A RECIPROCATING HEAT PIPE

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Operating results are presented for heat pipes where the transport of a heat-transfer agent is caused by reciprocating motion of the pipe itself. The experimental results are generalized with the aid of a correlation.

An interesting member of the diverse family of heat pipes is the pipe where transport of a heat-transfer agent results from reciprocating motion of the pipe itself. These pipes find application in components of machines and mechanisms that experience reciprocating motion and require cooling. In this type of pipe, as in the rotary heat pipe, the heat-transfer agent is set in motion by the pipe motion. Pipe length and diameter that would be most probable for practical use were chosen for an experimental investigation. The oscillation amplitude of the heat pipe was chosen to be comparatively small, 0.075 m, so that a considerable fraction of the pipe would be occupied by a nonordered liquid flow. For reasons of manufacturing simplicity, wickless pipes, both with a noncondensible gas and without it, were chosen for the investigation. To obtain data on the hydrodynamics and to determine the quantity of the heat-transfer agent, some of the tests were carried out with a transparent unheated and heated pipes. In the experiments, the variables were the frequency of heat pipe motion (the heat pipe moved only its axis), the transmitted power, and the orientation relative to the direction of gravity. Ethyl alcohol was the working liquid.

The experimental investigation was conducted on equipment consisting of an electric motor, a crank gear, the pipe, and measurement and control equipment. The glass pipes had a length of $l = 0.14$ m and a diameter of $\varnothing = 0.013$ m. They were heated with an insulated Nichrome spiral and cooled by means of a copper coil carrying water. The pipes were filled with a heat-transfer agent to one-twelfth, one-fifth, and one-third of their volume.

Copper heat pipes of length 0.1 and 0.3 m, and diameter 0.01 and 0.016 m, with an insulated transport zone were heated by the ohmic heating and cooled with water from a constant level bath, passing through a water jacket. The temperature at the pipe surface in the evaporation, transport, and condensation zones was measured as a function of pipe length at 9-12 points by means of thermocouples and a type KSP-4 recorder. The pipes were filled with the heat-transfer agent to one-fifth and one-third of the volume. The electric power varied in the range 50-600 W, and the frequency of motion varied from 0 to 60 rps. The experimental conditions are shown in Table 1.

Observations of the liquid motion in the unheated transparent vertical pipe showed that, beginning at $n = 4$ rps, the heat-transfer agent completely irrigated the internal surface, moving over the entire pipe volume. When $n = 6$ rps was reached, the spraying in the top part of the pipe became droplike in nature. With an increase in n the drop size became steadily smaller and the number of drops increased significantly. Of the heat-transfer fillings of one-twelfth, one-fifth, and one-third of the pipe volume examined, the last two were chosen.

Visual observations of the vapor space under conditions where the pipe was heated with the lower location of the condensation zone indicated that the vapor forming with increase of power prevents motion of the drops, and this accelerates drying up of the liquid film on the wall.

Most of the tests were carried out with copper heat pipes, filled to one-fifth and one-third of their volume. All three pipes operated successfully in the range of parameters investigated. It can be seen from Fig. 1 that for a heater power of 50 W, the temperature drop between the evaporator and the condenser for all three tubes decreased sharply with increase of frequency in the neighborhood of $n = 4$ rps. A further increase in the rps up to 10 improved the heat transfer, but the relative velocity effect did not increase as rapidly. Therefore, an increase in the rps has a strong influence on the heat transfer when the number of

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TABLE 1. Experimental Conditions

Heat pipe no.	Case material	Heat pipe length, m	l_e, m	l_a, m	l_c, m	ϕ_{in}, m	δ, m	Amt. of operating liq. as a fraction of int. vol.	Position rel. to pipe axis
1	Copper	0,3	0,1	0,1	0,1	0,016	0,002	1/5, 1/3	Vertical (condenser below and above)
2	Copper	0,1	0,033	0,033	0,033	0,016	0,002	1/5	Vertical (condenser below and above)
3	Copper	0,1	0,04	0,02	0,04	0,010	0,001	1/5	Vertical (condenser below and above)
4	Glass	0,14	0,04	0,06	0,04	0,013	0,001	1/12, 1/5, 1/3	Vertical

