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STUDY OF A HEAT TRANSFER PROCESS IN THE CONDENSATION ZONE OF ROTATING HEAT PIPES*

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Abstract—The experimental results are given for heat transfer in the rotary heat pipe condenser. High efficiency of longitudinally grooved condensers is demonstrated. Relationships are suggested for calculating heat transfer in the condensers of centrifugal heat pipes.

NOMENCLATURE

x, y	coordinates
F	condensation surface, m^2
Q	heat flux, W
q	heat flux density, W/m^2
T	temperature, C
p	pressure, Pa
δ	film thickness, m
v	liquid flow velocity, m/s
ρ	liquid density, kg/m^3
λ	thermal conductivity of liquid, W/m . K
r^*	evaporation heat, J/kg
μ	dynamic viscosity, m . s/m^2
ω	frequency of rotation, 1/s
R	radius, m
L	length
n	number of grooves
a	groove width, m
Indices	
k	condenser
T	transport zone
c	cylinder
GC	grooved cylinder.

INTRODUCTION

THE CENTRIFUGAL heat pipes (CHP) are applied for cooling rotating parts of different devices such as electric motors, bearings, fan shafts, etc. Of late, the CHP application in heat exchangers for heat recovery systems is especially urgent. This will be the subject of a later paper in this Journal.

The CHP efficiency depends on the heat transfer in the evaporator and condenser. With working fluid boiling in the evaporator, the dominating contribution to the CHP thermal resistance is made by the condenser with heat transfer by conduction through the condensate film. In the simplest CHP, the condenser is a smooth-wall cylinder which does not provide the required heat transfer in many cases. Heat transfer in the condenser may be enhanced by a small thickness of the fluid film covering the condenser surface. The main design facilities to enhance heat transfer in the CHP condenser imply different diameters of the condenser and evaporator, the internal surface as a truncated cone, the internal surface finning, the grooving of the internal surface, the setting of coaxial inserts providing co-current vapour and fluid flows.

* First presented at the International School Seminar, Dubrovink, 1982.

The authors [1, 2] have experimentally studied heat transfer in CHP condensers having internal surfaces of different geometry including a smooth-wall cylinder, a truncated one and a cylinder with a screw-finned internal surface.

In all the CHP designs tested, the condenser diameter was less than that of the evaporator which ensured a minimum thickness of a film at the condenser and the film flow dynamics in the condenser independent of the film flow in the evaporator. The study has demonstrated the internal wall filling of the condenser to be most effective. This condenser design provided a 2–3-fold temperature drop between the vapour and cooling water as compared to the smooth-wall cylinder. However, the implementation of such a geometry is laborious.

A simpler way of heat transfer enhancement in the CHP condenser is suggested in [3]. It implies longitudinal grooves, whose depth increases close to the evaporator, over the internal condenser surface. This seems to considerably decrease the fluid film thickness over the condenser surface. The condensate in this case falls down into longitudinal grooves rather than moving along the heat pipe. The condensate flows to the evaporator along these grooves, thus decreasing the film path and mass fluid flow rate and, hence, the film thickness in the film cross-section. The increase of the groove depth in the evaporator direction will provide intensive removal of the working fluid from the condenser and decrease the effect of the heat pipe space orientation on heat transfer.

THEORETICAL ANALYSIS

In [4] and [5], the analysis of the operating conditions on heat transfer in the CHP condenser has shown that in the majority of cases the laminar film condensation is realized with a laminar vapour flow whose interaction with a condensate film may be neglected. Therefore, consider approximately the condensation on the internal rotating cylinder surface to estimate the efficiency of grooves on the internal condenser surface. Take the approximations similar to those in Nusselt's condensation analysis [6]. Heat transfer over the entire condensation surface is assumed constant and equal to the mean value over the surface:

$$q = \frac{Q}{F} = \frac{\lambda \Delta T}{\delta} = \text{const}$$

where Q is the heat flux, F the condensation surface, λ the thermal conductivity of the fluid, δ the film thickness, T_v the vapour temperature, T_w the wall temperature, $\Delta T = (T_v - T_w)$.

