THEORETICAL AND EXPERIMENTAL STUDY OF CAPILLARY CONSTRAINTS ON HEAT TRANSFER IN HIGH-TEMPERATURE HEAT PIPES

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The results are presented of experimental and theoretical studies concerning capillary constraints on heat transfer in high-temperature heat pipes, also of the effect of the compressibility of vapor on the calculated power transfer and, accordingly, the limits of validity are established for theoretical methods not taking into account the compressibility of the vapor stream.

Experimental Method. Capillary constraints on power were measured in experiments with a horizontal cylindrical heat pipe operating with sodium and a composite wick of the screen type, an annular 0.5-mm-wide clearance left for passage of the liquid. The diameter of the vapor channel was 14 mm and the diameter of wick holes within the heating zone was 450 μm.

The heat pipe was heated with condensing sodium vapor over a 100-mm-long segment so as to ensure large thermal fluxes within the heating zone and to make a study of the limiting characteristics possible without the danger of overheating the body. The effect of gravitational forces on the performance was eliminated by inclining the pipe slightly upward for the duration of the experiments. The difference between the altitudes of both ends was 14-16 mm, the position of the pipe at various temperatures being checked with a plumb and maintained accurate within ± 2 mm. According to the authors' estimates, the maximum error of measurement of the pipe power did not exceed ± 7%. With the pipe departing from the horizontal position within those limits, the maximum power varied within ± 1.5%. The maximum power transfer in these experiments was determined during a gradual increase of the outgoing heat until the wick became dry within the heating zone, this condition being indicated by an abrupt drop of the pipe temperature within the adiabatic zone with a simultaneous decrease of the outgoing power and rise of the temperature at the entrance to the evaporation zone. After the wick had become dry and prior to the next power measurement, it was properly refilled with the heat carrier fluid.

Experimental Results and Their Comparison with Theoretical Calculations. The measured maximum power of this sodium heat pipe is shown in Fig. 1 as a function of the vapor temperature at the entrance to the evaporation zone. It is compared here with the calculated acoustic limit, also with calculated capillary constraints on the power in the case of a pipe operating with a compound wick (Q_{comp}) or a simple wick (Q_{clear}). The acoustic limit of power was calculated with compressibility and friction in the vapor stream taken into account [1-3].

The capillary constraints on power were calculated on the basis of the pressure balance along the liquid—vapor channel between the "wet" point and the "dry" point in the pipe

$$\int_{l_{\mathrm{eff}}} \frac{\partial P_{\mathrm{V}}}{\partial l}\, dl + \int_{l_{\mathrm{eff}}} \frac{\partial P_{\mathrm{L}}}{\partial l}\, dl + \Delta P_{\mathrm{ph}} \pm \Delta P_{\mathrm{m}} = (\Delta P_{\sigma})_{\max}. \tag{1}$$

The pressure drop between vapor phase and liquid phase is maximum at the "dry" point and zero (or minimum) at the "wet" point.

The pressure drop in the liquid was calculated according to the Darcy law, which for a compound wick with a coaxial clearance can be expressed as

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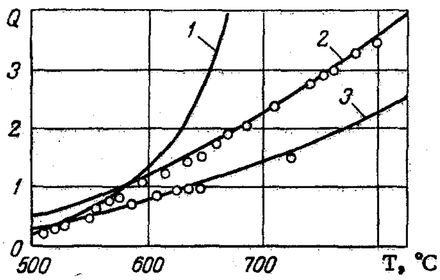

Fig. 1

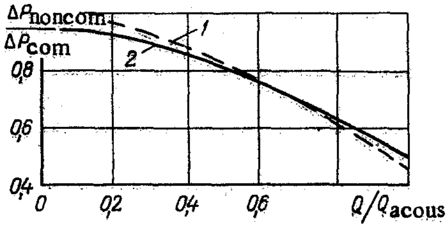

Fig. 2

Fig. 1. Temperature dependence of the maximum power Q (kW/cm²) of a sodium heat pipe (curves represent calculations, dots represent measurements): 1) Q_{acous}; 2) Q_{comp}; 3) Q_{clear}.