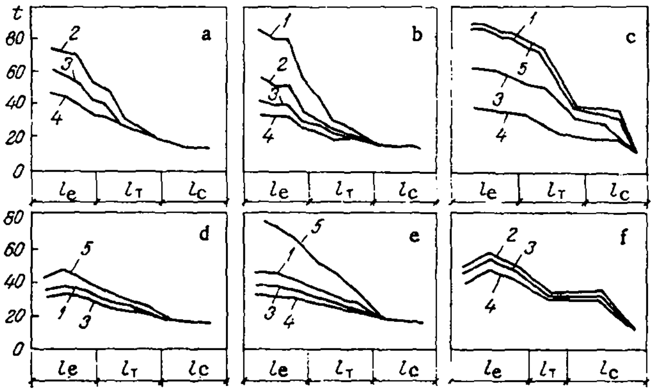

Fig. 1. Temperature distribution along a vertical heat pipe as a function of its length and diameter, the amount of heat-transfer agent, the position of the condenser, the frequency of motion and the presence and absence of noncondensible gas, for P_{el} = 50 W: 1) n = 4; 2) 5; 3) 6; 4) 10; 5) 0 rps. The fraction of the pipe filled with the heat-transfer agent and the location of the condenser were: ϕ = 0.016 m, a) 0.2, condenser below; b) 0.33, below; c) 0.2, above, with noncondensible gas; d) 0.2, above; e) 0.2, below; ϕ = 0.01 m, f) 0.2, below. The length is 0.3 m (a, b, and c) and 0.1 m (d, e, and f). a, b, and c relate to heat pipe No. 1, d and e refer to No. 2, and f refers to No. 3; t is in °C.

cycles is small. In the tests with the pipe of length 0.3 m, a significantly greater temperature drop was observed than for the pipe of length 0.1 m. The temperature drop decreased with increase of the amount of the heat-transfer agent in the pipe from one-fifth to one-third of its internal volume (Fig. 1b). For a smaller internal diameter of pipe (0.01 m) the temperature drop along the length is larger than for ϕ = 0.016 m, but its dependence on the frequency is less (Fig. 1f).

Several tests were conducted with a Pipe No. 1, fitted with a heat-transfer agent without pumping out the air. With an increase in the frequency, the negative influence of noncondensible gas on the pipe operation was considerably reduced (Fig. 1c). This may be explained by the fact that the liquid traps the noncondensible gas and returns it to the evaporation zone.

For P = 50 W and n = 4-10 rps the position of the condenser (above and below) has little influence on the operation of Pipe No. 2 when vertical (Fig. 1d and e). With increase of P to 400 W the temperature drop along the pipe increases from 30 to 70°C.

An increase in the heater power up to 400 W for Pipes Nos. 1 and 2, and up to 200 W for Pipe No. 3 (the data are not shown) does not change the nature of the heat transfer with increase of the frequency, but the absolute magnitude of the temperature drop along the pipe becomes larger (the maximum values of Δt for n = 8 rps for Pipe No. 1 is 85°C, for No. 2 it is 75°C, and for No. 3 it is 95°C).

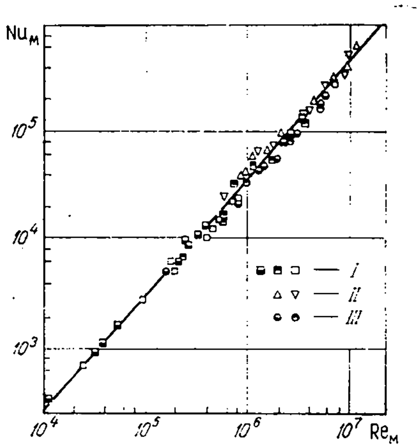

Fig. 2. Correlation of the test data ($Nu_M = 0.0138\ Re_M^{1.08}$): I) heat pipe No. 1, II) No. 2, III) No. 3.

