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PROGRESS IN HEAT PIPE AND POROUS HEAT EXCHANGER TECHNOLOGY

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Abstract—This is a review of the papers presented at the 1st International Heat Pipe Conference held in Stuttgart, 15–17 October 1973. The review deals with heat pipe application in different branches of technology, heat- and mass-transfer processes in heat pipes, design of variable-conductance heat pipes, optimization of their parameters, operation of heat pipes under weightlessness and in the field of gravity. Main principles of theoretical analysis of energy and substance transfer in heat pipes are presented; centrifugal and coaxial heat pipes, heat pipes with twisted tape, noncondensing gas, electric, magnetic and ultrasonic fields applied, etc. are described.

NOMENCLATURE

v_v, v_l, vapour and liquid flow velocity;
ρ_v, η_v, density and viscosity of vapour;
c, λ_v, heat capacity and thermal conductivity of vapour;
c_l, λ_l, heat capacity and thermal conductivity of liquid;
ρ_l, η_l, density and viscosity of liquid;
P_v, P_l, vapour and liquid pressure;
q, heat flux;
σ, surface tension coefficient;
P_c, capillary pressure;
R_v, vapour channel radius;
L, latent heat of evaporation;
T_v, vapour temperature;
g, free-fall acceleration;
z, r, coordinates;
θ, angle of liquid–vapour interface curvature;
α, angle of tube axis inclination to horizontal;
f, evaporation coefficient;
m, liquid flow;
k_l, porous body permeability;
Π, porosity;
b_l, degree of pore filling;
d, mean pore size;
D, dispersion coefficient;
w_x, convective transfer velocity;
D_m, molecular diffusion coefficient;
w_m, mean superstitial velocity;
d_e, equivalent diameter of channel;
$P_{e.f.}$, pressure drop for liquid motion under external forces in a heat pipe;
P_l, pressure drop in liquid phase;
P_v, pressure drop in vapour phase;
r_e, minimum pore radius in heat pipe evaporator;
l, heat pipe length;
l_a, length of adiabatic zone;
D_W, external diameter of wick;
δ_0, groove width;
δ_1, distance between grooves;
δ, wetted width of channel;
2α, angle of threading.

FOR THE last few years new possibilities appeared for application of heat pipes and vapour chambers in various fields of technology. Low thermal resistance of heat pipes and possibility of transmitting thermal power with slightest losses made it possible to successfully use heat pipes both to design a number of new devices and apparatus and to improve old ones.

So, the use of low-temperature heat pipes in electric machines, for cooling rotors and stators of engines, generators as well as transformer windings allowed increase of their electric power by 30–50 per cent [1–3]. The heat pipes are successfully used for cooling high-voltage high-power switchers, etc.

Low-temperature heat pipes are also resultive when used in various apparatuses and devices to flatten temperature fields and provide isothermal conditions for material treatment. Heat pipes, for example, make it possible to improve treatment of plastic materials in extruders, stamping of glass and plastic crockery [4–5], casting of metal (aluminum) parts, etc. Heat pipes with variable thermal resistance are successfully used to flatten temperature fields in internal combustion engines of cars and tractors. Great prospects are suggested by application of low-temperature heat pipes in petroleum and gas industry.

Use of heat pipes to flatten temperature fields and to increase heat-transfer rates in press moulds, iron moulds, stamps, in pistons of internal and external combustion engines (Stirling and Erickson engines) furnaces for crystal growth, dryers for thermolabile materials (medicines, etc.) thermal treatment of liquids, pastes, etc. is especially promising.

Heat pipes may successfully be used together with mirror paraboloids for concentration, accumulation and utilization of solar energy [6].

The use of different heat-transfer agents allows effective utilization of heat pipes as heat conduits at different temperature levels (Table 1) [7–10].

Table 1

Working fluid	Axial heat flux (W/cm^2)	Radial heat flux (W/cm^2)	Working temperature of pipe (°C)
NH_3	200–400	5–15	20–40
H_2O	1000–1500	25–100	150–250
Ka	3000–5000	150–250	700–750
Na	5000–15000	200–1400	850–950

The dimensions of heat pipes range from millimeters to hundreds of meters.

Heat pipes can be made as minute needles used, for example, for freezing tumors and eye-ball operations or as power-transmission lines up to 100 m long. There exist liquid–sodium evaporator–condenser systems with return of heat-transfer agent into the evaporator under the field of gravity designed for transport of hundreds of kilowatts.

Minute regulated heat pipes whose thermal resistance varies by tens and hundreds of times may be used for design of commutators, automatic devices and highly reliable electronic computers which may operate for 10–15 yr without repair and maintenance. It is planned to use heat pipes in nuclear reactors and MHD-generators.

Heat pipes may also be used:

1. For spacing of power sources from power sinks.
   (a) cooling electronic equipment on the earth and in space;
   (b) cooling bearings in engines, turbines and generators;
   (c) in heat exchangers;
   (d) for temperature stabilization and temperature control.
2. For transmission of large heat fluxes with small temperature drops.
3. For transformation of heat flux density.
4. For flattening temperature fields and temperature peaks.
5. For maintaining constant temperatures at variable heat fluxes.

Great prospects are suggested by application of low-temperature heat pipes to space engineering [11–13]. The main trend of investigations is thermal control of equipment with the aid of low-temperature heat pipes of variable thermal conductance, temperature stabilization of rocket tanks, living modules of spaceships. Of great interest are cryogenic heat pipes used for cooling detectors of infrared and nuclear radiation [14–16].

For many spaceships strict temperature control and stabilization are necessary while external heat fluxes and internal power sources may considerably change their own parameters.

In space, however, heat pipes may be used not only as temperature control elements. They may serve as heat shields for leading edges of reentry vehicles [17].

It would be a mistake to think that capillary-porous heat exchangers involving phase transitions are promising only for space applications. Still greater possibilities for their application are afforded by different branches of industry particularly by nuclear engineering [18–19].

Capillary-porous heat exchangers and heat pipes involving phase transitions possess a number of advantages as compared to traditional ones, for example, convection heat exchangers: they have no movable parts, they are noiseless, do not require power for pumping working fluid from the evaporator to condenser, possess high heat conductance compared to metal rods of the same sizes and have small weight.

These are the most important properties of capillary-porous materials:

1. Capillary suction potential characterizing the process of liquid interaction with capillary walls.
2. Permeability for gas or liquid.
3. Wide ranges of effective transfer coefficients (for example, effective thermal conductivity depending on whether temperature and concentration field directions in the material are the same or opposite). The effective thermal conductivity of capillary-porous materials may range by 8 orders from 10^{-4} to 10^4 W/m.deg.
4. High specific surface area.
5. Low density (compared to monolyth of which the porous material is made).
6. Electrokinetic effects in liquid flows over capillary-porous materials (flow potential).
7. Selectivity, that is permeability for one fluid and impermeability for other, permeability for heat fluxes in one direction and impermeability in the reverse direction.

