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INVESTIGATION OF THE CHARACTERISTICS OF A LOW-TEMPERATURE HEAT PIPE WITH A CRIMPED RETICULAR WICK

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A practical method of simulating weightlessness conditions for ground tests of a low-temperature heat pipe using Freon-22 is developed and checked experimentally.

The use of a wick with a reticular mesh enables one to produce heat pipes with a high heat transmission and low thermal resistance [1].

Possible deviations in the dimensions of the cells of the mesh or their contamination by solid inclusions do not have any considerable effect on the characteristics of the heat pipes. In view of the fact that the channels of the wick are open in vapor space, the establishment of the operating ability of the pipe after drying the wick (e.g., by the action of dynamic forces) is not limited by collapse of the vapor bubbles, as in arterial pipes, and occurs in a short time.

A crimped mesh enables one to construct most simply a capillary structure of the channel type in a long ($L/D \geqslant 50$) heat pipe, both straight and curved. Due to the elasticity of the crimped mesh it is possible to obtain satisfactory contact between the wick and the inner surface of the tube over its whole length.

The crimped mesh forms longitudinal open channels facing the vapor space and closed channels facing the body of the pipe. A feature of this wick is the hydraulic coupling between its channels due to the permeability of the mesh. For a fairly large heat pipe diameter and coarse wick channels not all the channels are filled with liquid under gravitational conditions, even when there is no thermal load. Hence in such heat pipes, when tests are made under gravitational conditions, there is always excess liquid which fills the lower part of the vapor channel, which henceforth will be called a "pool." The dimensions of the pool vary depending on the temperature and the power supplied and have a considerable effect on the thermal characteristics of the pipe when it has been tested. At the same time, when operating the pipe under conditions of weightlessness, the liquid completely fills all the channels of the wick, and there is no pool, and possible excess liquid (due to drying of the channels, excess priming, and an increase in the specific volume when the temperature increases) accumulates at the end of the condensation zone.

The problem therefore arises of determining reliable characteristics of a heat pipe intended for operation under conditions of weightlessness when it has been tested under gravitational conditions.

The problem is solved by determining the value of the slope of the pipe when it is tested under ground conditions, for which the heat-transfer properties of the pipe are not higher than under weightlessness conditions. The slope is determined by a method worked out on a model of the heat pipe with a wick whose channels lie in one plane.

1. Operating Process under Weightlessness Conditions. Consider a heat pipe with a

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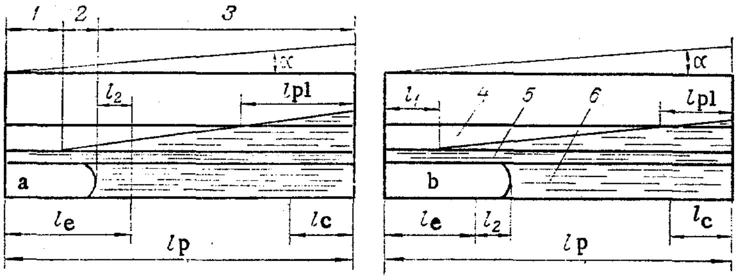


Fig. 1. Theoretical scheme of a plane heat pipe: 1, 2, 3) drainage parts of the channels and the mesh of the wick; 4) narrow wick channels; 5) mesh; and 6) wide wick channels.

crimped reticular wick having channels of two sizes: narrow channels turned towards the vapor space and wide channels turned towards the body of the pipe.

Under conditions of weightlessness when there is no thermal load the liquid uniformly fills the channels of the wick. In this case the curvature of the meniscus is the same in all the capillaries. When thermal power is introduced into the pipe the curvature of the meniscus in all the channels increases in the direction from the compensation zone to the evaporation zone. As the power increases the meniscuses become steeper, and for each value of the transmitted power there is a certain value of the radius of curvature of the meniscus. When a certain power N_1 is reached the curvature of the meniscuses takes a limiting value for the wide channels. A further increase in the power leads to a collapse of the meniscuses of liquid in these channels and their removal from the beginning of the evaporation zone. The drying boundary is established at the point where the vapor pressure is in equilibrium with the meniscus of maximum curvature. When a certain power N_2 is exceeded, corresponding to the limiting curvature of the meniscuses, drying of the narrow channels occurs and there is further displacement of the meniscuses of liquid from the evaporation zone in the wide channels. In that part of the evaporation zone where the wide and narrow channels are drained, the liquid moves along the capillary pores of the mesh. When there is a further increase in the power, overdrying of the mesh begins and there is an increase in the thermal resistance. The power N_{max}, corresponding to the beginning of overdrying of the mesh, is the limiting value for this heat pipe.

