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A CRYOGENIC HEAT PIPE WITH A RESERVOIR IN THE TRANSPORT ZONE

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The article discusses an explosion-proof design for a cryogenic heat pipe.

Cryogenic heat pipes have come into increasingly wide use in recent years; they are particularly promising for thermally stressed devices of modern design whose optimum operating conditions require intensive cooling. The basic requirements for the design of such devices include a combination of compactness, independence, and high reliability with minimum weight.

However, cryogenic heat pipes of traditional construction have a number of specific characteristics that substantially reduce the reliability of the design; the most important such characteristic is the sharp increase of pressure in the heat-pipe cavity when the temperature of the surrounding medium is increased. An absolute majority of cryogenic devices are in an inoperable condition an average of 50% of their total service time, i.e., at an ambient temperature of $\sim 20°C$; in addition, the possibility of breakdown situations that will bring increased temperatures is not excluded.

If in such a device cryogenic heat pipes are used for cooling some elements, the operating liquid may be in the supercritical state. Let us consider as an example a cryogenic heat pipe operating with liquid nitrogen and having the following geometric dimensions: length $l = 0.5$ m, inner diameter of jacket $d_{in} = 0.01$ m, thickness of cylindrical porous bronze wick baked onto the inner wall of the jacket $\delta = 0.002$ m, and porosity $\sim 40\%$.

The maximum heat flux transmitted by such a heat pipe at a working temperature of 70-80°K can be calculated by the formula [1]

$$Q = \frac{i\rho kA}{\eta l}\left(\frac{2\sigma\cos\theta}{r_e} + \rho g l \cos\varphi\right) \tag{1}$$

and amounts to approximately 4 W if the pipe is charged with liquid nitrogen in an amount of

$$m = \rho_l Alk + \rho_v \pi \left(\frac{d_{in} - 2\delta}{2}\right)^2 l, \tag{2}$$

which, with the heat-pipe parameters assumed above, amounts to $5.81 \cdot 10^{-3}$ kg, corresponding to a pressure of $p > 1{,}000$ abs. atm on the base at 300°K [2]. It is clear that the reliability of a system having such a stressed construction is sharply reduced, not to mention the fact that the increase in the thickness of the jacket walls leads to an increase in the temperature drop in the evaporation and condensation zones, which makes the main contribution to the total temperature drop along the pipe [3], and this must be kept down to a minimum.

In order to reduce the pressure in the operating cavity of a cryogenic heat pipe, we can make use of additional gas reservoirs connected to the jacket of the tube by a capillary or placed on the jacket of the heat pipe beyond the condensation zone. In some cases such con-

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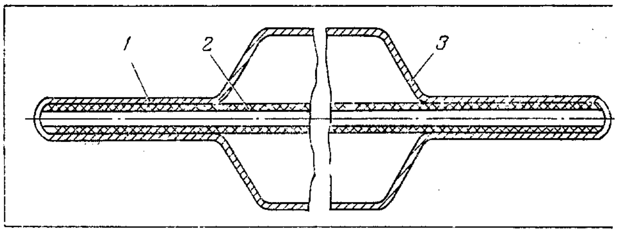

Fig. 1. Scheme of cryogenic heat pipe with reservoir: 1) end piece of evaporation and condensation zones; 2) wick; 3) reservoir.

TABLE 1. Characteristics of Heat Pipe

Q, W	ΔT, °C with reservoir	ΔT, °C without reservoir
1	1,5	1,45
2	1,6	1,6
3	2,9	2,95

structions of heat pipes are inconvenient in practice. In the heat pipe described below the gas reservoir is situated in the transport zone (see Fig. 1), i.e., its shell consists of sections to which a porous wick is baked in the evaporation and condensation zones and of the jacket of the gas reservoir, through which the porous wick passes axially without touching the walls of the jacket in the transport zone. The amount of liquid nitrogen charged into the pipe remains practically the same as before, since

$$m = m_l + m_v, \tag{3}$$

where m_v amount to a few hundredths of 1% of m_l.