8. Possibility of realizing phase transitions in pores involving power generation or absorption, etc.

Nowadays a great number of heat pipes of various designs are known. They may conventionally be classified as follows:

1. Wick heat pipes of constant or variable thermal resistance.
2. Thermosyphones with smooth inner walls, or with partial porous coating of constant or variable thermal resistance.
3. Rotatory heat pipes with a centripetal effect.
4. Other kinds of heat pipes (EHD-pipes, those with magnetic pumping, involving electric ultrasound and other fields).

Since the geometry of heat pipes may be very diverse, they should rather be called closed-type evaporation-condensation devices.

Theoretical analysis of the operation of heat pipes of each of the above kinds should naturally differ from others. No single well-composed theory of heat pipe operation can be developed.

Theoretical investigations of heat pipes follow the two lines:

1. The maximum of energy transmitted is determined based on the analysis of dynamics of vapour and liquid transfer, ultimate characteristics of the velocity of vapour transfer (sound velocity), vapour flow interaction with the surface of liquid film in a porous wick, onset of liquid boiling crisis in the evaporator of the porous wick.
2. Thermal resistance of heat pipes or temperature field along their external surfaces are determined as a function of heat flux density and heat-transfer conditions at the external surface in the evaporator and condenser.

The energy and substance transfer under conditions of a laminar compressible vapour flow inside a cylindrical heat pipe is mathematically formulated as follows [20] (Fig. 1)

Vapour transfer equations

$$\operatorname{div} \mathbf{v} = 0; \quad \rho_v \mathbf{v}(\nabla \mathbf{v}) = -\nabla P_v + \eta_v \nabla^2 \mathbf{v} \tag{1}$$

$$c_p \rho_v \nabla T_v = \lambda_v \nabla^2 T_v . \tag{2}$$

Energy transfer equations for a porous wick

$$c_l \Pi \rho_l w_l \nabla T_l = \operatorname{div} \mathbf{q} \tag{3}$$

$$q = \Lambda \operatorname{grad} T$$

where Λ is the thermal conductivity tensor of the capillary-porous wick saturated with liquid, since the thermal conductivity λ of the wick in the axial direction is different from λ in the radial direction

$$\nabla^2 = \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial}{\partial r}\right) + \frac{\partial^2}{\partial z^2}.$$

The system of equations (1–3) should be supplemented with a heat-conduction equation describing energy transfer in the heat pipe envelope as well as an equation of liquid filtration through the capillary-porous wick.

The maximum capillary pressure in the porous wick with effective pore size r_{eff} is

$$P_{c\,\max} = \frac{2\sigma \cos\theta}{r_{\text{eff}}}. \tag{4}$$

At steady-state conditions the momentum component of a vapour phase is directed along the normal to the porous wick surface in the evaporation and condensation regions that follows from the equality

$$P_v + \rho_v v_v^2 = P_l + \rho_l w_l^2 + \sigma k' \tag{5}$$

where k' is the curvature of the liquid–vapour interface. Equation (5) holds for any two points in the evaporator and condenser.

In most cases an approximate solution to the problem stated is only possible. As a result we have determined:

1. Pressure drop along a porous wick with liquid flow.

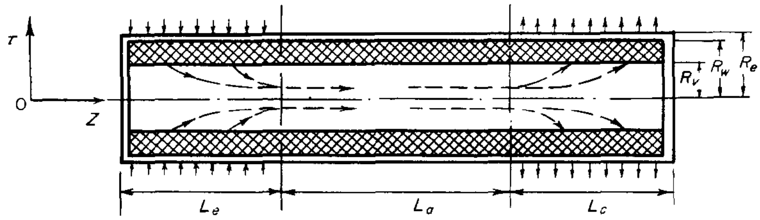

FIG. 1. Cylindrical heat pipe: L_e, length of evaporator; L_a, length of adiabatic region; L_c, length of condenser; R_e, external tube radius; R_w, external wick radius; R_v, radius of vapour channel.

2. Pressure drop in a vapour duct.

3. The minimum $\Delta(\sigma k')$ as a function of two variables corresponding to the coordinates of two neighbouring points 1 and 2.

$$\Delta(\sigma k') = \Delta P_v - \Delta P_l + \Delta(\rho_v \mathbf{v}_r^2) - \rho_l \Delta \mathbf{w}_r^2. \quad (6)$$

4. The condition

$$\max\{\Delta P_v - \Delta P_l + \Delta(\rho_v v_r^2) - \rho_l \Delta w_l^2\} \leqslant P_{c\,\max}. \quad (7)$$

The vapour flow conditions depend on the heat flux, dimensions and geometry of vapour duct, boundary conditions in the evaporator and condenser. The vapour flow may be either laminar or turbulent, and vapour is considered compressible or incompressible. Compressible vapour allows substantial drops in a vapour phase. The transition from a laminar to a turbulent flow may be characterized by the critical Reynolds number Re_{cr}.

Unfortunately, there are few data reported for Re_{cr}. Re_{cr} of 1250 may approximately be adopted. For estimation of the vapour flow velocity the Mach number in the adiabatic region of a heat pipe is also used

$$Re = \frac{QR_v}{\pi R_v^2 L \eta_v}; \quad M = \frac{Q}{\pi R_v^2 L P_v} \sqrt{\left(\frac{C_v}{C_p} RT_v\right)} \quad (8)$$

where R is the universal gas constant.

Depending on the kind of a wick, fluid transfer in porous wicks is described either by the Darcy or by the Poiseuille laws, fluid flow being assumed laminar.

In addition to the field of capillary forces, the fluid transfer is affected by the field of gravity. The expression for ΔP_l should therefore be supplemented by the term $\rho_l gz \sin \alpha$.

The kinetics of liquid–vapour phase transition, i.e. the rate of evaporation and condensation may be determined using the Kn number. The liquid temperature may differ from the vapour temperature (temperature drops). The maximum heat flux at evaporation from a flat surface into vacuum is defined by

$$q_{\max} = f \frac{LP_l}{\sqrt{(2\pi RT_l/\mu_l)}}.$$

In a number of cases the transfer of energy and substance in heat pipes may be accompanied by partial drainage of a porous wick. Thus, designing of heat pipes should be based on equations of fluid and vapour flow dynamics, kinetics of phase transitions at the liquid–vapour interface, energy transfer in capillary-porous materials.

STRUCTURAL CHARACTERISTICS OF POROUS WICKS IN CALCULATIONS OF CONVECTIVE TRANSFER IN POROUS MATERIALS

A liquid flow in porous materials may be governed by the Darcy law

$$m = \rho_l w_l = -\frac{k_l}{\eta_l} \operatorname{grad} P \quad (9)$$

where k is the permeability.

By integrating equation (9) we obtain the pressure drop at two points of the wick. The permeability k depends on the porosity Π of the wick and the degree of saturation

$$k_l = f(\Pi, b_l) \quad (10)$$

where b_l is the degree of pore filling.

At high heat fluxes the degree of saturation of the porous wick is changing along the pipe, i.e. it depends on coordinate.

