REVIEW

CONTROLLED HEAT PIPES

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Controlled heat pipes (CHP) is the name given to closed autonomous evaporator–condenser devices whose thermal resistance varies due to external forces (heat flux, electrical field intensity, variation in cross section of the vapor channel, critical liquid boiling in the evaporator, etc.).

There are several types of CHP's: gas-controlled (containing noncondensible gas), pipes with control of vapor flux, liquid flux, electrodynamic, electrokinetic (electro-osmosis effect), etc.

The most widespread type at present is the gas-controlled heat pipe (GCHP).

The CHP's hold great promise for heat-control processes, maintenance of constant temperatures in heat-generating elements, and prevention of transient over-temperatures in cooled elements. For example, using GCHP's one can manufacture thermostats with temperature control accuracy of ±0.002°C [1]. A thermostat with a GCHP can be built in the temperature range 80-1200°K.

The CHP's are being successfully used both in gravity fields and under weightless conditions.

Heat pipes (HP) with variable thermal resistance and heat diodes can be used for temperature control, both independently and associated with traditional thermal control systems such as circulation heat-transfer systems, supplementary heaters, louvered dynodes, etc. The principal heat-transfer agents in HP's for thermal control systems are water, methanol, ammonia, Freons, and acetone. The main interest at present is in creating cryogenic HP's of variable thermal resistance for cooling IR sensors and optical systems. For these HP's the heat-transfer agents used are cryogenic liquids (methane, ethane, nitrogen, oxygen, hydrogen, and helium).

Variable-resistance heat pipes (controlled heat pipes) have a number of advantages compared with traditional thermal control systems: they can be flexible and have different geometry; they can be used to separate the heat source and sink by large distances; to automatically maintain a heat-source temperature constant with a variable load; and for one-directional heat transfer.

The capillary structures of CHP's can vary widely, depending on a number of factors (the amount of heat transmitted, the heat-pipe geometry, the temperature range, the orientation in a gravity field, the presence of fields, etc.). Figure 1 shows some types of capillary structures for CHP's.

The capillary structure of a controlled heat pipe is chosen to meet the following requirements: required heat flux q_{max}, transmitted along the pipe; temperature drop between the evaporator and the condenser ($T_e - T_c$); temperature range ($T_h - T_{cr}$), within which the tube is capable of transmitting heat; the amount of liquid in the pipe; the reliability and lifetime of operation; the cost; the technical manufacturing details; the possible use for different functions; etc. Among the most common capillary structures used in CHP's, one finds metallic mesh, cermet, felt, capillary channels in the pipe body (longitudinal and helical cuts), and also composite capillary systems [helical cut with a lattice of arteries or porous plates made of cermet (constant or variable permeability), of metallic felt, with

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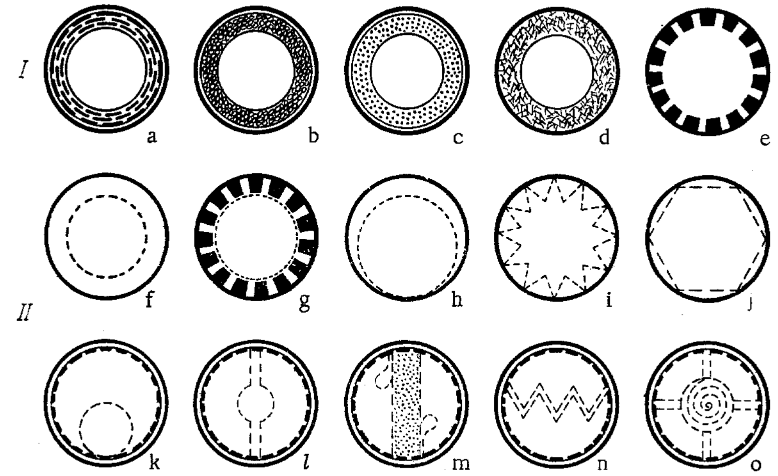

Fig. 1. Schematic illustrations of first generation wicks: a) mesh type; b) capillary apertures in a metal layer; c) sintered spherical metal particles; d) metallic felt; e) axial channels; second generation (heat transfer through a transport zone); f) coaxial gaps; g) axial channels, covered by capillary material; h) asymmetric channels; i) horizontal capillary wick; j) polygonal-type wick; third generation (heat transfer and transport of heat-transfer agent are separate); k) asymmetric arterial type; l) symmetric, arterial type; m) wick with arteries made of porous cermet with additional voids for excess heat-transfer agent; n) with arteries made of corrugated capillary material; o) tunnel wick.