This working process can be reproduced most closely under weightlessness conditions when making tests under gravitational conditions in a heat pipe with a "plane" wick, i.e., with a wick whose channels are situated in one plane. The excess of liquid which appears in the heat pipe when tests are made, forms a pool, the effect of which is the same for all channels and can be taken into account when describing the process mathematically.

2. Mathematical Model of the Operation of a Pipe with a Plane Wick. The theoretical scheme of a pipe with a plane wick is shown in Fig. 1. When formulating the system of equations we will consider two possible positions of the drainage boundary of the wide channels of the wick. These two cases correspond to any practically important ratios of the channel dimensions and the pores of the mesh. In the first case (Fig. 1b) the drainage boundary of the wide channels lies outside the limits of the evaporation zone, while in the second (Fig. 1a) the boundary lies inside the evaporation zone.

The equations are compiled for laminar flow of the vapor and liquid, and the losses due to friction in the pool are ignored in view of their smallness. It is assumed that the level of the liquid in the narrow channels (open to the vapor channel) for the limiting power varies linearly from zero in the evaporation zone to a maximum at the beginning of the pool. According to the theoretical model, the heat pipe is divided into three parts: in the first the liquid moves only along the mesh, in the second it moves in the mesh and in the narrow channels, and in the third part it moves along the mesh, and the narrow and wide channels.

For the first theoretical case the equation for the flow of liquid and vapor has the form

$$\sigma(\cos\theta)\left(\frac{1}{R_{\mathrm{m}}}-\frac{1}{R_{\mathrm{n}}}\right)=\left[\frac{1}{k}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_{\mathrm{m}}}+\frac{32}{D_{\mathrm{hv}}^2}\left(\frac{\mu}{\gamma}\right)_{\mathrm{v}}\frac{1}{S_{\mathrm{v}}}\right]\frac{Gl_1^2}{2l_{\mathrm{e}}}+\gamma_l\, l_1\sin\alpha, \tag{1}$$

and for the second part

$$\sigma(\cos\theta)\left(\frac{1}{R_n}-\frac{1}{R_w}\right)=\left[\frac{\xi_n}{2D_{hn}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_n}\right.$$

$$\left.+\frac{32}{D_{hv}^2}\left(\frac{\mu}{\gamma}\right)_v\frac{1}{S_v}\right]G\left(l_2+\frac{l_e}{2}-\frac{l_1^2}{2l_e}\right)+\gamma_l\,(l_2+l_e-l_1)\sin\alpha. \tag{2}$$

In Eq. (2) we have ignored the motion along the mesh of the wick, since the hydraulic resistance of the mesh is considerably greater than the resistance of the narrow channels.

For the third part the equation of motion of the liquid depends on the ratio of the length of the pool and the condensation zone. For $l_{pl} > l_c$ it has the form

$$\sigma(\cos\theta)\frac{1}{R_w}=\left\{\left[\frac{\xi_w}{2D_{hw}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_n\left(1+\frac{\xi_w D_{hn}^2 S_n}{\xi_n D_{hw}^2 S_w}\right)}+\frac{32}{D_{hv}^2}\left(\frac{\mu}{\gamma}\right)_v\frac{1}{S_v}\right]G+\gamma_l\sin\alpha\right\}\left(l_p-l_e-l_2-\frac{l_{pl}+l_c}{2}\right).$$

For $l_{pl} \leqslant l_c$ we have

$$\sigma(\cos\theta)\frac{1}{R_w}=\left\{\left[\frac{\xi_w}{2D_{hw}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_w\left(1+\frac{\xi_w D_{hn}^2 S_n}{\xi_n D_{hw}^2 S_w}\right)}+\frac{32}{D_{hv}^2}\left(\frac{\mu}{\gamma}\right)_v\frac{1}{S_v}\right]G+\gamma_l\sin\alpha\right\}\left(l_p-l_e-l_2-\frac{l_{pl}+l_c}{2}\right).$$

The length of the pool is given by the expressions

$$S_n l_1+S_w(l_e+l_2)+V_w\left(\frac{\gamma_{100}}{\gamma}-1\right)+V_{ex}=V_{pl} \tag{5}$$

$$V_{pl}=\frac{l_{pl}^2 b\sin\alpha\cos\alpha}{2}. \tag{6}$$

Equation (6) is approximate and can be used with sufficient accuracy for practical purposes when $l_{pl} < (1/2)l_p$.

