REVIEW

HEAT PIPES AND THEIR USE IN TECHNOLOGY

L. L. Vasil'ev

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I. INTRODUCTION

Heat pipes is the name given to closed autonomous evaporative–condensing devices. There is as yet no unique terminology for heat pipes and it is difficult to say what should be included in this concept. The name heat pipe came from English literature and does not fully reflect the special features of heat and mass transfer in evaporative– condensing devices of closed type.

Heat pipes are one of the astonishing inventions of science in the 1960's and, in the author's opinion, are capable of being as widely disseminated as lasers in their applications and their value for industry.

The main function of heat pipes is the transfer of heat from one region of space to another without significant loss. The perfect heat pipe should possess very small thermal resistance, less by a factor of tens of thousands than the thermal resistance of a copper bar of the same geometrical size.

In addition, it can be used successfully, for example, as a temperature regulator, a thermal diode, a transformer, an accumulator, a battery, and a device for converting heat energy into mechanical energy, calculated energy, etc. With the help of heat pipes one can store thermal energy and release it rapidly at the required time. Therefore, the requirements as regards these pipes differ from those applicable to classical examples of heat pipes.

II. SOME ASPECTS OF HEAT-PIPE THEORY

1. Since there is a liquid and its vapors in any heat pipe, the operating temperature range is limited, on the one hand, by the triple point T_{tr} and, on the other hand, by the critical point T_{cr}. Depending on the temperature at which the heat pipe is used (near T_{tr} or near T_{cr}) different factors will dominate: Near the triple point the dominant factors are vapor dynamics, the acoustic limit, the limit of interaction of the vapor with the liquid, the kinetics of phase transitions, etc.; near the critical point the high pressure may cause mechanical rupture of the structure, and the latent heat of vaporization of the liquid is decreased.

From the thermodynamic viewpoint an operating regime near the critical point is preferred, since the dependence of vapor pressure on temperature in this region is flatter, and the pressure drop produces only insignificant temperature changes.

According to the Clausius–Clapeyron equation, the phase equilibrium condition f(P, T) = 0 should hold at the liquid–vapor interface, and if we assume that the latent heat of vaporization is constant and that the vapor is a perfect gas, then the following relation is valid:

$$(P/P_0) = \exp\left(-\frac{1}{R^*T}\right). \qquad (1)$$

Because of the curvature of the liquid–vapor interface in the pores of the evaporator of a heat pipe and because of the temperature gradient, the liquid must be superheated in the evaporator. When the superheating reaches a critical value, liquid boiling begins.

The vapor is superheated relative to the liquid in the pores in the heat-pipe condenser. Therefore, under specific conditions there may be liquid droplets in the vapor (a mist), both in the evaporator and in the condenser.

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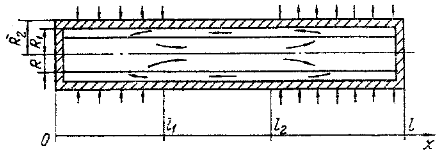


Fig. 1. A heat pipe. R is the radius of the perforated insert; R_1 and R_2 are the radii of the inner and outer body surfaces.

The process of vapor transport in a heat pipe is analogous, to some extent, to the transport process in a convergent–divergent nozzle. In the adiabatic zone the vapor can expand, accompanied by the Joule–Thomson effect [1]. We therefore find great variety (factors of 10) in the structures of heat pipes and vapor chambers in which the only common features are the heat-transfer process in the evaporator and in the condenser in the presence of phase transitions, while the method and type of mass transfer and the features of the phase transitions may be so diverse that one can only say that it is impossible to create a single theory for the processes of transition of energy and matter in heat pipes.

It is convenient to establish some general concepts for the transport process in heat pipes – for the classical heat pipes used under steady conditions. The problem of liquid and vapor mechanics in classical pipes has three components:

1) transport of vapor along the heat pipe from the evaporator to the condenser;

2) transport of liquid from the condenser to the evaporator under a capillary potential gradient;

3) interaction between the vapor and the liquid.

An analytical solution of all these problems is very difficult, especially for the problem of transport of liquid in the porous structure of a heat-pipe wick. The boundary conditions at the inlet and exit (evaporation and condensation) depend strongly on the temperature field, the liquid concentration field, and on the thermodynamic conditions of equilibrium.

A change in the operating conditions of a heat pipe can affect the nature of the flow of the transport processes. For example, depending on the thermal conditions and the operating temperature, the conditions of vapor motion may be laminar or turbulent, and the vapor may be compressible or incompressible. The characteristics of any particular vapor flow regime can be expressed as a function of the Reynolds number Re and the Mach number M in the adiabatic zone:

$$\mathrm{Re} = \frac{QR_V}{\pi R_V^2 L \eta_V}\;;\quad M = \frac{Q}{\pi R_V^2 LP}\sqrt{\frac{c_V}{c_p}\,\frac{R}{\mu}\,T_V}\,. \tag{2}$$

Therefore, the transport process in the vapor or the liquid phase can be considered separately only under specific assumptions. In particular, we assume that the vapor flow is isothermal and that the thermal conditions of heat transfer are constant in the evaporator and in the condenser; thus, we can consider the vapor flow dynamics independently of the temperature field and of the liquid flow in the wick.

We can probably formulate the transport problem in a heat pipe more rigorously than in the associated case where all the basic transport processes in the vapor and liquid phases are considered simultaneously. In particular, a formulation of this problem was given in [2]. For long, cylindrical, high-temperature pipes with a capillary structure in the form of a coaxial gap between the body and the perforated screen, assuming turbulent mixing vapor, parameters such as the density ρ_V and the temperature T_V depend only slightly on the radial coordinate r (Fig. 1).

Therefore, the averaged system of equations describing the vapor motion, averaged over the cross section of the vapor channel, has the following form:

$$\frac{d}{dx}(\rho_V v) = -\frac{2}{R}\,\tau^*|_{r=R}\,; \tag{3}$$

$$\frac{d}{dx}[\rho_V + \beta\rho_V v^2] = -\frac{2}{R}\,\tau_w^*\,; \tag{4}$$

$$\frac{d}{dx}\left[\rho_V v\left(H_V+\frac{\alpha v^2}{2}\right)\right]=-\frac{2}{R}(\tau^* H_V)|_{r=R}; \tag{5}$$

$$P_V=\frac{\rho_V R^* T_V}{M_V}; \tag{6}$$

$$v=\frac{2}{R^2}\int_0^R v_x r dr; \quad \beta=\frac{2}{v^2 r^2}\int_0^R v_x^2 r dr; \tag{7}$$

$$\alpha=\frac{2}{v^3 R^2}\int_0^R v_x^3 r dr,$$

where v_x is the vapor velocity component along the x axis.

The system of equations (3)-(5) can be closed if we know the momentum flux coefficient β, the friction force coefficient τ_w^*, and the condensation coefficient k.

The transport in the shell and the wick of a heat pipe include transport of heat and filtration of liquid through the capillary structure. In the shell the heat propagation is described by the equation

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}=0, \tag{8}$$

while the propagation of heat through the layer of heat-transfer agent in the coaxial gap is given by

$$\frac{\partial}{\partial x}\left[r\left(\tau_l^* H_l-\lambda\frac{\partial T}{\partial x}\right)\right]+\frac{\partial}{\partial r}\left[r\left(\tau_l^* H_l-\lambda\frac{\partial T}{\partial r}\right)\right]=0. \tag{9}$$

The value of Re for the liquid is less than 1, and we can therefore use the Stokes approximation:

$$\frac{\partial r u_x}{\partial x}+\frac{\partial r u_r}{\partial r}=0;$$
$$\frac{\eta_l}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u_r}{\partial r}\right)=\frac{\partial P_l}{\partial x}; \quad \frac{\partial P_l}{\partial r}=0, \tag{10}$$

where u_x and u_r are the liquid velocity components.