The condensate film thickness is taken to be less than the radius of the condensation surface curvature, i.e. $\delta \ll R$. Due to the latter assumption, the problem may be considered as in the case of condensation on a flat surface normal to mass forces. The assumption of a constant heat flux across the condensation surface is realized in many cases when heat transfer from the external condenser side is less intensive than from the internal side.

Set the co-ordinate x along the fluid film flow and the co-ordinate y normal to the condensate surface (Fig. 1). Then, the equations for heat transfer and condensate film flow may be written as:

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

Here ω is the rotation speed, μ, ρ, r^* are the viscosity, density and evaporation heat, respectively.

The boundary-value conditions are:

$$
\begin{aligned}
&x = 0 && \frac{\partial \delta}{\partial x} = 0, \\
&0 < x < L_K && q = \text{const}, \\
&L_K < x < L_K + L_T && q = 0, \qquad\qquad (2) \\
&x = L_K + L_T && \delta = \delta_{\min}, \\
&y = 0 && v = 0, \\
&y = \delta v \frac{\partial v}{\partial y} = 0, && p = p_v,
\end{aligned}
$$

where L_K is the condenser length and L_T the transport zone length.

Neglecting a vapour pressure variation along the axis x gives solution (1) for the film thickness under conditions (2) in the form:

$$\delta(x) = \left[K\left(1 + 2\frac{L_T}{L_K}\right) + \delta^4_{\min}\right]^{0.25}\left[1 - \kappa\frac{x^2}{L^2_K}\right]^{0.25} \tag{3}$$

where

$$K = \frac{6\mu q L_K^2}{\rho^2\omega^2 R r^*} \quad \text{and} \quad \kappa = \frac{K}{K(1 + 2L_T/L_K) + \delta^4_{\min}}.$$

The minimum thickness, $\delta_{\min}$, of the condensate film falling down at the end of the transport zone may be found from the condition of the minimum total specific energy [7] by expressing the volumetric condensate flow rate per unit cross-sectional length in terms of the heat flux density in the condenser:

$$\delta_{\min} = \left(\frac{q^2 L_K^2}{\omega^2 R \rho^2 r^{*2}}\right)^{1/3}. \tag{4}$$

The mean temperature drop in the condenser:

$$\overline{\Delta T} = \frac{1}{L_K}\int_0^{L_K}\frac{q\delta}{\lambda}\,\mathrm{d}x = \frac{q}{\lambda}\left[K\left(1 + 2\frac{L_T}{L_K}\right) + \delta^4_{\min}\right]^{0.25} I(\kappa) \tag{5}$$

where

$$I(\kappa) = \left(1 - \frac{1}{3.4}\kappa - \frac{1.3}{5.4.8}\kappa^2 - \frac{1.3.7}{7.4.8.12}\kappa^3 - \frac{1.3.7.11}{9.4.8.12.16}\kappa^4 - \dots\right).$$

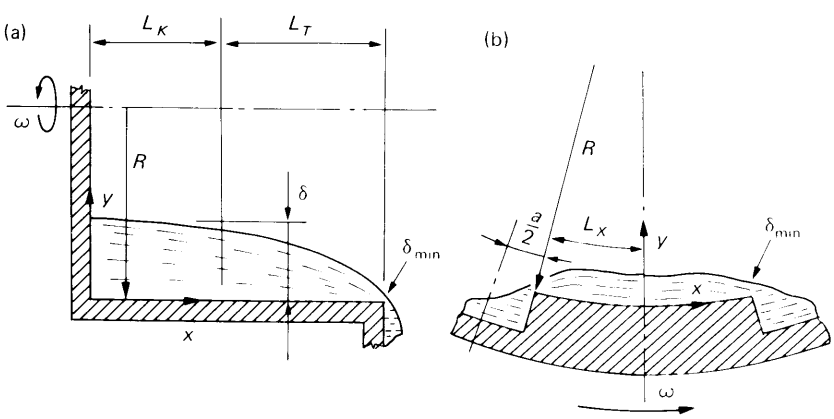

Fig. 1. To the problem of condensation on the internal rotating cylinder side: (a) cylinder; (b) longitudinally grooved cylinder.