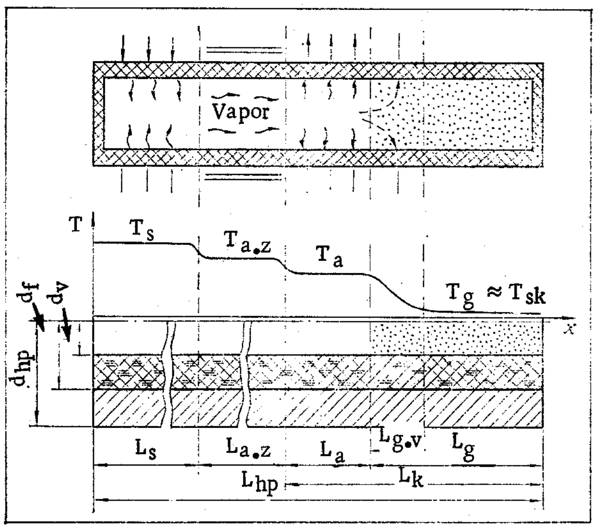

Fig. 2. Simplified scheme for a gas-controlled heat pipe and the temperature distribution along the external surface.

spiral, corrugated or tunnel arteries, etc.]. A detailed appraisal of the merits and defects of these and other capillary systems with relevance to CHP's has been given in [2].

The capillary structures for CHP's mentioned above can be divided into two groups. The first group of capillary structures, insensitive to noncondensible gases, contains longitudinal channels in the pipe body, and bundles of mesh, metallic felt and cermet, but these capillary structures have low permeability as capillary pumps. The second group contains arterial structures (lattice and tunnel arteries, etc.). They have a large liquid pumping rate, but are very sensitive to bubbles of gas inside the arteries. To overcome these difficulties a scheme has been proposed with an ejector [3] (a Venturi tube) and also composite wicks, e.g., porous plates located along the center of a channel with arteries on the surface.

Figure 2 shows a simplified scheme for a gas-controlled heat pipe and the temperature distribution along the external surface.

I. Theoretical Investigations of Heat- and Mass-Transfer Processes in GCHP

In one of the first papers describing experimental investigations of heat pipes [4], the authors met the effect of accumulation of noncondensible gas in the cold part, i.e., the condenser blocking effect. This effect is regarded as undesirable in the ordinary heat pipe, since the blocked zone increases the temperature drop between the evaporator and the condenser.

Cotter [5] was the first to study the usefulness of this fact, and he noted that the length of the blocked zone was proportional to the mass and the average temperature of the gas and inversely proportional to the pressure. The idea was suggested in [6] of controlling the heat transmission of a pipe containing a noncondensible gas and acting as a heat pipe with variable conductivity.

An analysis of the heat- and mass-transfer processes in the gas-controlled heat pipe was first made by Bienert [7]. Under the assumption of a relatively sharp vapor—gas interface, he suggested that the vapor temperature depended on the transmitted flux, the sink temperature, and the length of the condenser active zone as follows:

$$T_v = T_v\,(Q,\ T_{sk},\ V_g). \tag{1}$$

The variation of the pipe temperature is found by solving the differential equation in the form

$$\partial T_v = \frac{1}{1+S}\left[\left(\frac{\partial T_v}{\partial Q}\right)dQ + \left(\frac{\partial T_g}{\partial T_{sk}}\right)dT_{sk} + \frac{S}{\frac{(\partial \ln P_v)}{\partial T_g}}\left\{\frac{\partial T_g}{T_g} + \frac{dm_g}{m_g} - \frac{dV_g}{V_g}\right\}\right], \tag{2}$$

where

$$S = \frac{\partial Q}{\partial x}\cdot\frac{\partial T_v}{\partial Q}\cdot\frac{\frac{\partial}{\partial T_v}(\ln P_v)}{-\frac{\partial}{\partial x}(\ln V_g)} \tag{3}$$

is the temperature stabilization coefficient.