If the heat flux at the outer surface of the pipe is given, the corresponding system of boundary conditions and coupling conditions takes the form

$$v|_{x=0}=v|_{x=l}=0, \tag{11}$$

$$\left.\frac{\partial T}{\partial x}\right|_{x=0}=\left.\frac{\partial T}{\partial x}\right|_{x=l}=0; \tag{12}$$

$$-\lambda\left.\frac{\partial T}{\partial r}\right|_{r=R_1}=q_r\frac{R_2}{R_1}; \tag{13}$$

$$L\tau=-\lambda\left.\frac{\partial T}{\partial r}\right|_{r=R}; \quad \tau^*=\frac{2\alpha F}{2-\alpha}\sqrt{\frac{M}{2\pi R^*}}\left[\frac{P_V}{\sqrt{T_V}}-\frac{P(T_0)}{\sqrt{T_0}}\right]_{r=R}; \tag{14}$$

$$T_V|_{r=R}=T_0\left[1+\sqrt{\frac{2\pi R^*\tau_0^*}{M}}\frac{\tau^*}{8FP(T_0)[\alpha+\beta(1-\alpha)]}\right]; \tag{15}$$

$$u_x|_{r=R}=u_x|_{r=R_1}=u_r|_{r=R_1}=0; \quad u_r|_{r=R}=\frac{\tau^*}{\rho_l}, \tag{16}$$

where T_0 is the temperature of the liquid surface at r = R; $P(T_0)$ is the saturated vapor pressure at T_0; and F is the surface porosity. To solve this problem we must satisfy the condition

$$\max[(P_{V_1}-P_{V_2})-(P_{l_1}-P_{l_2})-(\rho_{V_1}v_{r_1}^2-\rho_{l_1}u_{r_1}^2)-$$
$$-(\rho_{V_2}v_{r_2}^2-\rho_{l_2}u_{r_2}^2)]\leqslant P_{k\,\max}; \tag{17}$$

$$P_{k\,\max} = \frac{2\sigma}{r^*}\cos\theta_0 - \frac{\sigma}{R}, \tag{18}$$

where r* is the radius of the screen pores; θ_0 is the limiting boundary wetting angle.

Naturally, this statement of the problem should be considered as one variant of a special problem for this specific type of high-temperature, liquid-metal heat pipe.

2. Transport Processes in Heat-Pipe Wicks. In most forms of low-temperature heat pipes the chief limiting factor in transport is the rate of motion of liquid along the capillary structure. From the viewpoint of liquid mechanics this is a very complex problem involving liquid-transport effects in the porous structure and interaction processes between the liquid and the vapor, surface-tension forces with pronounced effects at the liquid–solid wall and liquid–vapor interfaces (since the porous structure is extremely thin), and, finally, phase-transition processes (evaporation–boiling and condensation).

The requirements in regard to the capillary structure are conflicting: To achieve a high capillary pressure drop and to improve the heat-transfer conditions one should decrease the size of the pores and the wick thickness; to improve the transport properties of the porous structure, on the other hand, one should increase the size of the pores and the wick thickness. These conflicting requirements are satisfied by composite wicks which have low hydrodynamic resistance and high capillary potential.

3. Effect of Wick-Structure Characteristics on the Convective Liquid-Transport Process. The liquid flux in a porous solid or a capillary structure can be determined from the Darcy law

$$m = \rho_l w_l = -\frac{K_l}{\mu_l}\,\mathrm{grad}\,P \tag{19}$$

or by the Poiseuille formula. Correspondingly, the pressure losses for motion of liquid under the action of capillary forces are as follows:

for metal–ceramic, lattice, or fiber-type structures

$$\Delta P_l = \frac{\mu_l Q l_p^{ef}}{K_l \rho_l \Pi b a L} \pm l_p \rho_l g \sin\gamma; \quad l_p^{ef} = \frac{l_u + l_{con}}{2} + l_{a.z.}; \tag{20}$$

where b is the width; a is the thickness; and K_l is the permeability.

For a system of longitudinal channels and arteries of triangular, rectangular, trapezoidal, and circular shape

$$\Delta P_l = -\frac{32\mu_l Q l_p^{ef}}{\rho_l L A D_{ef}^2} \pm l_p \rho_l g \sin\gamma; \tag{21}$$

where D_{ef} = 4A/wetted perimeter of the channel, and A is the channel cross-sectional area. The maximum capillary pressure is

$$\Delta P_{cap\max} \geqslant \Delta P_l = 2\sigma\left(\frac{\cos\theta_1}{r_1} - \frac{\cos\theta_2}{r_2}\right), \tag{22}$$

where θ_1 and θ_2 are the contact wetting angles in the evaporator and condenser; r_1 and r_2 are the radii of curvature of the liquid–vapor interface in the evaporator and the condenser;

$$\Delta P_{cap\max} = \frac{2\sigma\cos\theta_1}{r_{ef}}; \tag{23}$$

and r_{ef} is the effective radius.

For an isotropic porous structure, the effective radius is the average pore radius. For a triangular channel [3]

$$r_{ef} = \frac{h}{\csc(\alpha - 1)}. \tag{24}$$

For semicircular channels

$$r_{ef} = \frac{\pi h}{\pi - 2}. \tag{25}$$

For square channels

$$r_{ef} = h. \tag{26}$$

A particularly complex problem is the transport process in the metal–ceramic, felt, and lattice structures. For these the permeability K depends on the porosity, the sinuosity of the path, and the degree of filling of the pores. For large heat loads the saturation of the porous wick with respect to the liquid varies along the pipe, i.e., it depends on the coordinate:

$$K = f(\Pi, b). \tag{27}$$

Most porous bodies are semicapillary structures having a differential curve for the distribution of surface permeability f(K) [4]:

$$f(K) = \sum_{i=1}^{N} \delta A_i, \tag{28}$$

where A_i are constants satisfying the condition

$$\sum_{i=1}^{N} A_i = 1; \tag{29}$$

and N is the final value.

4. Dynamics of the Transport Process in Heat Pipes. The motion of a liquid in a semicapillary wick with a temperature gradient can be calculated by solving the capillary diffusion equation

$$\frac{\partial u}{\partial \tau} = a_m \nabla^2 u + a_m^{\tau} \nabla^2 T. \tag{30}$$

If we neglect the temperature gradient along the pipe, it is valid to use the differential diffusion equation, allowing for laminar convective transport:

$$\frac{\partial \omega}{\partial \tau} = D \frac{\partial^2 \omega}{\partial z^2} - W_z \frac{\partial \omega}{\partial z}, \tag{31}$$

where D is a dispersion coefficient accounting for different transport mechanisms such as molecular diffusion, turbulent vorticity, recirculation of liquid in the pores due to local pressure gradients, thermal and viscous slip, macroscopic dispersion, etc.; W_z is a convective term given by the Darcy law; and ω is the mass concentration.

If there is turbulent liquid transport (an extremely rare case in porous media), Eq. (31) can be written in the form

$$\frac{\partial \omega}{\partial \tau} + v_i \frac{\partial \omega}{\partial z_i} = \frac{\partial}{\partial z_i} \left(D_{i,k} \frac{\partial \omega}{\partial z_k} \right)_{i,k\ldots=1,2,3} \tag{32}$$

where $D_{i,k\ldots}$ is a second-rank tensor.

5. Heat and Mass Transfer in Low-Temperature Heat-Pipe Wicks. In low-temperature heat pipes, according to the Lykov theory [5], the thermodynamic motive forces for transport of liquid in a porous medium are the pressure gradient (∇P), the temperature gradient (∇T), and the concentration gradient (∇U).

The system of differential heat- and mass-transfer equations has the form

$$\frac{\partial T}{\partial \tau} = k_{11} \nabla^2 T + k_{12} \nabla^2 U + k_{13} \nabla^2 P; \tag{33}$$

$$\frac{\partial U}{\partial \tau} = k_{21} \nabla^2 T + k_{22} \nabla^2 U + k_{23} \nabla^2 P; \tag{34}$$

$$\frac{\partial P}{\partial \tau} = k_{31} \nabla^2 T + k_{32} \nabla^2 U + k_{33} \nabla^2 P, \tag{35}$$

where $k_{i,j}$ (i, j = 1, 2, 3) are the heat- and mass-transfer coefficients and the thermodynamic characteristics.