Fig. 2. Comparison of the pressure drops in the vapor with and without the compressibility of vapor taken into account for evaluating the capillary constraints on the power when: 1) d_{men} = 5 μm, 2) d_{men} = 500 μm.

$$\Delta P_L = \frac{0.389 \nu_L l_{eff}}{L d_V \delta_L^3} Q. \tag{2}$$

The pressure drop in the vapor phase is determined by two factors: acceleration of the stream (inertial component) and friction. The friction was calculated by integrating over the effective pipe length (distance between the "wet" point and the "dry" point), taking into account the variability of both the vapor stream velocity and the friction coefficient expressed in terms of the Reynolds number for a purely axial flow. The inertial component in an incompressible vapor stream was calculated according to the relations in [3]. Its contribution was taken into account only within the effective length of the pipe. The hydrostatic head during operation with the pipe inclined in the gravitational force field was calculated with the "wet" point location as the reference.

During evaporation and condensation there appears a pressure drop at the phase transition point. When the "wet" point is located at the entrance to the condensation zone, then the pressure drop at the phase transition is significant only within the evaporation zone.

The "wet" point location was determined from the relation between the inertial component in the vapor stream, on the one hand, and the pressure loss due to friction in the vapor and in the liquid, on the other. In heat pipes with the optimum with respect to heat-transfer geometry the "wet" point lies, as a rule, at the entrance to the condensation zone. As the effect of friction becomes more appreciable, this point shifts from the beginning toward the end of the condensation zone.

For calculating the capillary constraints on the power of heat pipes there has been developed a program written in ALGOL-60 adaptable to a TA-2M translator. The maximum power was calculated on the basis of the pressure balance (1) and by the iteration method, as follows. From the given input quantities the moving head in the pipe was determined, taking into account gravity and the initial value Q_{ini} of power for starting the iteration process. According to the initial value Q_{ini} the mode of vapor flow was determined, also the pressure loss in the vapor and in the liquid as well as the relation between inertia and friction effects. When

$$\Delta P''_{ine} / (\Delta P''_{fr} + \Delta P'_{fr})_{cond} \geq 1, \tag{3}$$

then the "wet" point was found at the entrance to the cooling zone of the pipe and in the pressure balance (1) were included pressure losses along the vapor–liquid channel within the evaporation zone and the adiabatic zone only, i.e.,

$$l_{eff} = l_{evap}^{eff} + l_{adiab}. \tag{4}$$

When

$$\Delta P''_{ine} / (\Delta P''_{fr} + \Delta P'_{fr})_{cond} < 1, \tag{5}$$

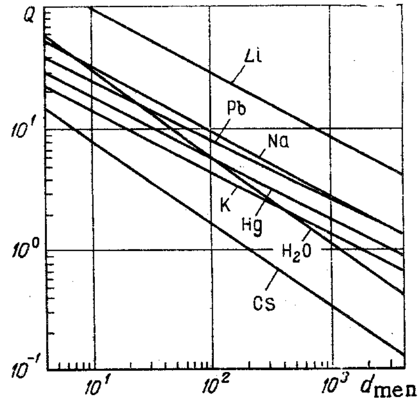

Fig. 3. Dependence of the capillary constraints on the power of heat pipes on the pore diameter d_{men} (μm) in the screen mesh of a compound wick (P_V = 1 atm abs, d_V = 10 mm, δ_L = 0.5 mm, l_{evap} = 400 mm, l_{adiab} = 200 mm, l_{cond} = 400 mm, cos θ = 1, g = 0).