Tests were also made with Pipe No. 1 horizontal, filled with a heat-transfer agent to one-fifth of its volume. The absolute temperatures obtained fell between the temperatures in the tests with a vertical pipe with the downward and upward positions of the condenser. The influence of frequency on heat transfer begins to show up at a smaller frequency, n = 1 rps.

The test data were correlated using the relation [1, 2]

$$Nu_M = f\,(Re_M),$$

in which

$$Nu_M = \frac{q l_*}{(t_e - t_c)\,\lambda_l} \cdot Re_M = \frac{w_b l_*}{\nu_l}, \quad w_b = \frac{q}{r\rho_b}.$$

The characteristic linear dimension was taken to be l_*, proportional to the size of the vapor bubble at the moment of its origin:

$$l_* = \frac{c_{pl}\,\rho_l\,\sigma t_{av}}{(r\rho_b)^2}, \text{ where } t_{av} = \frac{t_e + t_c}{2}.$$

The physical properties were taken at t_{av} [3]. Here $(t_e - t_c)$ is the largest temperature drop along the heat pipe between the heater and condensation zones. A correlation of the data is shown in Fig. 2. Within a scatter of 20% they correlate with the relation

$$Nu_M = 0.0138 Re_M^{1.08}, \quad q = 0.0138\,\frac{(t_e - t_c)\,\lambda_l\ Re_M^{1.08}}{l_*}.$$

This relation is valid for a film condenser in the range of parameters l = 0.1-0.3 m, $\varnothing_{in}$ = 0.01-0.016 m, P = 50-200-400 W, n = 4-10 rps. This relation was used to correlate the data on operation of Pipe No. 1 in the horizontal position for the case P = 50-200 W and n = 0-10 rps.

NOTATION

n	is the number of crankshaft rotations, rps;
l	is the heat pipe length, m;
$\varnothing$	is the pipe diameter, m;
δ	is the pipe wall thickness, m;
q	is the amount of heat removed, W/m^2;
t	is the temperature, °C;
λ	is the thermal conductivity, W/m · °C;
ν	is the kinematic viscosity, m/sec;
r	is the heat of vaporization, J/kg;
ρ	is the density, kg/m^3;

σ is the surface tension, N/m;
P is the heater power, W.

Subscripts

c is the condenser;
e is the evaporator;
l is the liquid;
a is adiabatic;
av is the average;
in is internal.

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STRUCTURE PARAMETERS OF METAL-FIBER HEAT PIPE WICKS

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The results of an experimental investigation of local and average values of permeability and effective pore diameter for metal-fiber wicks, using a laser-Doppler velocity measurement, are presented and discussed.

Capillary-porous structures made of metallic individual fibers or powder are used successfully as the wicks of heat pipes [1]. Existing methods of preparing powdered and fibrous materials do not yield completely uniform structures for these materials. The properties differ with direction and cross-sectional area.

One of the most important characteristics governing the choice of structure for a specific application is the permeability. The methods of measurement in [2] usually give the average value the specimen permeability, from which one cannot assess local nonuniformities. The authors of [3-5] tried to solve this problem by measuring the rate of filtration with thermisistors of miniature tubes displaced along the specimen surface. The defects of these methods are that they give indirect information, require calibration, have inertia, and cannot measure a rate of filtration that varies with time. In addition, the range of the measured quantity is quite narrow. A new method was suggested in [6]. However, here also one cannot avoid errors introduced by the source of thermal pulses.

The present paper studies the basic properties of permeable capillary-porous bodies using a laser-Doppler velocity measurement (LDVM). With this method one can obtain local permeability coefficients K_i, the distribution of pore diameters with size, the dimension of the maximum and minimum pores, and their number. With this information one can estimate the quality of the material, compare its characteristics with other types of structures, and reach a conclusion as to the suitability of a specific manufacturing technique for specific items. It is also possible to carry out tests to reject items with a large scatter in their properties.

The measurements were carried out on an equipment whose main element is a type LG-75 gas laser. The working section is a hollow cylinder with an inlet for gas carrying scattering particles and a pressure pickoff. The specimen is attached in the upper part in such a way that the gas flow passes through the entire section of the porous insert. Sealing is done at the end section on the mount surface (Fig. 1).

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