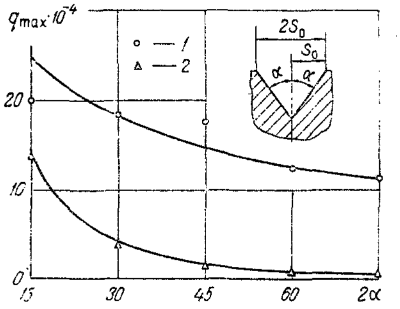


Fig. 2

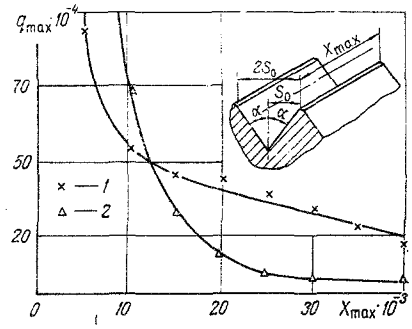


Fig. 3

Fig. 2. Heat flux in the evaporator of a heat pipe as a function of the vertex angle of a triangular channel (the heat-transfer agent is acetone; the wall material is brass): 1) experiment; 2) data of [42]. $2S_0$ = const = 0.4 mm; X_{max} = const = 40 mm; q_{max}, $W/m^2 \cdot deg$.

Fig. 3. The maximum flux in the evaporator as a function of the length of the triangular channel X_{max}: 1) experiment; 2) data of [42]. $2S_0$ = const = 0.4 mm; 2α = const = 30°; X_{max}, m.

If the heat-transfer conditions cause intense vaporization of the liquid in the pores, a supplementary pressure gradient (∇P_V)

$$\varepsilon\rho_0 \frac{\partial U}{\partial \tau} = \operatorname{div}(\psi \nabla P_V), \tag{36}$$

where ε is the dimensionless vaporization coefficient ($0 < \varepsilon < 1$); ρ_0 is the dry wick density; and ψ is the relative filtration coefficient. This additional pressure gradient of the vapor inside a porous body causes ejection of liquid droplets from the porous material into the vapor channel.

If we assume that the local vapor and liquid temperature within the porous body is equal to the wall temperature of the pores, then we can write, for the vapor moving along the pores,

$$c\rho_0 \frac{\partial T}{\partial \tau} = \operatorname{div}(\lambda \nabla T) + L \operatorname{div}(\psi \nabla P_V), \tag{37}$$

where c is the specific heat of the porous body, saturated with liquid.

Under steady conditions $\partial T/\partial \tau = 0$ the pressure distribution field will be similar to the temperature field.

For a liquid the unsteady transport process in a porous body in the presence of volume evaporation within the pores can be described by the equation

$$\frac{\partial U}{\partial \tau} = \operatorname{div}[a_m(\nabla U + \delta \nabla T)] + \rho_0 \operatorname{div}(\psi \nabla P_V). \tag{38}$$

Under steady-state conditions we find similar fields for the temperature, pressure, and concentration of liquid within the pores.

The Fourier–Kirchhoff law is customarily used for transport of energy in a porous solid. If this law is applied separately to the liquid in the pores and to the skeleton of the porous solid, we obtain the following equation:

$$[\Pi\rho_l c_l + (1-\Pi)\rho_0 c_0]\frac{\partial T}{\partial \tau} + c_l \rho_l \operatorname{div}[\Pi T u_l] = \lambda_l \operatorname{div}[\nabla(\Pi \overline{T}] + \lambda_0 \operatorname{div}[\nabla(1-\Pi)\overline{T}] + \operatorname{div} q^*, \tag{39}$$

where

$$T = \frac{1}{1-\Pi} T_0 = \frac{1}{\Pi} T_l \quad \text{(average quantities)}; \tag{40}$$

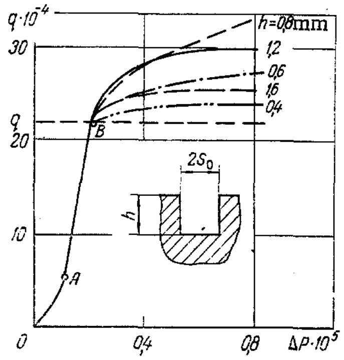


Fig. 4

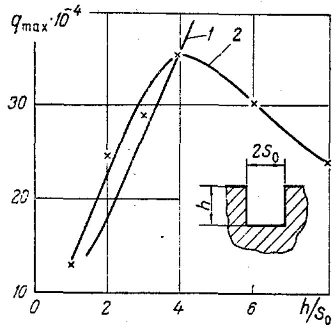


Fig. 5

Fig. 4. The heat flux in the evaporator as a function of the pressure drop between the evaporator and the condenser. The channels are rectangular (the dashed line is the Zuber formula [1]). $2S_0$ = const = 0.44 mm; q, W/m^2; ΔP, N/m^2.

Fig. 5. Heat flux in the evaporator as a function of the geometrical size of the channel: 1) data of [43]; 2) experiment.

$$\bar{u}_l = \frac{\gamma_l}{\rho_l} \text{ (average velocity)};$$

$$q^* = \rho_l c_l [\bar{u}_l T_l - \overline{u_l T_l}] + (\lambda_l - \lambda_0) \frac{1}{V} \int_{A_l} T \vec{n}_0^{\,1} dA. \tag{41}$$

The integration is carried out along the surface of contact of the liquid with the body pores A_i; $\vec{n}_0^{\,1}$ is the unit vector to the surface A, bounding the volume V.

Thus, according to Eq. (39), the transport process in a porous solid saturated with liquid is driven not only by thermal conduction, but also by contact heat transfer between the liquid and the skeleton, as well as by convective heat transfer, which is particularly important for large heat fluxes.

6. Heat Transfer in Heat Pipes. Heat transfer in heat-pipe evaporators takes place by means of evaporation or boiling of the liquid. In the condensers heat transfer occurs during condensation of vapor on the film of liquid or directly in the pores of the wick. Critical boiling of the liquid in the pores is one reason why the power transmitted by a heat pipe is limited.

We now consider heat transfer in evaporators and take, as an example, a structure in the form of channels of triangular and rectangular section. The tube wall material is brass, and the heat-transfer agent is acetone. The rectangular channels have vertex angles of 15, 30, 40, 45, and 90° [6].

The capillary head of a single triangular channel can be expressed as

$$dP_{\text{cap}} = \frac{\sigma \cos\theta}{R_e}. \tag{42}$$

According to the Hagen–Poiseuille formula, the pressure drop for a liquid moving in a channel under isothermal conditions is

$$-\frac{dP}{dx} = \xi \frac{\rho \bar{u}^2}{2D_e}; \quad \xi = \frac{\bar{f}}{\text{Re}}; \quad \text{Re} = \frac{\bar{v} D}{\nu_l}. \tag{43}$$

Equation (43) can be written in the form

$$-\frac{dP}{dx} = f \frac{\mu \bar{u}}{2D_e}; \quad D_e = \frac{4A}{\text{Perimeter}}; \quad R_e = \frac{h \sin\alpha}{1 - \sin\alpha}. \tag{44}$$

From equilibrium of forces at the phase interface

$$\frac{dP}{dx} = \frac{dP_{\text{cap}}}{dx}. \tag{45}$$

Along the evaporator we have

$$dP = dP_{\text{cap}}. \tag{46}$$

We can transform Eq. (45) to the form

$$f\frac{\mu\bar{u}}{2D_e^2} = \frac{\sigma\cos\theta}{R_e}; \quad D_e = \frac{2t\cos\alpha}{1-\sin\alpha}; \quad R_e = \frac{t\cos\alpha}{2(1-\sin\alpha)}. \tag{47}$$

Substituting values of D_e and R_e into Eq. (47) we obtain

$$\frac{f\bar{u}M}{\sigma\cos\theta} = \frac{\sigma t\cos\alpha\,(1-\sin\alpha)}{1-\sin\alpha}. \tag{48}$$

We have verified this calculation by an experimental check of the effect of the triangle vertex angle on the heat flux from the evaporator surface, for constant temperature on the external surface (boundary conditions of the first kind) of the evaporator and condenser (Figs. 2-3) in the temperature range 10-100°C. Heat fluxes up to 25 W/cm^2 were found over the entire area of the evaporator with acetone flowing in triangular channels of vertex angle 30°. The evaporator wall temperature was 64°C.