The parameter κ may range between 0 and 1, this corresponding to $I(\kappa)$ between 1 and 0.876.

Heat transfer along the condenser may be presented as:

$$\bar{\alpha} = \frac{q}{\overline{\Delta T}} = \lambda\left\{\left[K\left(1 + 2\frac{L_T}{L_K}\right) + \delta^4_{\min}\right]I^4(\kappa)\right\}^{-0.25} \tag{6}$$

or

$$\overline{Nu} = \frac{\bar{\alpha}L_K}{\lambda} = \left\{\left[\frac{6\mu q}{\omega^2 R\rho^2 r^* L_K^2}\left(1 + 2\frac{L_T}{L_K}\right) + \left(\frac{\delta_{\min}}{L_K}\right)^4\right]I^4(\kappa)\right\}^{-0.25}. \tag{7}$$

From expression (5), for comparing heat transfer efficiency in condensers being smooth-walled and longitudinally-grooved cylinders of the same length, find the temperature drop between the vapour and the condenser wall at equal heat flux transfer. The smooth-wall cylinder parameters (Fig. 1a) required for calculation are:

$$L_K = L_{KC}, \quad L_T = L_{TC}, \quad q = \frac{Q}{2\pi R_C L_{KC}}, \quad \delta_{\min,C} = \left[\frac{Q^2}{4\pi^2\omega^2 R_C^3 \rho^2 r^{*2}}\right]$$

where L_{KC} is the length of the cylindrical condenser, L_{TC} the length of the transport zone of the cylinder, R_C the cylinder radius, $\delta_{\min,C}$ the minimum film thickness in the cylindrical condenser. Then,

$$\overline{\Delta T}_C = \frac{Q}{2\pi R_C L_{KC}\lambda}\left[\frac{3\mu Q L_{KC}}{\pi\rho^2\omega^2 R_C^2 r^*}\left(1 + 2\frac{L_{TC}}{L_{KC}}\right) + \delta^4_{\min,C}\right]I_C(\kappa) \tag{8}$$

where $\overline{\Delta T}_C$ is the mean temperature drop in the cylindrical condenser, $I_C(\kappa)$ the value of the function for the cylindrical condenser.

For the longitudinally grooved condenser (Fig. 1b)

$$L_K = \left(\frac{\pi R_{GC}}{n} - \frac{a}{2}\right), \quad L_T = 0, \quad q = \frac{Q}{(2\pi R_{GC} - na)L_K},$$

$$\delta_{\min,GC} = \left[\frac{Q^2}{4n^2\omega^2 R_{GC} L_{KC}\rho^2 r^{*2}}\right]^{1/3},$$

$$\overline{\Delta T} = \frac{Q}{2n\left(\frac{\pi R_{GC}}{n} - \frac{a}{2}\right)L_{KC}\lambda}\left[\frac{3\mu Q\left(\frac{\pi R_{GC}}{n} - \frac{a}{2}\right)}{n\rho^2\omega^2 R_{GC} r^* L_{KC}} + \delta^4_{\min,GC}\right]I_{GC}(\kappa) \tag{9}$$

where a is the groove width, n the number of grooves, R_{GC} the radius of the grooved cylinder, $\delta_{\min,GC}$ the minimum film thickness in the cylindrical grooved condenser, $I_{GC}(\kappa)$ the value of the function for the cylindrical grooved condenser.

The analysis of the condensate film profiles along the condenser using expression (5) shows that the second term in comparison of temperature drops in expressions (8) and (9) may be neglected. Then

$$\frac{\overline{\Delta T}_C}{\overline{\Delta T}_{GC}} \approx \frac{2\pi R_{GC} - na}{2\pi R_C}\left[\frac{2n^2 R_{GC} L_K^2}{\pi R_C^2(2\pi R_{GC} - na)}\right]^{0.25}\frac{I_C(\kappa)}{I_{GC}(\kappa)}.$$

EXPERIMENTAL WORK

For experimental verification of the heat transfer enhancement in the longitudinally grooved CHP condenser, a heat pipe was assembled, whose longitudinal section is presented in Fig. 2.

