Freon-113 as the working fluids. By varying the intensity of the heat exchange with the outer side of the condenser, the vapor saturation temperature is kept at 62 ± 1°C. In Fig. 3 we show the results obtained by testing condensers No. 1 and No. 3 with a cooling length of 0.095 m using distilled water as the coolant at constant conditions of heat exchange on the outer surface of the cooling zone.

The condenser with longitudinal grooves ensures the transfer of heat at a temperature drop that is less by a factor of 2.5-3 than the cylindrical case, which corresponds to calculation by (10), while its condensation surface is 20% smaller.

In Fig. 3b we compare the results of experiments with the calculation carried out by formulas (8) and (9). Except for curve 4, we have satisfactory (±10%) agreement. The observed discrepancy between the experimental values of the calculated values in the range of high heat-flux density is evidently due to the fact that for this regime the grooves do not suffice to carry the condensate away from the cooling zone, and they are completely flooded, which causes a reduction in the effectiveness of the heat transfer.

To generalize the experimental results on heat exchange in the cooling zone of cylindrical condensers and condensers with longitudinal grooves, we used formula (7); the results are shown in Fig. 4 for all the working fluids. In the investigated range of working parameters of the CHP we observe a satisfactory (±20%) agreement between the calculated and the experimental data.

The condenser with longitudinal grooves was found to be less sensitive to the inclination of the CHP in the direction of the cooling zone. For an inclination angle of 6° the temperature drop in the cylindrical condenser increased by 10-20°C, depending on the rotation rate, while in the condenser with longitudinal grooves it increased by 1-2°C.

The condenser shaped like a truncated cone ensured heat transfer at temperature drops 1.5-2.5 as large as the condenser with longitudinal grooves and smaller by a factor of 1.2-1.5 than the cylindrical condenser.

The experimental investigation we carried out confirmed the effectiveness of using longitudinal grooves for intensifying the heat exchange in the condensation zone of centrifugal heat pipes with longitudinal grooves is very promising.

NOTATION

x, y, coordinates; P, pressure; T, temperature: ρ, density; μ, dynamic viscosity coefficient; λ, thermal conductivity; r^*, latent heat of vaporization; ω, rotation rate; v, liquid flow velocity; Q, heat flux; q, heat-flux density; δ, thickness of the condensate film; R, radius; L, length; α, heat-transfer coefficient; $Nu = \alpha L/\lambda$, Nusselt number. Subscripts: con, condenser; t, transport zone; cyl, cylinder; cg, cylinder with grooves.

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