Porous materials may be described by the following parameters:

1. Pore size d.

2. Linear dimension L of the porous body. Within this dimension $\langle w \rangle$ is averaged over the volume.

The mean bulk value of some function ψ of a tensor type that characterizes liquid in porous materials is described by the relationship

$$\langle \psi \rangle = \frac{1}{V} \int \psi \, \mathrm{d}V \qquad d \ll L \quad (11)$$

where L is the characteristic length of the volume v

$$\langle\langle \psi \rangle\rangle = \langle \psi \rangle \quad (12)$$

that implies that the mean of a mean is a mean.

The basic averaged relationship characteristic of the liquid in porous materials is of the form

$$\langle \operatorname{grad} \psi \rangle = \operatorname{grad} \langle \psi \rangle + \frac{1}{V} \int_{S_i} \psi \mathbf{n}' \, \mathrm{d}S \quad (13)$$

where surface integration is carried out along the liquid surface S_j at the liquid–vapour interface in the bulk of the wick; $\mathbf{n}'$ is the unit vector along the normal to the element surface dS.

Equation (13) is necessary for derivation of the liquid motion equation in porous materials.

3. The third characteristic of porous materials is the differential surface permeability curve $f(k)$ (similar to the pore radius distribution curve).

The curve $f(k)$ for homogeneous material may be presented as the δ-function of k or a linear combination of these functions

$$f(k) = \sum_{i=1}^{N} \delta A_i \quad (14)$$

where A_i is the constant satisfying the equation

$$\sum_{i=1}^{N} A_i = 1$$

where N is a finite value.

If $f(k)$ cannot be expressed by a finite number of functions δA_i, the material is inhomogeneous. If $f(k)$ involves only one term, the material is homogeneous. When the material is described by two or more functions $f(k)$, it is heterogeneous.

In a general case $f(k)$ depends on the rectangular coordinates $x_i (i = 1, 2, 3)$ and angular coordinates θ and ψ

$$P(k_1 \leqslant k \leqslant k_2) = \int_{k_1}^{k_2} f(x_i, \theta, \psi) \mathrm{d}x_i \mathrm{d}\theta \mathrm{d}\psi. \quad (15)$$

If $f(k)$ is independent of θ and ψ, the porous material is isotropic.

MASS TRANSFER

Different mass-transfer mechanisms in porous materials should be distinguished: (1) molecular diffusion; (2) turbulent transfer; (3) local nonuniformities and fluctuations due to nonisotropic structure; (4) recirculation due to local pressure fluctuations; (5) liquid absorption with walls; (6) viscous, diffusional and thermal slip, etc.

All these mass-transfer mechanisms may be characterized by a dispersion term.

Under isentropic conditions mass transfer is governed by a differential equation of a diffusion type which incorporates a convective term

$$\frac{\partial \omega}{\partial \tau} = D \frac{\partial^2 \omega}{\partial z^2} - w_z \frac{\partial \omega}{\partial z} \quad (16)$$

where D is the dispersion coefficient; w_z is the convective term characterizing the liquid velocity defined by the Darcy law.

The coefficient D is found from the Taylor–Aris formula

$$D = D_m + \frac{d_e^2 w_m^2}{192 D_m} \quad (17)$$

where D_m is the molecular diffusion coefficient; w_m is the mean superstitial velocity; d_e is the equivalent diameter of the duct.

The structural parameter d_e is

$$d_{\text{eff}} = D w^{-n} \quad (18)$$

where n is constant $(1 \leqslant n \leqslant 2)$.

The Taylor–Aris theory is applicable to the case when the dimensionless time τ^* is less than a definite value described by the inequality

$$\tau^* \leqslant 0{\cdot}6 Pe_D^2 [192 + Pe_D^2]^{-1} \quad (19)$$

where Pe_D is the diffusional Peclet number.

For a turbulent flow through porous materials mass-transfer equation is of the form

$$\frac{\partial \omega}{\partial \tau} + v_i \frac{\partial \omega}{\partial z_i} = \frac{\partial}{\partial z_i}\left(D_{ik} \frac{\partial \omega}{\partial z_k}\right); \quad ik = 1, 2, 3 \quad (20)$$

where the dispersion coefficient $D_{i,k}$ is a second-order tensor. If a liquid flow obeys the Darcy law, then D is proportional to the liquid velocity averaged over the characteristic length L. In a general case the characteristic length L is a fourth-order tensor $(L_{i,j,k,l})$ rather than a scalar value.

For a one-dimensional flow

$$D_{11} = \lambda_1 \mathbf{w}_m; \quad D_{22} = \lambda_2 \mathbf{w}_m^2 \quad (21)$$

where λ_1 and λ_2 are the components of the diffusional tensor for isotropic porous material, $\mathbf{w}_m$ is the mean velocity.

Diffusional equation (16) may be written as

$$\frac{\partial \omega}{\partial \tau} = -\frac{\partial}{\partial z}\left(D \frac{\partial \omega}{\partial z} - w_z \omega\right) \quad (22)$$

with the assumption of the velocity w_z independent of z.

The first term of equation (22) describes molecular diffusion, the second term describes convective mass transfer.

Application of the Darcy formula

$$m = \omega w_z = -\frac{k_l}{\eta_l} \frac{\partial P}{\partial z} \quad (23)$$

yields instead of (22)

$$\frac{\partial \omega}{\partial \tau} = D \frac{\partial^2 \omega}{\partial z^2} + \frac{k_l}{\eta_l} \frac{\partial P}{\partial z^2}. \quad (24)$$

With the use of formula (23) we may write the equation of liquid filtration through porous materials as

$$\frac{v_l}{k_l} \nabla^2 \bar{w}_l - \nabla P - \beta w_l = 0 \quad (25)$$

where β is the additional characteristic of the porous material.

INVESTIGATION ON KINETICS

The study of vapour transfer mechanism for a free-molecular flow (kinetic mass-transfer region) in a thin capillary (micro-capillary) with regard for the kinetics of liquid evaporation from the capillary wall under the temperature gradient is described in [30]. Here the account for surface diffusion on the capillary wall has been taken and the cases of specular and diffusional reflection of molecules from the capillary wall have been considered.