The heat pipe, fixed at the centres settled in two bearing supports (Fig. 2), was driven into rotation by an electric motor with controlled revolutions. The [illegible]

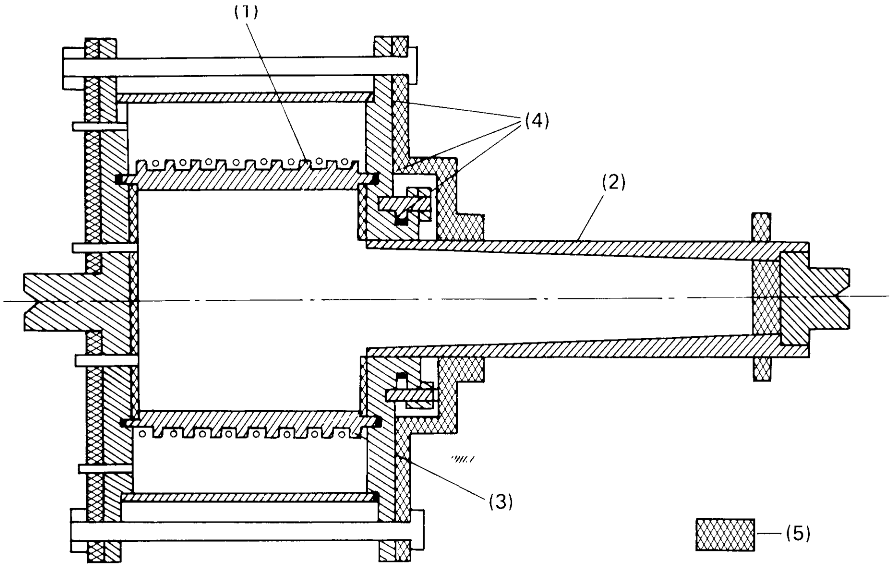

Fig. 2. Longitudinal view of the experimental CHP: (1) evaporator; (2) condenser; (3) heater; (4) vacuum packings; (5) thermal insulation.

hollow copper cylinder 71 mm in i.d., 92 mm in o.d. and 88 mm long. On the external cylinder surface, a screw groove is sharpened 5 mm deep for a heater of chromel 0.5 mm wire with glass fibre insulation to be set. The external surface is covered with 2 mm thick glass fibre. Power is supplied to the heater via graphite brushes from copper rings on the directing centres. The evaporator is set in the grooves sharpened in the directing flanges with vacuum eflon-4 packings. The hollow cylinder sealed coaxially with the evaporator serves to decrease heat fluxes. Air is evacuated from the annular cavity formed.

The condensers are made of copper as hollow cylinders with different internal geometries. The major geometric condenser parameters are cited in Table 1.

The outer diameter of all versions was the same to provide equal heat transfer conditions outside the condenser. The condensation area of the grooved cylinders,

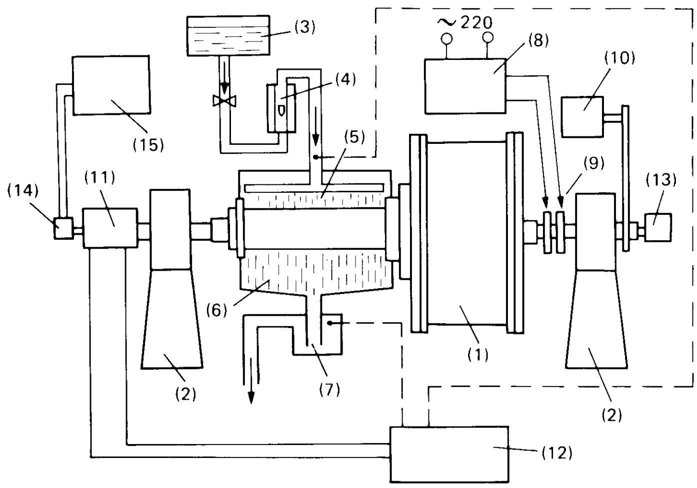

Fig. 3. Experimental set-up: (1) heat pipe; (2) bearing support; (3) constant-level tank; (4) rotameter; (5) sprayer; (6) cooling water tank; (7) mixing chamber; (8) controlled voltage source; (9) collector rings; (10) electric motor; (11) current collector; (12) digital meter system; (13) magnetic tachometer; (14) photoelectric rotation speed converter; (15) electronic frequency meter.