The operation of such a heat pipe at low temperatures is analogous to the operation of the pipe considered above for the condition when the temperature of the gas-reservoir jacket is higher by a fraction of a degree than the temperature of the wick in the working state, which precludes the possibility of condensation on the walls of the reservoir. When the heat pipe is in operation, the pressure in the reservoir from the outside of the transport section of the wick filled with liquid nitrogen is $p_{res} \approx p_{c.v.}$, and the transfer of vapor from the evaporation zone into the condensation zone takes place through the vapor channel as usual. When the temperature rises in a breakdown situation or in storage under normal conditions, the pressure inside the heat pipe is reduced below the pressure at the temperature considered above by the same factor by which the internal volume of the pipe exceeds the volume of the pipe without a reservoir.

The proposed heat pipe was developed for the cooling of systems in a vacuum with p = 10^{-6} torr, and therefore thermal insulation of the jacket was not required; in general, when such insulation of the jacket is required, we may use any type of insulation used in cryogenic technology. The calculation of the transmitted power was carried out in accordance with formula (1) for an ordinary heat pipe. The dimensions of the reservoir were selected in such a way that when the pipe is fully charged, the pressure in it under normal conditions will be 13 abs. atm.

In order to confirm that the ordinary calculation methods used for heat pipes are applicable to this type of construction, we conducted tests of the operating capacity of heat pipes with a reservoir in the transport zone in the horizontal position. Table 1 gives the characteristics of the heat pipe considered here and those of a heat pipe without a reservoir, taken from the data of Vasil'ev et al. [4].

From a comparison of the data obtained with the results found for an ordinary cryogenic tube charged with liquid nitrogen [4-5], it is clear that the above design for a heat pipe has the same characteristics but is more reliable in operation, storage, and transport.

NOTATION

Q, heat flux, W; i, evaporation heat, J/kg; k, porosity, %; l, length, m; ρ, density, kg/m³; A, cross-sectional area of wick, m²; η, dynamic viscosity, N·sec/m²; σ, surface-tension coefficient, N/m; θ, wetting angle, deg; r_e, effective radius of pores, m; φ, angle of inclination of pipe to the horizontal, deg; m_v, mass of vapor, kg; d_{in}, inner diameter of pipe jacket, m; δ, thickness of wick wall, m; p, pressure, abs. atm; m_l, mass of liquid, kg.

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INVESTIGATION OF THE MAXIMUM HEAT-TRANSFER CAPACITY OF CLOSED TWO-PHASE THERMOSIPHONS

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We determine the maximum heat fluxes transmitted by thermosiphons, depending on the geometric dimensions, the working pressure, the type of liquid used, and the degree of filling with coolant.

In recent years there has been an increase of interest in heat-transfer devices constructed in the form of wickless heat pipes, i.e., closed two-phase thermosiphons. The advantages of these devices (simplicity of manufacture, reliability of operation, fairly high heat-transfer capacity, etc.) open prospects for their widespread use in various fields of technology [1-3].

There are a good many studies devoted to the investigation of heat-transfer processes in closed evaporative thermosiphons. An absolute majority of these studies are concerned essentially with questions of heat exchange in the evaporation and condensation segments of the heat-transfer device. Limiting regimes of operation of closed two-phase thermosiphons have not been sufficiently investigated. Stoyanov [4] considered a heat-transfer crisis resulting from the complete evaporation of the intermediate coolant. The results of this analysis enable us to determine the maximum working temperature of a heat-transfer device as a function of the degree to which it is filled with coolant. Concerning the maximum heat fluxes transmitted by two-phase thermosiphons, there are only fragmentary data (for example, [5]), from which it is not possible to derive any generalized relationship.

In the present article we give the results of an investigation of the maximum heat fluxes transmitted by vertical two-phase thermosiphons, depending on their geometric, physical, and regime parameters. The range of variation of the parameters is shown in Table 1.

The investigations were carried on an experimental apparatus consisting of experimental thermosiphons with heat-input and heat-removal segments, electrical heating units, systems for the input of cooling water, and measuring devices. The experimental thermosiphons were

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