It has been found that inside the capillary partially filled with liquid evaporation and condensation may proceed simultaneously. S^* (the product of dimensionless heat of evaporation by the relative temperature drop along the capillary wall) and l^* (capillary length to radius ratio) are the main characteristic parameters. An approximate estimation of S^* for the evaporation into vacuum ($T_0 = 220°K$, $T = 10°K$) gives $S^* = 1{\cdot}3$. In the capillary, in this case, at $S^* = l^*$ and $l^* > l$ (capillary length is greater than its radius), vapour condenses at the liquid meniscus and evaporates near the exit at the capillary walls. However, if $l^* < l$ (capillary length is less than its radius), then at the same conditions of $S^* = l^*$ liquid evaporates at both capillary ends. Thus, the pore structure (l^*) essentially affects the mechanism of vapour migration. The value l^* is always positive ($l^* > 0$), while S^* may be either positive or negative depending on the sign of the temperature drop ΔT along the capillary ($S^* \gtrless 0$). If $S^* < 0$, then at $l^* = -S^*$ the migration mechanism will differ from the migration mechanism at $l^* = S^*$. The direction of the temperature gradient, therefore, affects evaporation and condensation in the capillary. This phenomenon is observed only at nonisothermal conditions and may be called a thermal effect of vaporous moisture migration in the capillary. Successive condensation and evaporation in capillary-porous bodies was considered earlier as well, but these processes were regarded to proceed at isothermal conditions. In the authors' case moisture evaporation and condensation in the capillary depends on the temperature drop along its length, and geometry of pore and capillaries of the material. In the same work [30], the kinetics of vapour transfer in a porous body is considered with regard for diffusional reflection of molecules. An equation of molecular transfer has been obtained with account for the finite velocity of mass propagation. Thus, using the molecular-kinetic method, a diffusion equation has been obtained with account taken of the finite velocity of mass propagation which leads to a hyperbolic diffusion equation. Hyperbolic heat and mass-transfer equations may describe liquid motion in capillary-porous bodies at nonisothermal conditions [31]. Thermomolecular pressure difference is known to exist at effusion of one-component gas through a thin capillary. With account taken of surface diffusion and adsorption time, we may consider that in gas mixture a selective motion of gas mixture components through a porous body takes place.

The study of transfer phenomena in capillary-porous bodies with anisotropic or homogeneous structure showed the effect of effusion anisotropy [32]. The anisotropic effusion effect strongly depends on the capillary radius increasing with its decrease. The effect of anisotropic effusion is determined by the criterion $Gr = D_s\tau/R^2$ where D_s is the coefficient of surface diffusion; R is the capillary radius and τ is the time of adsorption.

LIMITATIONS IMPOSED ON HEAT FLUXES TRANSFERRED ALONG HEAT PIPES

1. *Capillary sorption limit*

The capillary sorption limit restricts the possibilities of improving the dynamics of liquid and vapour flows in heat pipes. An increase in a heat flux should result in increasing capillary pressure, due to pressure drop in liquid and vapour phases with the circulation rate.

In a general case, the capillary suction limit for a heat pipe of a certain design is set by integrating the equation for the pressure drop along the pipe and by comparing the total pressure losses at all of the points with the local maximum capillary pressure. Since the maximum capillary pressure and external forces (gravitation) do not usually depend on the heat flux, the capillary suction limits may be described as follows

$$\Delta P_{c\,\max} - \Delta_{e.f} \geqslant \Delta P_l + \Delta P_v. \tag{26}$$

The heat pipe should operate normally until the heat flux achieves the value at which the r.h.t. of the above equation becomes equal to the l.h.s. This is valid for any point along the heat pipe.

The principle of the pressure balance within possible limits of capillary suction works for different kinds of heat pipes with several evaporators and condensers and porous wicks of different geometries. There is no need to numerically calculate $\Delta P_{k\,\max}$ in a number of cases. Thus, for a classical cylindrical pipe with uniform heat flux in an evaporator and condenser and with no pressure drop due to inertia forces, the following analytical solution to estimate $\Delta p_{k\,\max}$ may be obtained

$$\Delta P_{c\,\max} = \frac{2\sigma\cos\theta}{r_l} = \frac{Q(l+l_a)\mu_l}{2\pi k\rho_l L(R_W^2 - R_v^2)} + \frac{4Q(l+l_a)\mu_v}{\pi\rho_v LR_v^4} + \rho_l g[D_W\sin\theta + l\cos\theta]. \tag{27}$$

The solution for a maximum heat flux transferred via the pipe results in the expression

$$Q_{c\,\max} = \frac{\pi L}{(l+l_a)} \cdot \frac{[\Delta P_{c\,\max} - \rho_l g(D_W\sin\theta + l\cos\theta)]}{\left[\dfrac{\mu_l}{2k\rho_l(R_W^2 - R_v^2)} + \dfrac{4\mu_v}{\rho_v R_v^4}\right]}. \tag{28}$$

2. *Vapour–liquid interaction limit*

In ordinary classical heat pipes vapour and liquid move in opposite directions. At large velocities they, therefore, start interacting that leads to liquid drag and wave formation. This may be accompanied by entrainment of liquid droplets by the vapour flow, so the droplets entrapped by a vapour flow can never enter the evaporator. This will result in the decreased transferred power. The droplets, however, will be involved in liquid circulation and will increase the pressure drop along the pipe.

Thus, the maximum heat flux transmitted along the pipe will not correspond to the product of the liquid flow by the latent heat of evaporation. It will be smaller by the amount of liquid entrained by the vapour flow.

The conditions of drop separation during vapour and liquid interaction in terms of the ratio of inertia forces of the vapour to liquid surface tension forces depend on the Weber number

$$W = \frac{\rho_v V_m^2 z}{\sigma} \tag{29}$$

where ρ_v is the vapour density; V_m is the mean vapour velocity; σ is the surface tension coefficient; z is the characteristic dimension that refers to the liquid surface.

Some experimentators concerned with vapour–liquid interaction in mesh wicks at $We = 1$ report that the characteristic dimension z is equal to the diameter of the wire. This implies that the interaction may be decreased by using a fine mesh. Other authors consider the distance between wires a better characteristic diameter z.

Heat pipes with open grooves are especially sensitive to vapour–liquid interaction.

When We of 1 is taken, the ultimate axial heat flux at which liquid drops entrainment by the vapour flow starts can be found from

$$q_x = \left[\frac{\rho_v \sigma L^2}{z}\right]^{1/2} \tag{30}$$

where q_x is the heat flux in the axial direction; L' is the latent heat of evaporation.

At high temperatures the surface tension of liquid decreases considerably thus decreasing the limit of liquid–vapour interaction. At the stagnation point σ and L are vanishing.

3. *Limit due to sound velocity*

There is an analogy between the flow in a heat pipe of a uniform cross-section with mass injection (evaporation) and removal (condensation) and a constant mass flow in a convergent–divergent nozzle. The end of the evaporator section of a heat pipe is at the neck of the nozzle. In case of a limit due to the sound velocity ($M = 1$) in nozzles, similar restriction to the vapour velocity should be at the exit of the heat pipe evaporator.

For the given temperature at the evaporator exit and chosen working fluid, this condition of shock wave generation imposes a fundamental restriction to axial heat transfer. In order to increase the axial heat flux in heat pipes, the cross-sectional area of the vapour channel should be increased.

The sound limit is calculated by assuming the vapour flow velocity in the continuity equation to be equal to the sound velocity

$$q_x = L\rho_v V_s \tag{31}$$

where V_s is the sound velocity.