Table 1

Sample No.	Internal surface geometry	Condenser length (m)	Heat-insulated zone length (m)	Inner diameter (m)	Outer diameter (m)
1	Smooth-wall cylinder	0.14	0.065	0.04	0.035
2	Cylinder with 6 longitudinal grooves, 1 mm wide	0.14	0.065	0.04	0.03
3	Smooth-wall cylinder	0.093	0.076	0.04	0.035
4	Cylinder with 6 longitudinal grooves, 1 mm wide	0.093	0.076	0.04	0.03

assuming no condensation in the grooves, amounted to 80% of the smooth-wall cylinder area. But the calculation using expression (10) has manifested a 2.8 fold temperature drop of the grooved condenser compared to that of the smooth-wall cylinder, equal heat fluxes being transferred at equal saturation temperatures.

The condensers are sealed into a conic orifice in the directing flange. They are cooled by water spraying. The sprayer-tube and condenser section are placed in a sealed heat-insulated cooling water tank.

Temperature is measured by copper-constantan thermocouples with 0.2 mm wires with cotton insulation. The condenser and evaporator wall temperatures are measured by six and three thermocouples, respectively, sealed in the wall 1 mm far from the internal surface. The vapour temperature inside the CHP is measured by two thermocouples along the pipe axis in the evaporator and condenser centres. Cold thermocouple junctions are set in a rotating heat-insulated capsule filled with oil whose temperature is measured by a semiconductor thermoresistor. Copper thermocouple wires are led to a brush collector via a power shaft. The discharge-free pneumatic pressing device (that presses brushes only for the time of measuring), cooling system for erosion pump brushes, brushes and rings of special materials have provided an essential decrease in the contact e.m.f. of $+0.01$ mv and contact resistance fluctuations of ± 0.01 Ohm at 3000 rev/min.

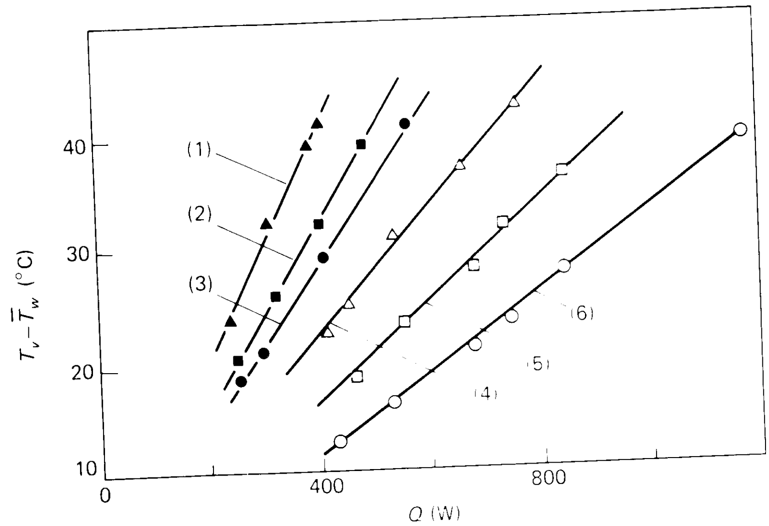

Fig. 4. Mean temperature drop between the vapour and the condenser wall vs heat flux: (1), (2), (3), condenser No. 1; (4), (5), (6), condenser No. 2; rotation speed, 1000, 1600, 2000 rev/min; acetone is the working fluid; vapour temperature amounts to 62 ± 1 C.