Figure 4 shows the specific heat flux as a function of the pressure drop along rectangular channels of different depths. The curve $q = f(\Delta P)$ can be divided into three sections. In section OA is extracted heat by evaporation of liquid in the channel from the channel surface; in section AB there is a zone of stable bubble boiling; at the point B we observe critical boiling (q_{cr} = 21.8 W/cm^2). Starting at point B we observe differentiation of the curves $q = f(\Delta P)$ for different channel depths. The heat flux q in zone AB is proportional to the vapor pressure difference between the evaporation and condensation zones:

$$q = N_l\left(\frac{S}{X_{\max}}\right)^2 \cos\theta\, R_{cr} K_1 \nabla P + C;$$

$$k_1 = 10^{-1}; \quad R_w = \frac{2\sigma T_s}{r\rho_v(T_s - T_w)},$$

where T_w is the wall temperature; and T_s is the saturation temperature.

The length of sections OA and AB differs very little for all the rectangular channel sections investigated. The experimentally determined value $q_{cr}^{\max}$ = 21.8 W/cm^2 agrees well with the Zuber formula [1]

$$q_{cr}^{\max} = \frac{\pi}{24}\rho_v L\left[\frac{\sigma(\rho_l - \rho_v)}{\rho_v^2}\right]^{1/4}\sqrt{\frac{\rho_l}{\rho_l - \rho_0}}.$$

The maximum value of q under boiling conditions is 35-40 W/cm^2, and the channel depth is twice the width, being 0.8 mm. Thus, when the liquid is boiling, the channel depth has an appreciable influence on q. The limiting factor is the liquid mass flux along the channel and the discharge resistance of the vapor bubbles. The deeper the channel, the greater the liquid mass flux and the larger the discharge resistance of the vapor bubbles. The dependence of q_{max} on the channel depth is shown in Fig. 5. The relation can be approximated by the formula

$$q_{\max} = 0.25N_l\cos\theta\left(\frac{S_0}{X_{\max}}\right)^2 n^{-1.46}\exp(0.365n), \quad n = \frac{h}{S}; \quad S_0 = 0.5t.$$

III. APPLICATIONS OF HEAT PIPES

The applications of heat pipes can be as diverse as their shapes. We can certainly say that these engineering devices are important for many fields. It is difficult to give preference to any particular branch of industry. For example, the introduction of heat exchangers operating on the heat-pipe principle in the oil and gas industries in the USSR has solved the fundamental problem of laying oil and gas lines in the permafrost regions, cooling hot gas following compressor stages, utilizing the heat of waste gases, gasefying liquefied gas at storage and demand locations; for thermal stabilization of soil along gas and oil lines, to prevent freezing and damage; and for constructing ice pontoons through marshes and the foundations of drilling towers, as well as roads in permafrost regions.

About 100,000 heat pipes, ranging in length from 10 to 20 m, are being used for thermal control of the permafrost zone in constructing an oil pipeline in Alaska, in the USA, of length 798 miles. Each of these is capable of giving up to 300 W of thermal power for a temperature drop of less than 1°C over its length for a period of 30 years without maintenance.

The USA is developing a drill for ultradeep drilling of a bore in the form of a miniature fast-neutron reactor cooled by means of heat pipes.

There are great possibilities for using heat pipes in the power and electrical-engineering industries. For example, centrifugal heat pipes can be used effectively as heat exchangers for heating air and cooling liquids and for temperature control of windings, shafts, and rotors of electrical machinery, fans, bearings of gas turbines, and oven fans. Centrifugal heat pipes are used for cooling asynchronous motors with short-circuited cast rotors.

Such motors are used mainly in mechanical engineering. With the use of centrifugal heat pipes in a rotor it has become possible to control the motor speed electrically, eliminating the need for complex transmissions and gear trains. The asynchronous motor becomes a competitor of the dc motor, and its cost is several times less. Significant success has been achieved in this direction in several countries. The Japanese firm Fujitsu Fanuka and the Siemens Company (Federal Republic of Germany) began commercial production in 1975 of dc torque motors with a heat pipe in the rotor.

The centrifugal heat pipe has been used successfully as a high-mass motor in a number of special asynchronous electric motors. The thermal resistance of this type of rotor has been decreased by a factor of 10, and the heat flux from the stator to the rotor has been increased significantly, leading to an appreciable reduction in the stator winding temperature (by 15-20°C). The rotor temperature is then decreased by more than 30°C. Besides the axisymmetric centrifugal heat pipe, centrifugal heat pipes with near-axial rotation, located at some distance from the rotor axis, are used for similar purposes. Classical heat pipes with a capillary pump are used to cool the stator windings of electrical machinery.

In the electrical-engineering industry heat pipes are used successfully not only to cool electrical machinery, but also to cool high-voltage copper contact switches. The material of the pipe is copper, and the heat-transfer agent is water. These tubes reduce the temperature of the moving contacts by 45-50°C and the temperature of the fixed contacts by 30°C. The transmitted power is 200-300 W.

Investigations are currently in progress in the USSR to explore the possible use of heat pipes to cool transformers, both air-filled and oil-filled, miniature and high-power, and for cooling of electrical busbars.

Broadly based research is in progress to apply heat pipes in high-power electrical engineering and electronics to cool semiconductor devices (thyristors) and current rectifiers and to suppress short-circuit currents.

For example, the West German firm Brown Boveri Corporation has developed a system of electronic devices with heat pipes.

A. Thyristor systems of power greater than 1 kW; the thermal resistance R of the heat pipe is 0.035°K/W, and the cooling air velocity is v = 6 m/sec.

B. A device for a portable current rectifier system (700 W; thermal resistance 0.055°K/W; cooling air velocity v = 6 m/sec).

Heat pipes have proved adaptable to the incorporation of electronic equipment, thereby increasing the cooling effect by factors of 10.

Products of the British SRDE laboratory (Signal Research and Development Establishment) include the following: heat pipes in the form of planar electrical insulators, a heat pipe of very small diameter, and various combinations of heat pipes and thermally insulating modules.

Very interesting possibilities have opened up for producing static batteries and thermal energy converters based on heat pipes, thermal diodes, vapor chambers, etc., and materials which vary their aggregate state (fused salts, metals, sulfur with halogens, etc.; operating temperature 500-800°C); the material of the heat pipes is stainless steel, and the heat-transfer agent is sodium. The thermal power stored can be up to 10-100 kW · h. High-temperature heat pipes using alkali metals can be employed successfully as electrodes in plasma generators.

In the energy industry there is a trend to build electric stations using solar energy and hot springs. At present, an electric station of power of at least 100 kW is under construction in the southern USA; it takes the form of a battery of high-temperature heat pipes, heated by the sun, and working into water-vapor generators or thermoelectric converters. Such batteries of heat pipes, linked to heat-storage units, will make it possible to develop electrical energy around the clock. There are plans to use heat pipes as electric cables and distribution lines.

Heat pipes, in conjunction with thermal-energy-storage units, offer a complete solution to the problem of energy production and cooling of autonomous meteorological radio stations, operating on batteries in inaccessible locations (the Far East, Antarctica, etc.). When transmitting, these stations can have short-term energy requirements up to 2 KW, and the thermal energy liberated is transmitted by means of heat pipes to thermal-storage units, filled with a substance which undergoes a phase transition (octadecane). When the station is receiving, the required power is no more than 100 W. In this case it is necessary to supplement the heating of the battery from the thermal storage by means of heat pipes.

Heat pipes can produce inexpensive temperature control cells for chromatographs, for machinery used in high-speed drying of paper and cloth, for heat-treatment of medicines, and for pasteurization of milk and fruit juice. In the automobile industry, heat exchangers using heat pipes are used to cool oil in the crankcase, for temperature control of liquid in the cooling and transmission systems, for air conditioning of the cabin, and so on.

According to data of the United Kingdom National Engineering Laboratory in Glasgow (East Kilbride), heat pipes and thermosiphons are being used successfully in internal combustion engines to vaporize gasoline accumulating in the lower part of the carburetor in order to supply a uniform mixture to the cylinders. The exhaust gas heat is used for this, and the fuel saving is substantial.

The same laboratory uses heat exchangers employing heat pipes to heat the cabins of automobiles and other types of industrial vehicles, as regenerators to utilize heat, and so on.