When using this formula for prediction of the sound velocity limit in a flat pipe, the parameters in a shock wave region (at the evaporator exit) should be found. However, it is often more convenient to calculate this limit with the known parameters at the initial length of the evaporator. This is possible with the use of the Levy equation

$$q_x = \frac{L\rho_v V_s}{\sqrt{[2(k+1)]}} \tag{32}$$

where

$$k = \frac{C_P}{C_V}.$$

4. *Heat transfer in heat pipes*

Evaporation or boiling as well as condensation in heat pipes proceed with heat supply or removal through the wall. Temperature drop at the place of heat supply and removal is a dominant one in a heat pipe. Temperature drops are also known to exist in a vapour phase and at the liquid–vapour interface, but they are negligible against that in a porous wick. In many cases in heat pipes heat is transferred through a porous wick by conduction in the evaporator and condenser. Convective heat transfer inside a porous wick is small compared to heat conduction. In heat pipes temperature drop is therefore proportional to the local heat flux, thickness of a porous wick and inversely proportional to the effective thermal conductivity of a porous wick and wick material.

Heat may be transferred in the presence both of surface evaporation of liquid out of the wick pores and of boiling in pores. Liquid boiling in pores imposes restrictions on the power transmitted through the heat pipe due to onset of boiling crisis. In heat pipes vapour is in a state of saturation. In the evaporator liquid is overheated against vapour because of the curvature of the liquid–vapour interface in pores with a radial temperature field. The maximum overheat region lies at the junction between the wick and heat pipe envelope.

In case of great overheat, the incipient boiling inside porous material is possible.

When porous wicks allow steady-state boiling, the mechanism of heat transfer becomes more complicated and essentially large heat fluxes may be achieved. A heat-transfer limit in this case is determined by the onset of boiling crisis inside the porous body. Unfortunately, since the structure of the porous material changes the fluid dynamics of vapour and liquid, here it is difficult to make use of the mathematical apparatus that describes the onset of pool boiling crisis. In porous wicks with prescribed pore radius distribution (metal ceramics, sintered powders) incipient boiling causes liquid recession from the heated wall and heat is transferred through a thin vapour film on this surface [21]. The thickness of liquid film is, presumably, independent of the wick thickness. Its value approaches the particle diameter consisting the wick. In accordance with the theory, constant heat conduction through thin vapour film is survived even with bubble formation inside porous wick. In this case the mechanism of boiling liquid transfer inside porous material that is peculiar for nucleate boiling is of no importance.

With this model adopted, we should obtain a constant heat-transfer coefficient α for liquid "boiling" in porous material.

The maximum heat flux takes place when the temperature drop necessary for vapour bubbles to be carried away exceeds the pressure developed by capillary forces and required to preserve liquid film.

1. ARTERIAL HEAT PIPES

Consider some particular cases of heat pipe application to different branches of technology. Sometimes heat pipes may successfully be made in liquid–liquid or liquid–gas heat exchangers, the pipes may be either vertical or inclined that allows not only capillary forces but also a field of gravity to be used for transfer of a heat-transfer agent inside the pipe. As to operation in a vertical position, artery heat pipes with triangular grooves on the inside walls of the evaporator and condenser seem promising. In such heat pipes liquid is transferred from the condenser to evaporator via the artery under gravity forces while the grooves serve to pump liquid from the wall into the artery in the condensation zone and to distribute liquid along the pipe periphery in the evaporation zone. All this provides the efficient operation of the pipe. Thermal resistance of such pipes is practically independent of the heat power and is less significant than in smooth-walled thermosyphons. Due to different driving forces affecting working fluid in the grooved section and in artery, normal operation of vertical heat pipes requires that the following relations be fulfilled simultaneously

$$\begin{aligned} \Delta P_l^e &\leqslant \Delta P_{\mathrm{cap}}^e \\ \Delta P_l^c &\leqslant \Delta P_{\mathrm{cap}}^c \qquad (33) \\ \Delta P_l^a &\leqslant \Delta P_g. \end{aligned}$$

It is assumed in further considerations of heat transfer in artery heat pipes that (i) triangular grooves are trended on the pipe walls; (ii) heat flux density in evaporator and condenser is constant; (iii) depth of the grooves is much less than the diameter of the heat pipe; (iv) temperature drop over the wick in the radial direction is quite small for the temperature dependence of liquid properties to be neglected; (v) there is no effect of the evaporator on the condenser; (vi) pressure losses in vapour phase motion are neglected; (vii) liquid motion along the artery is governed by the Darcy law; (viii) the capillary potential is quite small compared to the gravity potential; (ix) pipe is operating with evaporation.

Consider the case when the groove with δ_0 is larger than the liquid–vapour surface curvature in the artery ($r_a \cos\alpha$).

If the condition $\delta_0 > r_a \cos\alpha$ is fulfilled in the condenser, the equations describing liquid motion in a groove should be written (Fig. 2) for the two sections: (1) from $\pi R/2$ to l_0 where motion occurs at a constant wetted width of the groove and (2) from l_0 to $l = 0$ where the wetted width of the groove is variable.

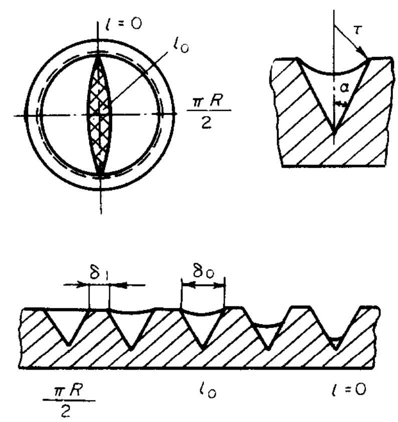

FIG. 2. Cross-section and profile of grooves in vertical artery heat pipe.

The mass flow rate m at $q = \mathrm{const}$ decreases proportionally to l, whence

$$m(l) = m_0\left(1 - \frac{2l}{\pi R}\right); \qquad m_0 = \frac{q(\delta_0+\delta_1)\pi R}{4L}. \quad (34)$$

This mass flow rate is characteristic of the liquid flow cross-section from l_0 to $l = 0$ with the area

$$S_l = \frac{\delta^2(l)}{4}\left[\tan\alpha + \cot\alpha - \frac{(\pi/2-\alpha)}{\cos^2\alpha}\right]\cos\alpha \quad (35)$$

and the hydraulic diameter

$$d_h(l) = \frac{\delta(l)}{2}\left[\tan\alpha + \cot\alpha - \frac{(\pi/2-\alpha)}{\cos^2\alpha}\right]$$

$$\cos\alpha = \frac{\delta(l)}{2}c(\alpha). \quad (36)$$

Since in practical cases the hydraulic diameter is quite small for the liquid flow in the groove to be considered laminar, in accordance with the Poiseuille law we may write

$$\Delta P_l = -\frac{32m(l)h_l}{S_l(l)d^2(l)\rho_l}.\Delta l. \quad (37)$$

From equations (34)–(37) we obtain

$$\Delta P_l = -\frac{128q(\delta_0+\delta_1)\pi R\left(1-\frac{2l}{\pi R}\right)\eta_l}{\delta^4(l)L\rho_l c(\alpha)}.\Delta l. \quad (38)$$