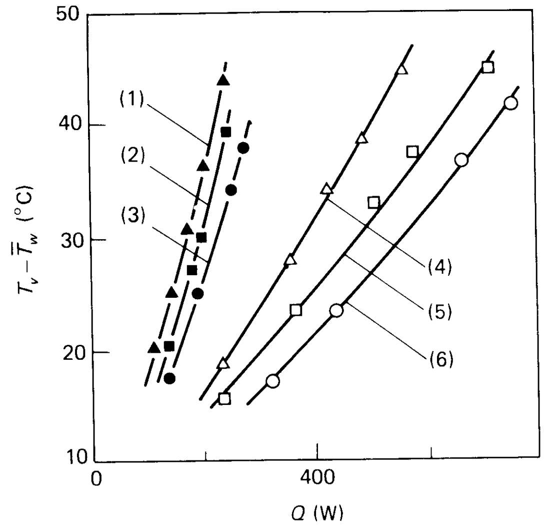

Fig. 5. Mean temperature drop between the vapour and the condenser wall vs heat flux: (1), (2), (3), condenser No. 1; (4), (5), (6), condenser No. 2; speed of rotation, 1300, 1600, 2000 rev/min. Freon-113 is the working fluid; vapour temperature amounts to 62 ± 1 C.

The cooling water temperature is measured with the thermocouples in mixing chambers at the inlet of the sprayer and outlet of the liquid collector.

All the thermocouples are calibrated within ±0.3 C. The thermocouple readings are registered by the digital measuring system Φ − 30K. The rotameter with the highest error ±1% is used to measure the cooling water flow rate. The speed of rotation is measured by the magnetic-induction tachometer TM-211 type within no more than 1% and by the electronic digital frequency meter 43–33 with the photoelectric rotation speed converter.

Before the experiments, the internal CHP surface was chemically treated and then washed with distilled water and ethyl alcohol to provide better wetting of the surface with the heat transfer agent. The heat pipe was vacuum-treated up to 5.10^{-2} mm Hg and checked for hermetic sealing. 50 ml of degased acetone, Freon-113 and distilled water used as working fluids were poured into the pipe. The thermocouple readings, cooling liquid flow and speed of rotation were recorded under steady heat pipe operating conditions. The heat flux in the condenser was found from the heat balance for the cooling water, corrected with allowance for the thermal friction effects in the

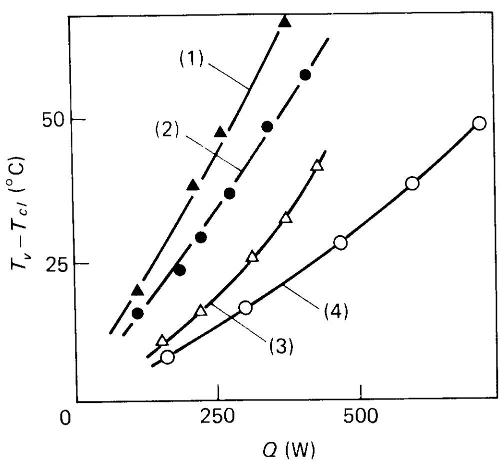

Fig. 6. Temperature drop between the vapour and the cooling liquid vs heat flux: (1), (2), condenser No. 3; (3), (4), condenser No. 4; speed of rotation, 1000, 2000 rev/min; acetone is the working fluid.

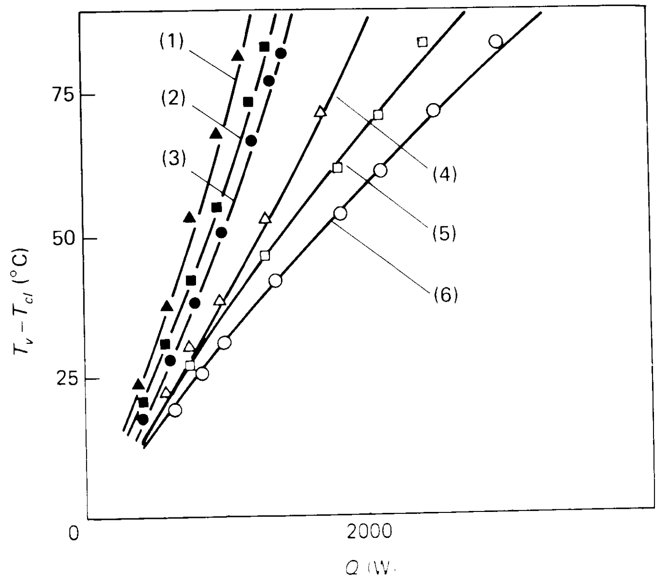

Fig. 7. Temperature drop between the vapour and the cooling liquid vs heat flux: (1), (2), (3), condenser No. 3; (4), (5), (6), condenser No. 4; speed of rotation, 1000, 1500, 2000 rev/min; water is the working fluid.

packings and viscous energy dissipation in the cooling water in rotation. The experiments were made at CHP rotation speeds of 1000 and 2000 rev/min. The cooling water temperature was kept within ± 1°C.