for the entire condensation zone, then this relation was verified by the iteration method for successively smaller segments of the heat-transfer zone. In the limiting case the "wet" point could lie at the end of the pipe so that l_{eff} had to represent the total length of the pipe

$$l_{eff} = l_{evap}^{eff} + l_{adiab} + l_{cond}^{eff} \tag{6}$$

and the inertia effect to be completely excluded from the pressure balance (1). After the "wet" point location had been established, the pressure losses along l_{eff} were calculated and compared with the moving capillary head according to relation (1). When the capillary head was found to be larger (smaller) than the pressure losses along the heat carrier channel at a given power level, then Q_{ini} was corrected to a higher (lower) level and the calculation process repeated. The iteration process was continued until the pressure losses differed from the capillary head by an amount defining the accuracy of calculations and stipulated in the program.

According to such calculations for the experimental heat pipe, under capillary constraints on heat transfer the "wet" point is located at the entrance to the condensation zone. The main contribution to pressure losses along the vapor—liquid channel in the heat carrier comes from the inertia effect in vapor (up to 80-90%). Pressure losses due to friction in the liquid amount to 5-7% of the moving capillary head.

A comparison of experimental and theoretical data on the maximum power of the experimental heat pipe in Fig. 1 indicates a satisfactory agreement. Within the range of vapor pressure near the acoustic limit in the pipe the theoretical values for capillary constraints are higher than the experimental values. This overestimate is a result of not taking into account the compressibility of vapor in calculation of the pressure drop in the vapor. Farther away from the acoustic limit the effect of compressibility becomes less significant.

Calculations of the pressure drop in the vapor along the pipe length with and without the compressibility of vapor taken into account are compared in Fig. 2. The method of taking the compressibility of vapor into account was described in an earlier study [2]. Calculations were made for two different trends of capillary constraints, each corresponding to a different pore diameter (d_{men} = 500 and 5 μm, respectively) with the geometrical parameters of the experimental heat pipe. According to the calculations, not accounting for the compressibility of vapor will yield only approximately half the pressure losses at $Q = Q_{acous}$. This underestimate becomes smaller as the power transferred by the pipe drops farther below the acoustic limit. The pressure losses in the vapor with a predominant inertia effect are proportional to the power squared ($\Delta P''_{ine} \sim Q^2$) and, therefore, for evaluating the power of a heat pipe, if this power is $Q \leqslant 0.7Q_{acous}$, one may calculate (with an error not exceeding 10%) the pressure drop without taking into account the compressibility of vapor. In the given experimental heat pipe this condition was satisfied at temperatures above 650°C. On the basis of the pressure drops in the vapor calculated in a similar manner, it appears that in the case of comparatively short heat pipes the compressibility of vapor is negligible when $0 < Q \leqslant 0.7Q_{acous}$ and becomes significant when $Q > 0.7Q_{acous}$ so that not taking it into account then will result in an overestimate of the capillary constraints on the power as large as 30% when $Q = Q_{acous}$.

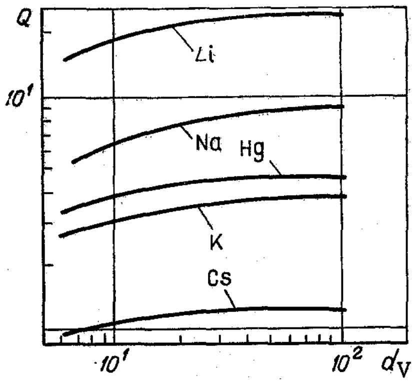

Fig. 4

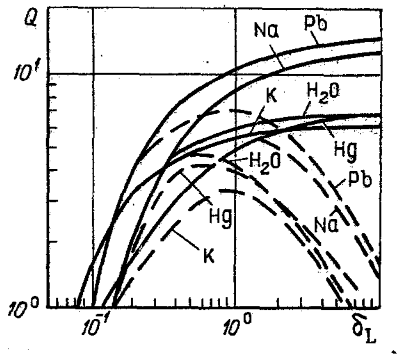

Fig. 5

Fig. 4. Dependence of the capillary constraints on the power of heat pipes on the diameter d_V (mm) of the vapor channel (l = 1 m, δ_L = 0.5 mm, d_{men} = 200 μm, cos θ = 1, P = 1 atm abs, g = 0).