The capillary pressure from l_0 to $l = 0$ depends on the change of the meniscus curvature. Since this is caused by recession, then

$$\Delta P = \frac{\sigma\cos\theta}{r^2}\Delta r. \quad (39)$$

We may get from (39)

$$\Delta P = \frac{4\sigma\cos\theta}{\delta^2(l)}.\frac{\partial\delta}{\partial l}.\Delta l; \quad \Delta r = \frac{\partial\delta}{\partial l}f(\alpha)\Delta l$$

$$r^2 = \frac{\delta^2(l)}{4}f(\alpha). \quad (40)$$

By substituting expressions (38) and (40) into (33) we write an equation which expresses the dependence of the wetted groove width on the coordinate

$$\delta^2(l)\,\mathrm{d}\delta = K(\delta_0+\delta_1)(\pi R-2l)\,\mathrm{d}l \quad (41)$$

where

$$k = -\frac{32q\eta_l}{\sigma L\rho_l c(\alpha)\cos\theta}. \quad (42)$$

Integration of (41) with regard for $l = l_0$ at $\delta(l) = \delta_0$ yields

$$\delta^3(l) = \delta_0^3 - 3k(\delta_0-\delta_1)(l_0-l)(\pi R-l_0-l). \quad (43)$$

As liquid moves along the groove to the artery, meniscus decreases but it cannot become less than the mean curvature of the meniscus of the liquid–vapour interface in the artery. Thus, at $l = 0$, $\delta(l) = r_a$ and from equation (43) we may write

$$k = \frac{\delta_0^3 - r_a^3\cos^3\alpha}{3(\delta_0-\delta_1)l_0(\pi R-l_0)}. \quad (44)$$

By equating right-hand sides of equations (42) and (44), we obtain an expression for the maximum heat flux of the condensation surface

$$q^c_{\max} = \frac{(\delta_0^3 - r_a^3\cos^3\alpha)\cos\theta c(\alpha)}{96(\delta_0+\delta_1)l_0(\pi R-l_0)}.\frac{\sigma L\rho_l}{\eta_l}$$

where

$$c(\alpha) = \left[\tan\alpha+\cot\alpha-\frac{\left(\frac{\pi}{2}-\alpha\right)}{\cos^2\alpha}\right]^3\cos^2\alpha. \quad (45)$$

Equation (45) includes l_0 depending on the transfer processes at the section from $\pi R/2$ to l_0. To find l_0, we consider the relationship between forces in this section of the grooves. To write the relationship, the following simplifying assumption is made: the cross-sectional area l of the flow and hydraulic diameter within the section from $\pi R/2$ to l_0 are constant and equal to their arithmetic mean at these two cross-sections. Proceeding from the above, an expression for pressure drop due to friction forces will be written as

$$\Delta P_l = -\frac{32m_m\left(\frac{\pi R}{2}-l_0\right)\eta_l}{S_l d_h^2\rho_l} \quad (46)$$

where

$$S_l = \frac{\delta_0^2}{4}\left[\cot\alpha+\tan\alpha-\frac{\left(\frac{\pi}{2}-\alpha\right)}{\cos^2\alpha}\right]\cos\alpha \quad (47)$$

$$d_h = \frac{\delta_0}{2}\left[\cot\alpha+\tan\alpha-\frac{\left(\frac{\pi}{2}-\alpha\right)}{\cos^2\alpha}\right]\cos\alpha \quad (48)$$

$$m_m = \frac{m_0}{2}\left(1-\frac{2l_0}{\pi R}\right). \quad (49)$$

Using (47–49), we shall rewrite (46) as

$$\Delta P_l = -\frac{128q\left(\frac{\pi R}{2}-l_0\right)^2\eta_l(\delta_0+\delta_1)}{\delta_0^4\rho_l c(\alpha)}. \quad (50)$$

The capillary pressure from l_0 to $\pi R/2$ is expressed as

$$\Delta P_c = \frac{2\sigma\cos\theta\cos\alpha}{\delta_0}$$

From equations (49, 50), we obtain an expression for

$$l_0 = \frac{\pi R}{2} - \sqrt{\left(\frac{\sigma\delta_0^3.\cos\theta.\cos\alpha\rho_l Lc(\alpha)}{64Q(\delta_0+\delta_1)\eta_l}\right)}. \quad (51)$$

Then the maximum heat flux at the heat pipe condenser may be calculated from the equation

$$q^c_{\max} = \frac{(3\delta_0^3\cos\alpha+2\delta_0^3-3r_a^3\cos^3\alpha)c(\alpha)\cos\theta}{24\pi^2R^2(\delta_0+\delta_1)}\frac{\sigma L\rho_l}{\eta_l}. \quad (52)$$

The liquid condensed at the surface of a condenser is sucked by the artery where from it moves into the evaporator under the gravity forces.

From the Darcy law the mass velocity of liquid along the artery may be written as

$$m = \frac{\rho_l S_a k}{\eta_l}.\frac{\mathrm{d}P_l}{\mathrm{d}h} = \frac{q}{L}; \quad \frac{\mathrm{d}P_l}{\mathrm{d}h} = -\rho_l g. \quad (53)$$

Since motion proceeds under the action of gravity forces, the maximum transmitted power which depends on the transport properties of the artery will be written as

$$q_{\max} = \frac{\rho_l^2 k L S_a g}{\eta_l}. \quad (54)$$

The condensed liquid is pumped along the artery into the evaporator and is distributed via the grooves over its inner wall.

Three variants are possible of heat pipe filling with liquid when $\delta_0 > r_a \cos\alpha$; $\delta_0 = r_a \cos\alpha$; $\delta_0 < r_a \cos\alpha$.

The first two variants are omitted in the paper.

At $\delta_0 < r_a \cos\alpha$ the liquid flow along the condenser grooves will proceed at the constant wetted width of the groove, with the radius of the meniscus curvature ranging from ∞ to $r_a \cos\alpha$. Then, instead of equations (47)–(49), we have

$$S_l = \frac{\delta_0 r_a \cos\alpha}{2} \quad (55)$$

$$d_h = \frac{r_a \cos^2\alpha}{2} \quad (56)$$

$$m = \frac{m_0}{2}. \quad (57)$$

The use of the above expressions may give the value of $q^k_{\max}$ in the form

$$q^c_{\max} = \frac{r_a^2 \cos\theta \cos^2\alpha}{4\pi^2 R^2(\delta_0 + \delta_1)} \cdot \frac{\sigma\rho_l L}{\eta_l}. \quad (58)$$

We shall consider transfer processes in the evaporator.