Their associates in Wales (in Cardiff) have constructed heat pipes using organic heat-transfer agents, capable of transmitting several kilowatts of thermal power to distances of 3-4 m at an operating temperature of 300°C. The shell of these pipes is stainless steel, and the heat-transfer agent is Thermax (a eutectic mixture of 73.5% diphenyl oxide and 26.5% diphenyl). The wick is a stainless steel mesh. Stainless steel heat pipes are also used with an internal copper shell and a copper wick, and these can operate for more than 10,000 h at temperatures of 250°C and above. This laboratory is currently producing heat-pipe devices operating in the temperature range 50-2500°C. For improved wetting of stainless steel, 0.05% diethyl(sodium sulfonate) succinate is added to the water. This surface-active substance considerably improves the wetting power of the tube surface with regard to water. The Fiat Company is carrying out a wide range of studies to use aqueous heat pipes in the automobile and aviation industries.

To meet the needs of the automobile and tractor industries, the USSR is investigating the parameters of heat exchangers based on heat pipes, the objective being to produce a radiator-temperature controller using gas-generator heat pipes and heat exchangers using oil–water, oil–air, water–gas, etc. There is great interest in using heat exchangers employing heat pipes to heat the oil in the crankcases of tractors and automobiles operating in winter and for cooling or heating the oil in transmissions, for heating the fuel in refueling tanks, and so on.

Heat exchangers based on heat pipes can be used with success to cool the molds in the casting of various engine parts.

Another objective of great interest is to use the cooling effect of heat pipes to construct a cutting tool of increased cutting rate.

Heat exchangers involving thermal siphons are beginning to find use in cooling and freezing equipment.

Abroad, particularly in the USA, there has been great progress in the introduction of heat exchangers based on heat pipes for using and regenerating heat in factories, industrial plants, and in the heating and air-conditioning fields. This type of heat exchanger can be useful for air conditioning in chemical factories, cooling of gas, heating of water and air, and so on. For example, the Q-Dot Corporation has already marketed more than 1000 heat exchangers in which about 125,000 heat pipes are used. It is particularly important and necessary, under the conditions of the energy crisis, to use energy for secondary purpose, following a primary use.

Heat exchangers using heat pipes have appreciable advantages in comparison with circulation-type heat exchangers:

1) They contain no moving parts and are completely self-contained; each heat pipe in a heat exchanger is insulated from the others, i.e., the heat exchanger has very great reliability;

2) they have a guaranteed operating life, there being no need for preventive maintenance (a life of 15-20 years or more);

3) they do not require an external drive to pump the working substance; the temperature drop at the ends of the heat pipe is the source of mass transfer inside the pipe;

4) they guarantee complete isolation of one gas stream from the other;

5) there are no rigorous limits with regard to the geometrical dimensions of the heat exchangers;

6) heat exchangers using heat pipes are more compact than the circulation type;

7) it is possible to change the direction of the heat flux insides these heat exchangers.

Heat exchangers using heat pipes are particularly promising for use in drying technology, e.g., for drying cloth. For example, the use of one heat exchanger in a chamber for drying textiles has utilized up to 70% of the exhaust heat, which is 150-200 kW. A heat exchanger using heat pipes has a dual function: a) It makes use of the energy; b) it cools the working vapor or air from a temperature of 140°C down to the 60°C value necessary for normal operation of the monitoring instrumentation which is checking for contamination of the medium. One very necessary application of this type of heat exchanger is to utilize waste heat in drying lacquer coats in the automobile industry. It is possible to make heat exchangers with a power up to 500 kW with an efficiency of 65-70%. These are also promising for use in the drying of enamel coatings and semiconductor products in ovens with an infrared heat source.

Presses have been made in the form of heat pipes and are already used for pressing glass and plastic vessels, to improve the quality of gas hotplates, and so on.

An interesting application is cryogenic heat pipes using liquid nitrogen for local freezing of skin and human tissue. The Hughes Aircraft Company (USA) has successfully developed and introduced flexible heat pipes in the form of bellows for cooling moving articles, for example, individual parts of machine tools, and so on.

Heat pipes are being used with success in the instrumentation industry to produce precision instruments, such as temperature controllers, viscometers, instruments for measuring thermophysical properties over a wide temperature range, absolute black-body models, and so on.

It should be noted that these developments are novel. For example, it was not known, even in 1974, that one could maintain the surface of temperature control devices to be isothermal to within 10^{-3} deg using heat pipes. At present, such thermostats exist and are capable of temperature control to within ±0.1°C over the temperature range 100-1000°C (for comparison, water and oil thermostats can be used up to 250°C, and thermostats with a fluidized bed can be used up to 600°C).

In the European Atomic Energy Community the principle of action of gas-controlled heat pipes is used to detect microleaks and flaws in flanged joints, used over a wide range of temperature and pressure. This leak-detection method is more accurate and requires considerably less costly instrumentation than the traditional methods.

It is promising and economically attractive to use heat pipes in spacecraft under weightless conditions. At present, an international program is in progress for research on heat pipes and heat exchangers in use under weightless conditions (The International Heat-Pipe Experiment). The participants are the USA, the Federal Republic of Germany, Great Britain, and Holland.

On October 4, 1974, a sounding rocket was launched into space (the Black Brant Sounding Rocket), which carried heat pipes made by the NASA/Goddard Space Flight Center; ESRO; GFW; Hughes Aircraft Company, NASA/Ames;

a) ESRO constructed two aluminum heat pipes of length 885 mm and diameter 5 mm. The wick was a single layer of stainless steel mesh with an artery diameter of 0.5 mm. One pipe was filled with ammonia and the other with acetone. The acetone heat pipe transmitted 8.4 W of power, and the ammonia pipe transmitted 21 W. The heat sink was an aluminum block;

b) GFW (Geselfsehaft für Weltraumforschung) of the West German Ministry of Technology constructed a flat aluminum heat pipe in the form of a disk of diameter 150 mm and a titanium heat pipe of length 600 mm, charged with methanol, with its end face joined to the disk by an aluminum tube. The flat heat pipe was filled with acetone, and the other end was joined to a heat-storage device (a canister with a molten substance – "Eicosane" – with a fusion temperature of 35°C. This system transmitted 26 W of power;

c) The Hughes Aircraft Company constructed two flexible heat pipes made of stainless steel (6.4 mm in diameter; 270 mm in length). The working liquid is methanol, and the wick is a metal mesh;

d) NASA/Ames constructed two stainless steel heat pipes of length 910 mm and diameter 12.7 mm. The liquid is methanol, and the inert gas is nitrogen. The wick is a screw thread on the body, and the artery is a wafer of metallic felt. This kind of artery is insensitive to the presence of noncondensible gas;

e) NASA constructed a cryogenic heat pipe made of aluminum with longitudinal channels of length 910 mm and diameter 16 mm, charged with methanol.

Thus, in the international experiment on October 4, 1974, the organizations NASA/GSFC (Grumman and TRW), NASA/Ames (Hughes), Hughes (Hughes), ESRO (the IKE Institute in Stuttgart), and GFW (Dornier) took part in the testing of heat pipes in space. Of these, Grumman constructed five different groups of heat pipes, and TRW constructed three.

The French National Center for Space Research, CNS, independently of the American and European Space Center (USA), developed and operated a program of space experiments with heat pipes, constructed by the Aerospatiale and SABCA compancies. In November, 1974, the French sounding rocket ERIDAN 214 was launched, carrying a radiator of heat pipes. The aim of the experiment was to verify the operational capability of heat pipes under weightless conditions; to verify that the heat pipes would be ready to operate at the start of a rocket flight; and to select various heat-pipe structures for spacecraft equipment.

Three types of heat pipe were investigated.

1) A curved heat pipe made by SABCA, of length 560 mm and diameter 3.2 mm, made of stainless steel, the filter being a stainless steel mesh, with ammonia as the heat-transfer agent. The transmitted power was 4 W. The pipe was flexible.

2) A heat pipe made by the CENG organization (the atomic center in Grenoble) of length 270 mm and diameter 5 mm, made of copper, with a wick made of sintered bronze powder. The heat-transfer agent was water. The transmitted power was 20 W.

3) A SABCA heat pipe, similar to No. 1, but straight. The transmitted power was 5 W.

The heat sink was a box with a variable-phase fusible substance $T_f = 28.5°C$ (n-octadecane). The energy source was an electric battery with $U = 27$ V. The total weight of the experimental equipment was 2.3 kg.