For the second theoretical case, corresponding to the position of the drainage boundary of the wide channels within the evaporation zone, Eqs. (2), (3), and (4) are somewhat different and have the form

$$\sigma(\cos\theta)\left(\frac{1}{R_n}-\frac{1}{R_w}\right)=\left[\frac{\xi_n}{2D_{hn}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_n}+\frac{32}{D_{hv}^2}\left(\frac{\mu}{\gamma}\right)_v\frac{1}{S_v}\right]G\left(\frac{l_e}{2}-l_2+\frac{l_2^2-l_1^2}{2l_e}\right)+\gamma_l\,(l_e-l_2-l_1)\sin\alpha, \tag{7}$$

$$\sigma(\cos\theta)\frac{1}{R_w}=\left[\frac{\xi_w}{2D_{hw}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_w\left(1+\frac{\xi_w D_{hn}^2 S_n}{\xi_n D_{hw}^2 S_w}\right)}\right.$$

$$\left.+\frac{32}{D_{hv}^2}\left(\frac{\mu}{\gamma}\right)_v\frac{1}{S_v}\right]G\left(l_p+l_2-l_{pl}-l_e-\frac{l_2^2}{2l_e}\right)+\gamma_l\,(l_p-l_e-l_{pl}+l_2)\sin\alpha, \tag{8}$$

$$\sigma(\cos\theta)\frac{1}{R_w}=\left[\frac{\xi_w}{2D_{hw}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_w\left(1+\frac{\xi_w D_{hn}^2 S_n}{\xi_n D_{hw}^2 S_w}\right)}+\frac{32}{D_{hv}^2}\left(\frac{\mu}{\gamma}\right)_v\frac{1}{S_v}\right]G\left(l_p\right.$$

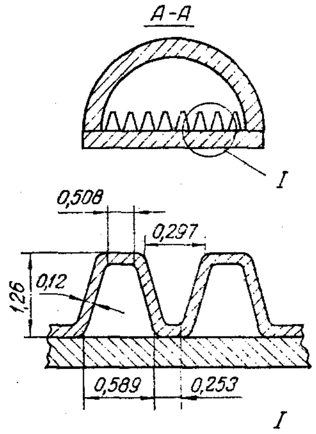
$$\left.-l_e+l_2-\frac{l_2^2}{2l_e}-\frac{l_e-l_{pl}}{2}\right)+\gamma_l\,(l_p-l_e-l_{pl}-l_2)\sin\alpha. \tag{9}$$

The limiting heat-transmitting capability of the heat pipe N_{max} is calculated from the equation

$$N_{max}=rG. \tag{10}$$

The value of the flow rate G is found by solving Eqs. (1), (2), (3), (4), (5), and (6), or (1), (5), (6), (7), (8), and (9).

3. Investigation of an Experimental Pipe with a Plane Wick. The heat pipe shown schematically in Fig. 2 consists of a wick having 17 wide channels (including two side channels) and 14 narrow channels with an overall width of 15.83 mm. The length of the wick is 600 mm, the evaporation zone is 110 mm, and the condensation zone is 110 mm [1]. The heat pipe has a body of semicircular cross section with a flat base, on which the crimped reticular wick is situated. The ends of the tube are made of transparent material for visual observations on the position of the liquid in the tube. The hydraulic diameter of the vapor channel was



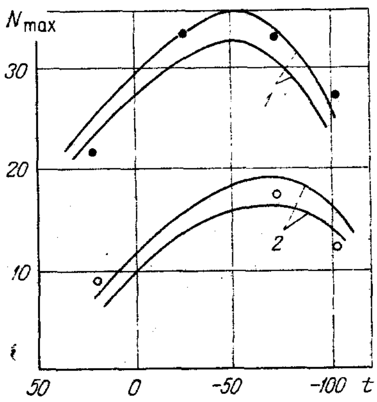


Fig. 2 Fig. 3

Fig. 2. Sketch of the plane pipe (open channels 14, closed channels 17).

Fig. 3. Comparison of the theoretical and experimental values: 1) experiment and calculation for a slope h = 1.9 mm; 2) the same for h = 5.1 mm; N_{max}, W; t, °C.

chosen to be equal to the hydraulic diameter of the simulated pipe of circular cross section. The dimensions of the channels, shown in Fig. 2, are average, obtained from a large number of measurements on the length of the channels. The heat pipe was charged with 16 grams of Freon-22. The amount of the charge ensures complete filling of the wick at a temperature of −100°C. The volume of the wick was determined by calculation and experiment and was 10.5 ml and 10.4 ml respectively.

The purpose of the research program was to determine the heat-transmitting capability of the heat pipe in the temperature range from +20°C to −100°C for different slopes of the heat pipe. The error in setting the tube in a given position in the tests was not greater than 0.1 mm. The unmonitored heat influx to the pipe did not exceed 1 W.

During the tests we made observations on the change in the volume of excess liquid in the condensation zone and drainage in the evaporation zone. As the power increased at constant temperature this volume increased, which indicated drainage of wide channels (facing the body). We then observed drainage of the narrow channels with a continued increase in the volume of liquid in the condensation zone.

The limiting power was found from the sharp increase in the temperature of the initial part of the evaporation zone, which corresponded to practically complete drying of the wick in this part.