With the assumption of $\delta_0 \ll r_a \cos\alpha$, after simple manipulations we obtain an expression for the maximum heat flux at the second section

$$q^e_{\max} = \frac{\delta_0^3 \cos\alpha \cos\theta \,.\, c(\alpha)}{64(\delta_0 + \delta_1)(\pi R - l_0)\left(\dfrac{\pi R}{2} - l_0\right)} \cdot \frac{\sigma\rho_l L}{\eta_l}. \quad (59)$$

Since maximum heat fluxes obtained for each of the sections are equal, $q_{\max}$ can be found from

$$q_{\max} = \frac{3{\cdot}26 \times 10^{-2} \delta_0 \cos\alpha \,.\, \cos\theta \,.\, c(\alpha)}{(\delta_0 + \delta_1)\pi R} \cdot \frac{\sigma\rho_l L}{\eta_l}. \quad (60)$$

In practice different ratios between the groove width and the artery pore size for the evaporator and condenser may be encountered in one and the same pipe. In this case the heat pipe may be designed based on the above equations.

Experimental investigation of vertical artery heat pipes [22] showed that the above devices allow effective cooling of different apparatuses.

2. COAXIAL HEAT PIPES

Power transport in heat pipes in radial direction when evaporator and condenser are arranged coaxially are of great interest [23, 24, 10]. Extremely tempting is the designing of an isothermal envelope around a tank in which some technological process requiring constant temperature occurs.

Thus, for example, growing of monocrystals should be performed under isothermal conditions. Isothermal conditions are desirable in melting furnaces, combustion chambers of nuclear reactors [23–26, 10]. Figure 3 furnishes longitudinal section of a coaxial heat pipe. Regular porous partitions in an interpipe space connect the evaporator and condenser and allow fluid motion from the region of condensation into the region of evaporation.

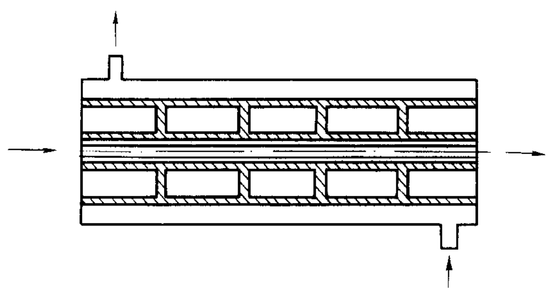

FIG. 3. Coaxial heat pipe with liquid heat exchangers.

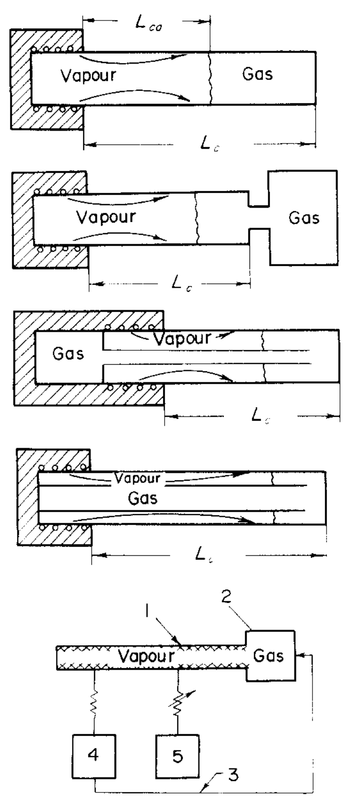

FIG. 4. Different kinds of gas-controlled heat pipes: 1, heat pipe; 2, gas tank; 3, feedback; 4, heater; 5, heat sink.

Since porous partitions do not effect heat transfer either in evaporation or in condensation zone, they can be made of material of low effective thermal conductivity but possessing higher permeability than the porous structure in the evaporator or condenser.

Also, they can be made of electric insulation material which allows electric potential between the evaporator and condenser of a heat pipe.

3. HEAT PIPES WITH NONCONDENSABLE GAS

Heat pipes of variable thermal resistance and particularly heat pipes with noncondensable gas are of great importance in practice [11–13, 18]. Some of the schematic diagrams of the heat pipes with noncondensable gas are plotted in Figs. 5 and 6.

The heat pipes considered may be used either as temperature regulators or as heat pipes of one-directed action (heat diodes). The principle of the heat pipe operation is that power is supplied to a part of the pipe, liquid evaporates, vapour displaces the noncondensable gas into the cold zone that results in a plug which prevents the vapour from penetrating into the heat-transfer surface. The space occupied with the plug is proportional to the saturated vapour pressure, i.e. it depends on its temperature. By varying the temperature of the saturated vapour, the heat-transfer surface of the condenser of the heat pipe may be increased or decreased. At present heat pipes-thermal regulators are provided with a hot wickless tank of noncondensable gas or cold tank with walls covered with porous wick [26].

4. CENTRIFUGAL HEAT PIPES

In centrifugal heat pipes liquid from the condenser is returned into the evaporator by a centrifugal field. Centrifugal heat pipes may be subdivided into two classes, namely, centrifugal pipes with axial power transfer [2, 3] and radial transfer (coaxial heat pipes) [9, 10, 23].

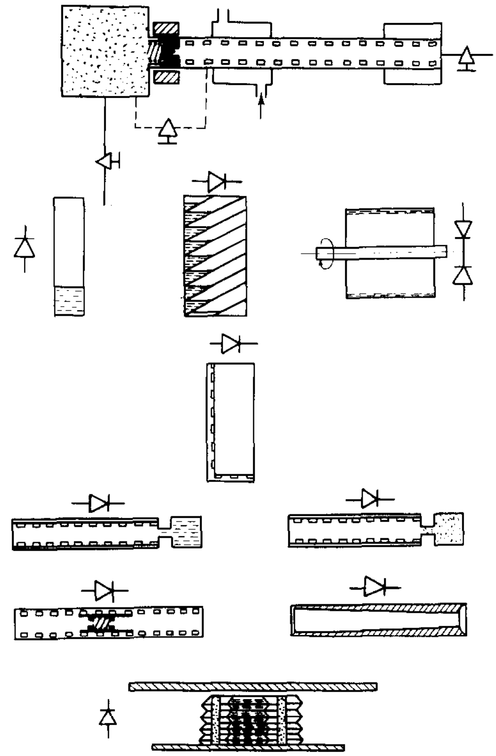

FIG. 5. Heat diodes.

Since the efficiency of a centrifugal pump is essentially higher than that of a capillary one, the boiling crisis in the evaporator and heat transfer in the condenser is one of the factors controlling the maximum value of the heat power transmission.

It should be pointed out that in the condenser heat transfer is better at the coaxial centrifugal heat pipe than near the axial one due to spraying of condensate with the centrifugal field, which decreases the liquid film thickness on the heat-transfer surface.

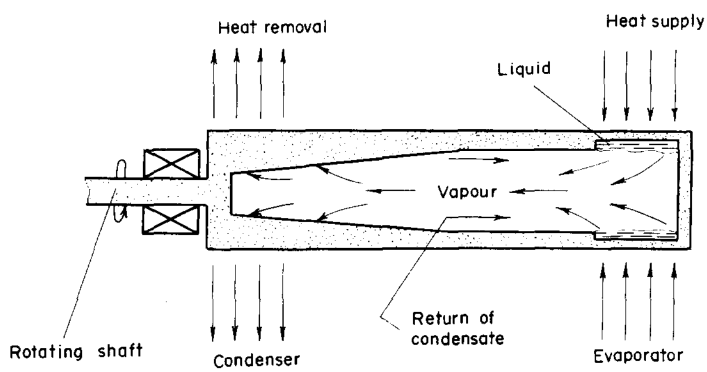

FIG. 6. Centrifugal heat pipe with axial heat transfer.