In addition to the sounding rockets, NASA has used a number of satellites for testing heat pipes, to evaluate the effect of long-term weightless conditions on heat-pipe parameters (the spacecraft Skylab, OAO-III, ATS-6, CTS, etc.).

These investigations point very clearly to positive gains at present, and we can confidently assert that heat pipes will find wide applications in space in the near future. For example, the USA plans to use heat pipes for thermal control and thermal protection of the reusable shuttle and also for the Spacelab space laboratory. For these, the heat-sensitive equipment will be located in boxes or canisters within which the temperature will be held constant by means of heat pipes located in the walls of the enclosure.

In a similar way, the engineering applications of high-temperature heat pipes (based on liquid metals) is being investigated.

In recent years there have been realistic proposals for the use of liquid-metal heat pipes in the atomic energy and instrumentation fields.

For example, heat pipes using potassium and sodium are being used successfully as probes to investigate the properties of various structural materials in fast-neutron reactor cores, and these can remove heat flux up to 10 kW at temperatures of 870-920 and 970-1050°K, respectively. The heat-pipe diameter is 22.2 mm, and the length is 540 mm. Since the pipes are mounted vertically and the liquid returns under gravity, a special type of wick is used to obtain uniform distribution of liquid in the pipe evaporator, in the form of a single layer of mesh, and a spiral imparts swirl to the liquids.

There is a very great potential for sealed gaseous and gasdynamic lasers using metal vapor, operating on the heat-pipe principle and capable of large output power. The evaporator of a liquid-metal heat pipe is a metal-vapor heat source (copper, mercury, silver, etc.). The application of an electrical discharge in the discharge in the transfer region of the heat pipe produces thermodynamic nonequilibrium of the system and leads to laser illumination of the medium. An experimental model of a metal-vapor laser was constructed and investigated in the Los Alamos laboratory in the USA.

The laser has an external diameter of 15 cm, a height of 10 cm, and a wall thickness of 0.65 cm. The working pressure inside the laser is 7 atm. The heat-transfer agent is mercury. The laser radiation is emitted through four quartz windows, which are also used for optical measurements. The quartz windows were mounted on the end faces of four gas-generator heat pipes, attached to the main heat pipe – the laser. These tubes have a diameter of 1.9 cm and a length of 15 cm. An inert gas is used to protect the glass from metal-vapor condensation. A fifth auxiliary vertical heat pipe is used to insert electrodes into the laser and to protect the high-voltage hermetic Teflon seal from the harmful action of the metal vapor. Its diameter is 6.25 cm, and its length is 20 cm. The entire laser is mounted in a container of diameter 60 cm and height 50 cm, through which air is circulated for added protection from the harmful action of the mercury vapor.

Liquid-metal heat pipes are of great practical interest in the construction of new types of engines (the Stirling or the Ericson engines). The main successes in this direction have been achieved in Holland (the Phillips Company at Eindhoven).

Scientists at Phillips have used liquid-sodium heat pipes to connect a chemical energy source (a chemical reactor) to a Stirling engine. An experimental model of the machine, together with the chemical reactor, has already operated for more 2 months without appreciable change in its parameters. The facility is 6 m long, 3 m high, and 2.5 m wide.

The chemical reactor consists of six sections in which a chemical reaction occurs, liberating heat from interaction of lithium with sulfohexachloride, resulting in formation of lithium chloride and lithium sulfide. The energy released is 1200 kW·h at a temperature of 950°C.

Liquid-metal heat pipes are also used as heat sources in large vacuum ovens.

Liquid-sodium heat pipes of 20 kW power have a diameter of 114 mm and length 1.8 m (the evaporator is 0.3 m, the transfer zone is 0.6 m, and the condensation zone is 0.9 m). The pipes are mounted horizontally and heated by methane. The heat flux in the evaporator is 190 kW/cm^2. The operating temperature inside the vacuum oven is 1000°C. The oven is intended for vacuum fusion of metals (steel alloys). When heat pipes are used, the oven can be heated or cooled in 0.5 h (from room temperature to 1000°C). The total melt cycle takes no more than 4 h. The weight of metal fused is 250 kg.

IV. HEAT-PIPE STRUCTURES

The classical heat-pipe structure is a closed, hermetically sealed shell whose inside surface is covered with a layer of porous material filled with liquid, while the remaining space is filled with vapor (see Fig. 1). Heat pipes of this type are used to transfer heat from one region of space to another.

The intensive development of possible use of heat pipes has taken a number of interesting directions: from simple cylindrical pipes to radiators for internal combustion engines of complex shape, from 2-3 layers of metal mesh as a porous structure to multicomponent porous wicks with arteries.

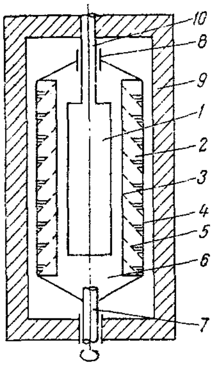
There are presently several tens of different heat-pipe structures. They can be classified according to a number of parameters.

1. According to temperature range: a) cryogenic heat pipes 4°K < T < 200°K; b) low-temperature 200°K < T < 550°K; c) intermediate temperature heat pipes 550°K < T < 750°K; d) high-temperature heat pipes T > 750°K. Heat pipes can operated in the temperature range 4-2800°K.

2. According to type of heat-transfer agent: a) alkali metal agent (used in high-temperature heat pipes); b) nonmetallic agent (water, ammonia, acetone, Freon, alcohol, carbon tetrachloride, benzene, cryogenic liquids), high-temperature organic agents (diphenyl, Dowtherm, sulfur with halogen additives, fused salts, nitrogen tetroxide, etc.).

3. According to heat-technology parameters: a) thermal resistance R = (0.01-1°K/W); b) heat-flux density through the pipe cross section $q = 10\ W/cm^2 - 15 \cdot kW/cm^2$; c) radial heat-flux density $q_r = 0.1\ W/cm^2 - 1.5\ kW/cm^2$; d) thermal activity of the pipe.

4. According to the material of the shell and the wick: a) aluminum heat pipes with staggered mesh-type wick of stainless steel or aluminum cermet; b) copper with a wick made of copper mesh, felt, or ceramic; c) of stainless or ordinary steel with a mesh-type wick made of cermet of the same material; d) of nickel or tungsten–rhenium with a cermet of the same material or a nickel or stainless steel mesh; e) heat pipes with longitudinal channels of different shape (copper, nickel, steel, aluminum) and with grooves covered with mesh; f) heat pipes with a screw thread and a mesh or a cermet (felt) artery of the same or of another material. The



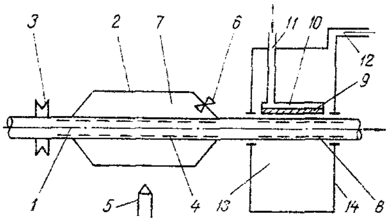


Fig. 6 Fig. 7

Fig. 6. Rotational heat-pipe oven: 1) working volume; 2) isothermal envelope — heat pipe; 3-4) inner and outer pipe walls; 5) perforated connectors; 6) heat-transfer agent; 7) axis of rotation; 8) bearing; 9) thermal insulation; 10) tube for holding specimens of material.

Fig. 7. Rotational heat pipe for heat processing of liquids: 1) channel for liquid; 2) inner pipe wall; 3) actuator; 4) endless screw; 5) heat source; 6) safety valve; 7) vapor phase; 8) endless screw; 9) porous plug for water filtration; 10) reservoir; 11) channel for water supply; 12) tube for pumping out water vapor; 13) vapor vapor space; 14) cooling device.

porosity and the capillary potential of the artery can vary along the pipe axis; g) glass or ceramic heat pipes with glass or silica wicks; h) with elastic polymer inserts (polythene, Teflon, etc.) and elastic silica wicks.

5. According to type of jump for transporting the liquid from the condensation zone to the evaporation zone: a) with capillary wicks; b) gravitational and antigravitational thermosiphons; c) centrifugal; d) electrodynamic, electrokinetic, and electrostatic; e) heat pipes operating on the vapor-layer pump effect; f) operating in an electromagnetic field.