The results of tests of the heat pipe are shown in Fig. 3, where the continuous lines represent the theoretical dependences of the heat-transmitting ability on the temperature for different slopes of the heat pipe. In this case the end of the compensation zone was situated 1.9 mm and 5.1 mm below the beginning of the evaporation zone. Theoretical curves were obtained for the geometrical dimensions of the wick shown in Fig. 2. For each slope the upper and lower theoretical curves were obtained for values of the radius of the pores of the mesh equal to 18×10^{-6} and 23×10^{-6} m, and values of the permeability of the mesh of 10×10^{-12} and 2×10^{-12} m^2 respectively. These values of the radius of the pores and the permeability are the limiting values of these characteristics obtained experimentally for the wick considered. The coefficients of friction of the liquid in the channels were taken from [2] and the thermal properties of Freon-22 from [3].

It follows from a comparison of the experimental results and the theoretical curves shown in Fig. 3 that the mathematical model used enables the heat-transmitting capability of a pipe with a crimped wick to be determined with satisfactory accuracy when making tests under gravitational conditions.

For example, for a slope of the tube of 1.9 mm the difference between the calculated values of the heat-transmitting capability and the experimental values did not exceed 5-10% over the temperature range from +20°C to −100°C.

For a slope of the tube of 5.1 mm the difference between the theoretical and experimental results increased somewhat and was approximately 10-15%. The difference between the theoretical and experimental results is due to the complexity of the geometrical shape of the channel and the related error in determining their dimensions.

The satisfactory results obtained enable the above mathematical model of the operating process of the heat pipe with a crimped wick to be used to calculate its heat-transmitting capability under weightlessness conditions.

4. Method of Calculation and Tests of Heat Pipes Intended for Operation under Weightlessness Conditions. To calculate the heat-transfer ability of the pipe under conditions of weightlessness we must ignore the term $\gamma_l l \sin\alpha$ in Eqs. (1)-(9) and also disregard the pool, assuming the position of the excess of liquid to be in the form of a plug at the end of the condensation zone. We can then obtain from the above set of equations a formula for calculating the heat-transfer ability of a circular pipe with a cramped reticular wick under weightlessness conditions. This formula has the form

$$N_{\max} = r\frac{\sigma\cos\theta}{l_{\mathrm{eff}}}\left[\frac{\frac{1}{R_m}-\frac{1}{R_n}}{\frac{1}{k}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_m}+\frac{32}{D_{hv}^2}\frac{1}{S_v}\left(\frac{\mu}{\gamma}\right)_v}+\frac{\frac{1}{R_n}-\frac{1}{R_w}}{\frac{\xi_n}{2D_{hn}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_n}+\frac{32}{D_{hv}^2}\frac{1}{S_v}\left(\frac{\mu}{\gamma}\right)_v}\right.$$
$$\left.+\frac{\frac{1}{R_w}-\frac{1}{R_p}}{\frac{\xi_w}{2D_{hv}^2}\left(\frac{\mu}{\gamma}\right)_l\frac{1}{S_w\left(1+\frac{\xi_w D_{hn}^2 S_n}{\xi_n D_{hw}^2 S_w}\right)}+\frac{32}{D_{hv}^2}\frac{1}{S_v}\left(\frac{\mu}{\gamma}\right)_v}\right]. \tag{11}$$

where $l_{\mathrm{eff}} = l_p - (l_e + l_c)/2$ is the effective length of the heat pipe.

To design heat pipes with crimped reticular wicks the heat-transfer ability of the wick must be investigated on a plane model. Using the results obtained one can determine the theoretical characteristic of the pipe under weightlessness conditions. The experimental investigation of the characteristics of a pipe when it is operating under gravitational conditions in the apparatus for which it is intended should be made for those values of the slope for which its heat-transfer ability does not exceed the values for the weightlessness conditions [4]. The value of the required slope is chosen experimentally.

NOTATION

σ is the surface tension of the liquid, N/m; θ, the wetting angle, degrees; R, the radius of curvature of the meniscus, m; k, the permeability of the mesh, m^2; μ, the viscosity of the working material, $N\cdot sec/m^2$; γ, the density of the working material, kg/m^3; γ_{100}, the density of the liquid at a temperature of −100°C; S, the area of cross section, m^2; D_h, the hydraulic diameter, m; l, the length of the part of the tube, m; b, the width of the wick, m; G, the flow rate of the working material, kg/sec; α, the slope of the pipe with respect to the horizontal, degrees; ξ, the coefficient of friction of the liquid; V_w, the volume of the liquid absorbed by the wick, m^3; V_{ex}, the volume of excess charging of working material, m^3; and r, the specific heat of vaporization, J/kg. The subscripts employed are as follows: m, mesh; n, narrow channel; w, wide channel; v, vapor; l, liquid; e, evaporation zone; c, condensation zone; pl, pool; and p, heat pipe (wick).

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