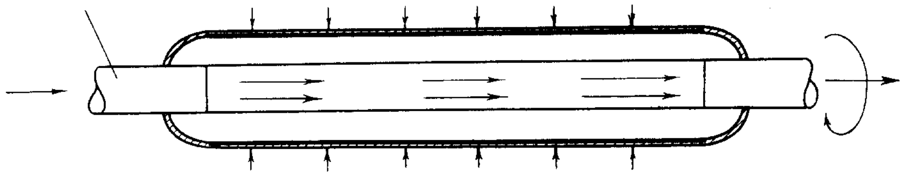

FIG. 7. Centrifugal coaxial heat pipe with radial heat transfer.

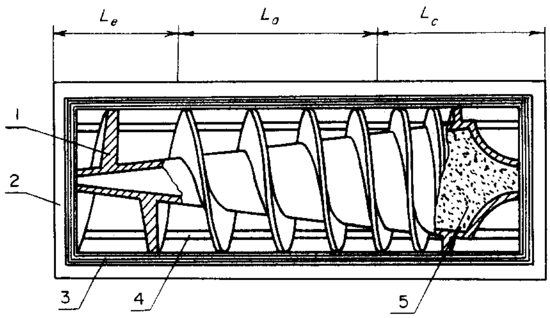

FIG. 8. Heat pipe with twisted-tape vapour swirler: 1, blades of twisted tape; 2, pipe casing; 3, porous wick; 4, artery; 5, central artery inside twisted tape.

Centrifugal heat pipes allow transfer of a heat flux which is several times larger than that in heat pipes of capillary structure.

5. HEAT PIPES WITH ELECTRIC, MAGNETIC AND ULTRASONIC FIELDS

In literature, the principle of operation of electrohydrodynamic heat pipes is described [27]. In these pipes liquid from a condenser to evaporator is transferred under an electrostatic field applied to a narrow slot between two walls with the liquid moving along.

The applicability of the pipes of this type is limited because of their complex manufacture.

Magnetic field [28] for pumping electrically conducting or magnetically susceptible liquid as well as ultrasonic field [10] allowing intensification of liquid suction in a porous field may be used as pumps.

Heat pipes may be used as direct transformers of heat power into electric power [29] when a vapour flow inside the pipe provides the medium for electric power transport. Such electric gasodynamic transformers may be of great interest for the future power engineering.

6. HEAT PIPES WITH VAPOUR FLOW SWIRLING BY MEANS OF TWISTED TAPE OR SWIRLERS

In low-temperature heat pipes for heat-transfer intensification in the evaporator and condenser, twisted tapes (Fig. 7) or other kinds of vapour flow swirlers which impart rotary motion to the flow vapour inside the tube, may be used [22, 10].

Vapour flow swirling in the condenser allows intensification of vapour condensation and involves rotation of liquid film if such exists on the surface of porous wick.

Rotational motion of a vapour flow makes it possible to force noncondensing gas, formed in the heat pipe, back from the surface of condensation.

In the heat pipe evaporator a swirled vapour flow improves heat release through boiling by pressing the liquid film to the heating surface, promotes separation of vapour and liquid drops ejected by bubbles from the porous wick. A twisted tape may be used as an artery for axial transfer of liquid and its distribution along the heat pipe perimeter. The experiments described in [22] showed that the thermal resistance of the heat pipe with a twisted tape may be considerably less than the thermal resistance of a heat pipe without a twisted tape.

Other kinds of vapour flow turbulizers and swirlers may be used instead of a twisted tape.

CONCLUSION

At present there are tens of heat pipe designs with different mass flow drivers. Simple designs, high efficiency and low cost of manufacturing provide their wide application in various branches of technology.

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PROGRES TECHNOLOGIQUES DANS LES CALODUCS ET LES ECHANGEURS DE CHALEUR POREUX

Résumé—C'est une analyse des mémoires présentés au premier congrès international sur les caloducs, 15–17 octobre (1973). La revue traite de l'application du caloduc à différentes branches technologiques, procédés de transfert de chaleur et de masse dans les caloducs, conception de caloducs à résistance variable, optimisation de leurs paramètres opératoires du point de vue de la légèreté et dans le champ de gravité. On présente les principes fondamentaux de l'analyse théorique du transfert d'énergie et de matière dans les caloducs; on décrit les caloducs avec des rubans torsadés, des gax incondensables, des champs imposés électriques, magnétiques et ultrasoniques, etc...

FORTSCHRITT IN DER TECHNOLOGIE DES WÄRMEROHRES UND DER WÄRMEÜBERTRAGER AUS PORÖSEN STOFFEN

Zusammenfassung—Dies ist ein Überblick über die Vorträge von der 1. Internationalen Wärmerohrkonferenz in Stuttgart vom 15–17 Oktober 1973. Der Überblick behandelt unterschiedliche Anwendungsgebiete des Wärmerohres, Wärme- und Stoffaustauschprozesse in Wärmerohren, den Entwurf von Wärmerohren mit variablem Widerstand, die Optimierung der Wärmerohrparameter, sowie die Funktion von Wärmerohren in der Schwerelosigkeit und im Schwerefeld.

Die Grundzüge der theoretischen Analyse von Energie- und Stofftransport in Wärmerohren werden angegeben. Zentrifugale und koaxiale Wärmerohre, Wärmerohre unter Anwendung von gewickelten Bändern, nicht kondensierendem Gas, elektrischen, magnetischen und Ultraschallfeldern usw. werden beschrieben.

ПЕРСПЕКТИВЫ ИСПОЛЬЗОВАНИЯ ТЕПЛОВЫХ ТРУБ И ПОРИСТЫХ ТЕПЛООБМЕННИКОВ В НОВОЙ ТЕХНИКЕ

Аннотация — Статья является обзором работ, представленных на I Международную конференцию по тепловым трубам, состоявшуюся 15–17 октября 1973 г. в г. Штуттгарте, ФРГ.

Основная тематика обзора посвящена использованию тепловых труб в различных отраслях техники, процессам тепло- и массообмена в тепловых трубах, созданию тепловых труб переменного термического сопротивления, оптимизации их параметров, использованию тепловых труб в условиях невесомости и в поле тяжести.

В обзоре изложены основные принципы теоретического анализа процессов переноса энергии и вещества в тепловых трубах, описаны различные виды тепловых труб, такие как центробежные, коаксиальные, трубы со шнеками, с наличием неконденсирующегося газа, использованием электрических, магнитных, ультразвуковых полей и т. д.