Figures 4 and 5 present the results of testing condensers numbers 1 and 2 with acetone and Freon-113 as working fluids. The heat transfer intensity outside the condenser being varied and the saturation temperature kept constant.

In Figs. 6 and 7 the results are given for condensers numbers 3 and 4 with acetone and distilled water as working fluids and constant heat transfer conditions outside the condenser. The heat transfer characteristics are seen to be improved with increasing speed of rotation and are 2–3 times higher for longitudinally grooved condensers than for smooth-wall ones, their condensation surface being 20% less.

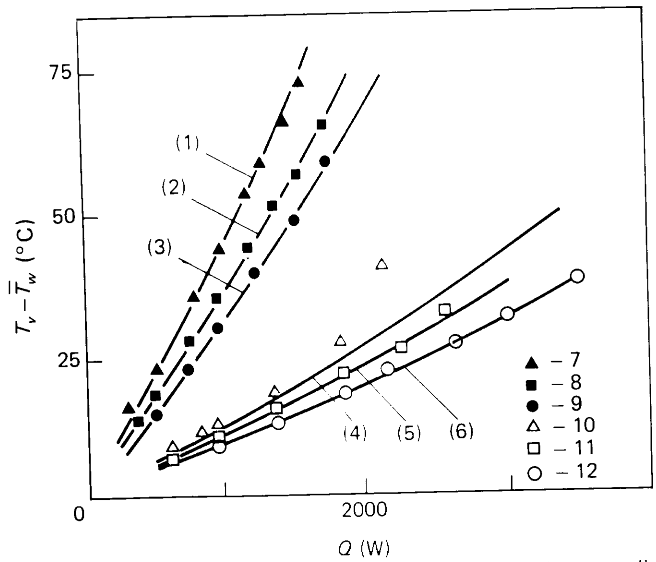

Fig. 8. Mean temperature drop between the vapour and the condenser wall vs heat flux: (1), (2), (3), condenser No. 3, calculation by (8); (7), (8), (9), experimental values; (4), (5), (6), condenser No. 4, calculation by (9); (10), (11), (12), experimental values; speed of rotation, 1000, 1500 and 2000 rev/min; water is the working fluid.