Fig. 5. Dependence of the capillary constraints on the power of heat pipes on the width of the clearance δ_L (mm) for passage of the liquid (d_{men} = 160 μm, l = 1 m, d_V = 10 mm, P = 1 atm abs, cos θ = 1, g = 0).

Dependence of the Capillary Constraints on the Power of Heat Pipes on Thermophysical and Geometrical Parameters. Thermophysical properties of heat carriers (surface tension, heat of evaporation, viscosity of the liquid and of the vapor, density) influence the maximum power transferred in the system in a way which makes each heat carrying substance most effective within a certain temperature range. The problem of selecting the best heat carriers for different temperatures has already been considered in earlier studies [3, 4]. The role of the pipe geometry including the pore diameter in the screen mesh, radius of the vapor channel, and width of the clearance for passage of the liquid can be analyzed on the basis of calculations pertaining to the capillary constraints on the power and shown in Figs. 3-5. The graph in Fig. 3 depicts the dependence of the capillary constraints on the power of heat pipes with different heat carriers on the pore diameter in the screen mesh under a pressure of 1 atm in the pipe, disregarding the effect of gravity. The results of these calculations indicate that a quite intensive heat transfer can be realized by means of heat pipes. With a technologically entirely feasible pore diameter of 100-10 μm [2, 5], e.g., the specific heat transfer in sodium heat pipes can reach and exceed 10 kW/cm². At this time experiments are being performed on sodium and lithium heat pipes with a specific axial heat flow up to 15 kW/cm² [5, 6], and even this does not represent the ultimate capacity of heat pipes. In water heat pipes such levels of heat flow may not be attained because, e.g., of boiling of the heat carrier in the wick.

Changing the diameter of the vapor channel has an appreciable effect on the performance as long as this diameter d_V remains smaller than 40 mm, when the pressure lost in the moving head is due to friction and inertia effects (Fig. 4). As the diameter of the vapor channel is increased, friction losses in the vapor decrease till eventually the pressure losses along the vapor—liquid channel are determined by the inertia effect when the width of the clearance for passage of the liquid is optimum.

The effect of changing the clearance for passage of the liquid is shown in Fig. 5. In these calculations the variable quantity was the clearance width at a fixed diameter of the vapor channel. The results are presented here in the form of two relations. Solid lines depict the dependence of the capillary constraints on the power of a heat pipe on the clearance width (with the power referred to the cross section of the vapor channel) and dash lines depict the power referred to the total cross section of the vapor channel and the wick. As the clearance widens, the friction losses in the liquid decrease and the power of the heat pipe increases. This increase is limited, however, because a certain clearance width is eventually reached at which the pressure loss due to friction in the liquid becomes negligible and a further increase of the clearance width will have no effect on the power (solid lines). The power referred to the inside cross section of a pipe reaches a maximum beyond which it again decreases (dashed lines). This peaking of the power is a result of its initially increasing faster than the total inside pipe cross section as the cross section for passage of the liquid is increased. When the clearance width corresponds to the maximum transferred power, then the main loss of moving head is due to the pressure drop in the vapor. A further increase of the clearance results basically in an increase of the total pipe cross section without an increase of transferred power. Consequently, the specific axial heat flow referred to the total pipe cross section will now decrease.

NOTATION

Q, power of a heat pipe; $\partial P/\partial l$, pressure gradient; P, pressure; l, pipe length; ν, kinematic viscosity; L, heat of evaporation; d, diameter; and δ, clearance width; subscripts: V, vapor; L, liquid; "eff," effective; σ, capillary; m, mass; ph, phase; and "men," meniscus.

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