6. According to principle of action: a) true heat pipes; b) heat pipes with variable thermal resistance, particularly gas-generator heat pipes; c) thermal diodes; d) thermal switching devices; e) heat pipes with swirling vapor flow (using a swirl-generating screw); f) with separate channels for the vapor and the liquid; g) with hydrostatic shutters; h) ejector type.

7. According to shape of shell and wick: a) cylindrical; b) flat plate; c) vapor chambers; d) coaxial, annular, and other types of heat pipes.

The length of a heat pipe can be as large as tens of meters, and its diameter can vary from several millimeters up to meters.

V. HEAT-PIPE MODIFICATIONS

The thermal resistance of heat pipes or the transmitted heat flux can be controlled by means of a noncondensible gas by adjusting the flux of vapor of liquid or by means of centrifugal, electric, or magnetic fields. Other means of control are also possible [7].

1. Gas-Controlled Heat Pipes [8-11]. At present, wide use is made of heat pipes with a noncondensible gas for stabilization of surface heating. In addition to the active heat-transfer surface, these have a gas reservoir. The following types of gas reservoir exist:

a) cold reservoir with porous walls;

b) hot reservoir with porous walls, or a constant-temperature reservoir;

c) a hot reservoir without a porous layer;

d) a series combination of several heat pipes with gas reservoirs;

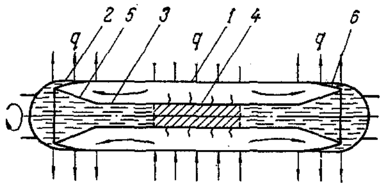


Fig. 8. Rotational heat pipe with an artery: 1) evaporator; 2) condenser; 3) artery; 4) porous part of the artery; 5) coupling region between the artery and the condenser; 6) liquid film in the condensation zone.

e) a reservoir filled with liquid in which a noncondensible gas is dissolved. In these heat pipes conditions for good solubility of the gas in the liquid must be obtained (the Ostwald criterion >1);

f) a gas reservoir in which the wall-temperature change is controlled by proportional change of heat-source temperature by means of an electric field or mechanically;

g) heat diodes with a noncondensible gas.

2. Heat Pipes with Controlled Vapor Flux [12]. Control of the vapor flux in a heat pipe can be achieved by means of a valve located in the vapor channel of the tube and connected to an actuator (a bellows connection with a liquid reservoir in good thermal contact with the heat source).

3. Heat Pipes with Controlled Liquid Flux [13]. The vapor-pressure variation in the heat pipe due to a valve can be used to control the liquid flux through the porous wick of the heat pipe, artificially causing drying of the wick in the evaporation zone. If the temperature of the heat-generating object rises, the valve opens, the vapor pressure in the evaporator decreases, and the liquid passes along the porous wick (or the artery) to the evaporator, cooling it.

A liquid can also be used in a heat pipe to produce heat diodes [14].

4. Centrifugal Heat Pipes. In these, the transfer of liquid from the evaporator to the condenser is accomplished by means of a centrifugal field. Considerably larger heat-flux densities can be achieved in centrifugal heat pipes with boiling of liquid (up to 260 W/cm^2 in water), compared with ordinary pipes having capillary-porous pumps. Centrifugal pipes are found with axial heat transfer [15-18] and radial transfer [19-21]. In the first case there is a continuously moving film of liquid between the condenser and the evaporator. In the second case liquid from the condenser reaches the evaporator in the form of drops, sprayed by the centrifugal field. It is particularly efficient to use centrifugal heat pipes for cooling and temperature control of the rotors of electrical machinery, turbine shafts, and bearings; and to generate efficient gas–gas, gas–liquid, and liquid–liquid heat exchangers.

At present, more than 10 different forms of centrifugal heat pipe are known, with both constant and variable thermal resistance. Unfortunately, the theory lags behind the practical application of heat pipes. The well-known analyses of the operation of these heat pipes are based mainly on the Nusselt theory of heat transfer with condensation of vapor in a laminar liquid film.

Important and interesting questions include the following: interaction of the vapor and the liquid, along with their thermodynamic equilibrium in a centrifugal field; the compatibility of materials; the effect of the centrifugal field on the liquid boiling process; and critical fields.

Figure 6 shows a centrifugal heat pipe developed at the A. V. Lykov Institute of Heat and Mass Transfer, Academy of Sciences of the Belorussian SSR (ITMO), which can be used as an oven for melting metals or polymers; and Fig. 7 shows a heat pipe for heat processing of milk, fruit juice, medicinal preparations, and other heat-sensitive liquids.

There is great practical interest in a centrifugal heat pipe developed at ITMO which has an artery for axial liquid transfer from the condenser to the evaporator (Fig. 8). Supply of liquid from the artery to the evaporator is accomplished by spraying it through a porous insert under the centrifugal field.

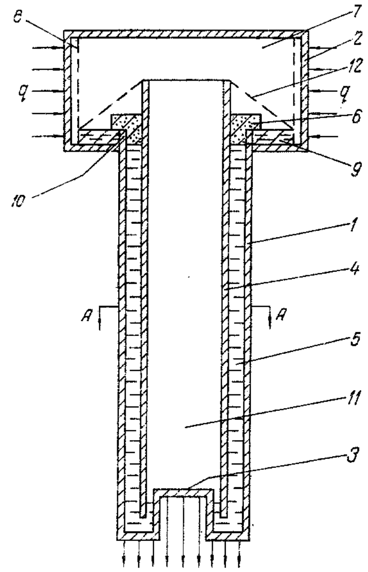


Fig. 9

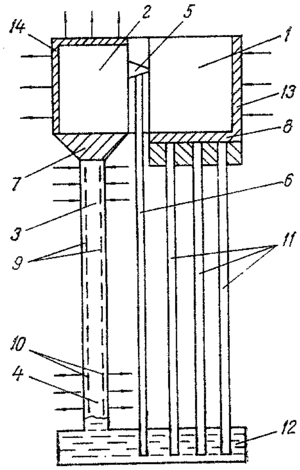


Fig. 10

Fig. 9. Heat pipe with motion of liquid against gravity: 1) body; 2) evaporator; 3) condenser; 4) wall of the vapor channels; 5) liquid layer; 6) porous plug; 7) vapor chamber of evaporator; 8) porous structure of evaporator; 9) transport zone of porous wick; 10) connection of transport zone to porous plug; 11) vapor channel; 12) porous membrane as a water seal.

Fig. 10. Heat pipe with vapor-ejection pump: 1) vapor generator; 2) auxiliary condenser; 3) heat-pipe evaporator; 4) heat-pipe condenser; 5) ejector nozzle; 6) tube to supply liquid to the nozzle; 7) porous water seal above evaporator; 8) porous water seal of the vapor generator; 9, 10) porous structure of heat-pipe wick in the evaporator and condenser; 11) tubes to supply liquid from the liquid collector 12 to the porous water seal of the vapor generator; 13) porous wick of the vapor-generator wall; 14) porous structure of the auxiliary condenser wall.

5. Multicomponent Heat Pipes. In addition to multicomponent heat pipes using liquids or noncondensible gas, there are pipes with two or several liquids [1], either with a mixture of liquids and solid particles [22] (heat pipes with noncondensible gas are described in Para. 1). It is known that by using a mixture of liquids (e.g., water and alcohol) one can frequently solve problems associated with starting heat pipes in a better manner than with a one-component system and can considerably expand the temperature range. For example, heat pipes with a mixture of water and alcohol can be used in the temperature range 160-647°K, while water pipes operate in the temperature range 273-647°K and alcohol pipes, in the range 158-516°K. A peculiarity of two-component heat pipes is that their thermal resistance depends only slightly on temperature [6], while there is a pronounced maximum in the dependence for a one-component pipe.

6. Endless-Screw Heat Pipes. In low-temperature heat pipes, used both in weightless conditions and in a gravitational field, it is often necessary to reduce the thermal resistance. Installation of an endless screw inside the heat pipe is one convenient solution to this problem [19, 29, 30]. The swirling of the vapor flux generated by the screw makes it possible to intensify the heat-transfer process in vaporization and condensation and to keep the noncondensing gas, formed in the pipe as time passes, away from the condensation surface. The endless screw can be porous and has the function of an artery.

It was shown in [19, 28, 31] that the thermal resistance of heat pipes with an endless screw can be less by a factor of 2-3 than that of heat pipes without a screw. The screw can be movable and rotate under the action of the vapor flux, performing mechanical work.