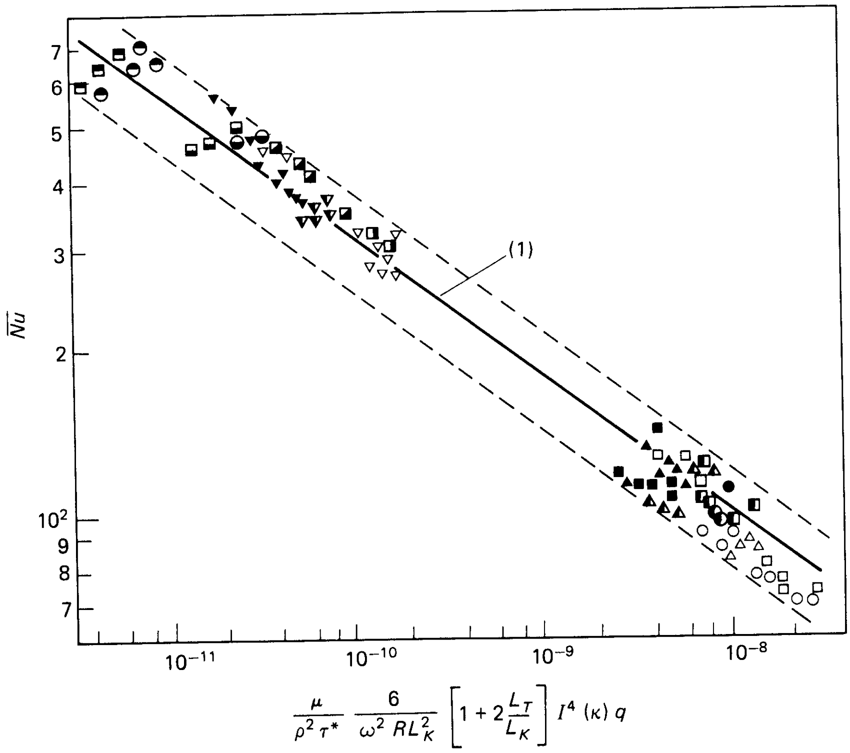

Fig. 9. Heat transfer in the heat pipe condenser: 1, calculation by (7). Condenser No. 1: acetone, ⬓, 1000 rev/min, ⬒, 2000 rev/min, Freon-113, ◒, 1000 rev/min, ◓, 2000 rev/min. Condenser No. 3: water, ▽, 1000 rev/min, ▼, 1500 rev/min, ▼, 2000 rev/min; acetone, ◨, 1000 rev/min, ◧, 1500 rev/min, ◪ 2000 rev/min. Condensers Nos. 2 and 4: water, △, 1000 rev/min, ▲, 1500 rev/min, ▲, 2000 rev/min; Freon-113, ○, 1000 rev/min, ◐, 1600 rev/min, ●, 2000; acetone, 1000 rev/min, ◧, 1500 rev/min, ■, 2000 rev/min.

Figure 8 gives the comparison of experiment and calculation by relations (8) and (9). All the curves but curve 4 show satisfactory ($\pm 5\%$) agreement. The deviation of the experimental values from the calculated ones for large heat fluxes is presumably due to the situation that under these operating conditions the grooves fail to remove fluid from the condenser and are completely flooded, thus decreasing heat transfer efficiency.

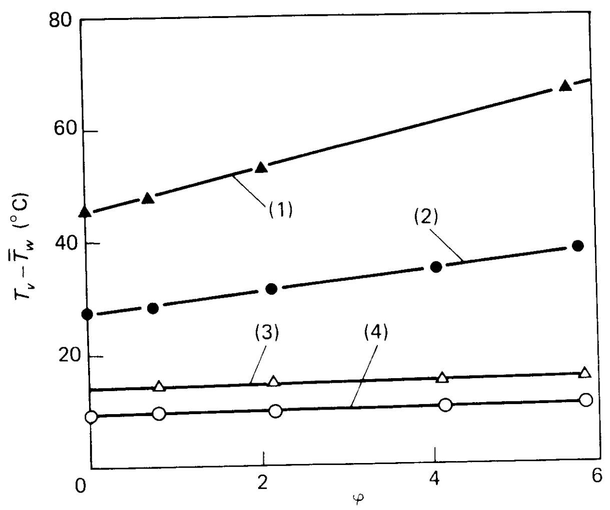

Fig. 10. Mean temperature drop between the vapour and the condenser wall vs heat pipe inclination angle: 1, 2, condenser No. 3; 4, 5, condenser No. 4; speed of rotation, 1000 and 2000 rev/min; working fluid is water; heat flux, 1000 W.

The experimental results on heat transfer in the condenser are correlated by relation (7) and presented in Fig. 9 for all condenser designs and working fluids studied. Heat transfer in the condenser for the test range of CHP operating parameters is satisfactorily ($\pm 20\%$) described by relation (7).

Figure 10 presents the experimental results on the effect of the heat pipe orientation in the space upon heat transfer characteristics. The longitudinally grooved condenser proved to be less sensitive to the inclination of the rotation axis to the condenser. Thus, at the inclination angle of 6° the temperature drop is increased by 10–20°C in the cylindrical condenser and by 1–2°C in the longitudinally grooved condenser depending on the speed of rotation.

CONCLUSIONS

The experiments have proved the efficiency of longitudinal grooves for enhancing heat transfer in the CHP condensers and made it possible to recommend the obtained analytical expressions for heat transfer calculation.

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