7. Gravitational Two-Phase Thermosiphons and Heat Pipes with Water Valves. Two-phase closed thermosiphons are heat pipes without a porous structure, where the role of the pump to return the liquid from the

condensation zone to the evaporation zone is performed by gravity [24, 25]. This type of heat-pipe structure finds wide use in heat exchangers, devices to maintain the frozen conditions of the soil when oil and gas pipes are installed in permafrost regions, etc. Usually, heat is removed in the heating zone in two-phase thermosiphons by boiling the liquid, although a possible regime is surface evaporation of a liquid film. The condenser is located above the evaporator. Thermosiphons have large thermal resistance compared with heat pipes, and their value lies in the simplicity of fabrication.

In the last 3 or 4 years the so-called "antigravity" thermosiphon has been described in the literature [23, 26]; it uses the "airlift" effect, i.e., the pressure drop in the vapor phase, to return liquid from the condensation zone to the evaporation zone. Antigravity thermosiphons probably include heat pipes with porous evaporators and membranes [27]. In these devices the evaporator is located above the condenser. The distance between them can be 1-2 m.

The ITMO has developed heat-pipe structures which allow heat to be pumped downward to a distance of 1-2 m and more. Figures 9-11 show schematic diagrams of these devices. In the heat pipe (Fig. 9) the evaporator is lined inside with a metal–ceramic or a metal sheet and separated from the liquid channel by a water-seal membrane. Thus, the water seal is separated from the heat-transfer surface. The liquid channel takes the form of a coaxial gap between the vapor channel and the outside wall of the heat pipe. The liquid is pumped from the condenser to the water seal by the pressure gradient of the vapor, due to the temperature difference between the evaporator and the condenser.

Figure 10 shows the construction of a variable thermal resistance heat pipe with a vapor-ejector pump. Control of transfer of liquid from the condenser to the evaporator is achieved by means of a supplementary evaporation device to generate vapor, which moves through the ejector and sucks liquid upward from below.

The variable thermal resistance heat pipe in which liquid is sucked from the condenser to the evaporator by means of an additional evaporation device which supplies vapor to a capillary filled with liquid, and transports liquid upward from below with the help of the airlift phenomenon, is shown in Fig. 11.

8. Electrodynamic Heat Pipes. In these pipes the pump which moves dielectric liquids from the condenser to the evaporator takes the form of a high-voltage electric field between the pipe body and one of several internal electrodes. The great advantages of the electrodynamic heat pipes are the minimum resistance to the flow of moving liquid (there is no porous wick) between the electrode and the wall and the intensified heat transfer when the liquid boils (bubbles of vapor are ejected from the liquid by the electric field) and the vapor condenses. These pipes operate in a stable fashion even when the surface of the electrodes is partially dried. The thermal resistance of the pipe can be varied smoothly by adjusting the electric-field intensity [7, 32-33].

9. Electroosmotic Heat Pipes. An electrostatic field, applied to the porous wick of a heat pipe, made of a dielectric and filled with a dielectric or weakly polar liquid (alcohol), allows the liquid to be pumped from the condensation zone to the evaporation zone [34, 35]. In contrast with electrodynamic heat pipes, electroosmotic pipes can use a lower electric-field intensity.

10. Heat Pipes Using a Magnetic Field. A colloidal solution of magnetosensitive solid particles (ferrites) in a liquid, used as a heat-pipe heat-transfer agent, allows the liquid to be moved from the condenser to the evaporator under the action of a magnetic field. The variable magnetic field causes the liquid flow to swirl, makes it turbulent, and intensifies the heat-exchange process in the evaporation and condensation zones [22].

11. Heat Pipes Using the Airlift Phenomenon and Vapor-Ejector Pumps. The decrease in effective density of a two-phase medium (liquid–vapor), in comparison with a single-phase medium (liquid), can be used successfully to transfer liquid in heat pipes [36]. Usually the two-phase liquid–vapor bubble medium forms when the liquid boils in narrow channels or pipes and the vapor bubbles drive the liquid upward like a piston.

Therefore, by varying the intensity of the heat supplies to the pipe walls, one can control the flux of liquid from the condenser to the evaporator, i.e., one can change the thermal resistance of the heat pipe.

Vapor-ejector pumps are used to raise the liquid up the tubes against gravity. These pumps are also used to remove vapor bubbles or noncondensible gas from the arteries of heat pipes [37]. The principle of action of the pump is that when vapor flows through a slit or a shaped nozzle a local region of reduced pressure is created which causes the liquid to rise in the capillary.

12. Coaxial Heat Pipes. Coaxial heat pipes find wide use as high-grade temperature controllers (temperature constant to 0.001°C) for cooling and thermal control of cylindrical heat-generating elements (fuel elements, crystals, etc.). In these the evaporator and the condenser have different diameters and are mounted

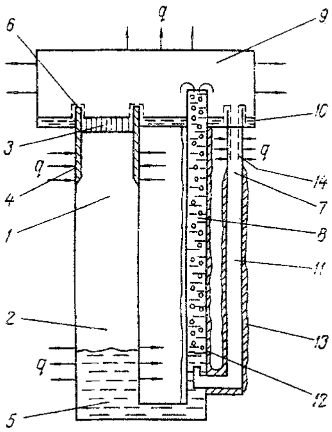


Fig. 11. Heat pipe with the airlift effect: 1, 2) evaporator and condenser; 3) water seal; 4) porous evaporator structure; 5) condenser collector; 6) porous connector; 7) auxiliary vapor generator; 8) two-phase liquid flow, vapor (airlift); 9) auxiliary condenser; 10) layer of liquid in the auxiliary condenser; 11) vapor channel of the vapor generator; 12, 13) thermal insulation; 14) porous structure of vapor generator.

coaxially, and the heat transfer occurs in the radial or radial–axial directions. Transport of liquid from the condenser to the evaporator takes place by means of porous intermediate connections between the capillary structures of the evaporator and the condenser [19, 38-39]. Since the intermediate connectors do not affect the heat-transfer process in the evaporator and the condenser, they can be made of material with low thermal conductivity (fiberglass, porous polymer, etc.).

In addition to ordinary coaxial heat pipes, gas-controlled coaxial heat pipes are used with a reservoir for noncondensible gas [40, 41].

NOTATION

T	is the temperature;
P	is the pressure;
R^*	is the universal gas constant;
Q	is the heat flux;;
c_p, c_V	are the heat capacities;
L	is the latent heat of vaporization;
R_V	is the radius of the vapor channel;
μ, η	are the kinematic and dynamic viscosities;
T_V, ρ_V	are the vapor temperature and density;
r, x	are the radial and axial coordinates;
v	is the vapor velocity;
τ^*	is the vapor flux density through the lateral surface of the vapor channel;
τ_w	is the friction force acting per unit lateral channel surface;
M_V	is the molecular weight of vapor;
R	is the radius of the vapor channel;
β	is the flow momentum coefficient;
k	is the condensation coefficient;
λ	is the thermal conductivity of the liquid;
H_l	is the enthalpy of the liquid;
u	is the velocity of the liquid;
F	is the surface porosity;
θ	is the wetting angle;
σ	is the surface-tension coefficient;
ρ_l	is the liquid density;
K_l	is the permeability of the capillary structure;

w_l is the volumetric liquid flow rate;
b is the width of the capillary structure;
a is the thickness of the capillary structure;
Π is the porosity;
l_p is the effective pipe length;
$l_{a.z.}$ is the length of the adiabatic zone;
l_{ev} is the length of the evaporator;
l_{con} is the length of the condenser;
A is the cross-sectional area of channel;
D_{ef} is the effective channel diameter;
g is the acceleration of gravity;
γ is the angle of inclination of the pipe to the horizontal;
h is the channel height;
2α is the vertex angle of triangle;
a_m^T is the coefficient of thermal diffusion of liquid in the porous body;
ΔU is the concentration gradient;
f is the friction coefficient;
q is the heat flux removed by one evaporator channel;
t is the channel width;
X_{max} is the maximum length of liquid travel in the channel ;
N_l is the parameter describing the liquid properties ($N_l = \rho\sigma L/\eta$);
R_{cr} is the critical bubble radius.

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