From analysis of the dimensionless parameter S one can conclude that the temperature stabilization factor will be higher with more intense heat transfer $\partial Q/\partial x$ and also with a larger value of $(\partial/\partial x)(\ln V_g)$, i.e., to maintain F_v/V_g at a minimum one must reduce the channel cross section or introduce an additional reservoir with noncondensing gas.

In order to evaluate the thermal stabilization of different gas-controlled heat-pipe structures Bienert introduced the concept of the temperature sensitivity factor

$$\sigma^* = \frac{\partial Q}{\partial T_v} + \frac{\partial Q}{\partial x}\cdot\frac{\frac{\partial}{\partial T_v}(\ln P_v)}{-\frac{\partial}{\partial x}(\ln V_g)}. \tag{4}$$

Marcus [8] introduced a one-dimensional model, accounting for the effect of partial pressure vapor in the blocked zone of the pipe, with type III boundary conditions on the condenser wall. Several types of heat pipe were examined, with wick-type and nonwick-type gaseous reservoirs. For temperature stabilization of GCHP's one must calculate the required ratio of reservoir volume to volume of the condenser section of the heat pipe. For a wick-type reservoir this ratio can be written in the form

$$\frac{V_r}{V_c} = \frac{\xi_1 - \xi_2}{\xi_3}, \tag{5}$$

where

$$\xi_1 = \left[1 - \frac{Q_{\min}}{L_c h F' (T_v^{\min} - T_{sk}^{\min})}\right]\left[\frac{P_{v.a}^{\min} - P_{v.sk}^{\min}}{T_{v.a}^{\min}}\right],$$

$$\xi_2 = \left[1 - \frac{Q_{\max}}{L_c h F' (T_v^{\max} - T_{sk}^{\max})}\right]\left[\frac{P_{v.a}^{\max} - P_{v.sk}^{\max}}{T_{sk}^{\max}}\right],$$

$$\xi_3 = \left[\frac{P_{v.a}^{\max} - P_{v.r}^{\max}}{T_r^{\max}} - \frac{P_{v.a}^{\min} - P_{v.r}^{\min}}{T_r^{\min}}\right].$$

If the reservoir is at the heat sink temperature T_{sk}, Eq. (5) takes the form

$$\frac{V_c}{V_r} = \left[\frac{P_{v.a}^{\max} - P_{v.sk}^{\max}}{P_{v.a}^{\min} - P_{v.sk}^{\min}} \cdot \frac{T_{sk}^{\min}}{T_a^{\max}} - 1\right]. \tag{6}$$

In order to reduce the effect of various changes in sink temperature on the thermal sensitivity factor, very often provision is made for good thermal contact of a wick-free reservoir and the evaporator, i.e., $T_r = T_p$. For this case Eq. (5) takes the form

$$\frac{V_c}{V_r} = \left[\frac{P_{v.a}^{\max} - P_{v.sk}^{\max}}{P_{v.a}^{\min} - P_{v.sk}^{\min}} \cdot \frac{T_{sk}^{\min}}{T_{v.s}^{\max}} - \frac{T_{sk}^{\min}}{T_{v.a}^{\max}}\right]. \tag{7}$$

Equations (5)-(7) can be used to calculate the gaseous reservoir volume if the temperature sensitivity factor or the allowable fluctuation in vapor temperature is specified.

It was shown in [8] that variations of pressure P and temperature T are subject to the Clausius–Clapeyron law

$$\frac{d(\ln P_{v.a})}{dT_{v.a}} = \frac{r^* M}{R_0 T_{v.a}^2} = K \frac{1}{T_{v.a}^2}. \tag{8}$$

Here K is the sensitivity factor for gas control.

In deriving the theoretical relations the interface is assumed to be infinitely small. Because of its simplicity this model has enjoyed widespread use in the design of gas-controlled heat pipes. However, for a correct analysis of the heat- and mass-transfer processes one should take into account that the vapor–gas interface region is quite extensive. Thus, one should consider not a boundary, but a vapor–gas interface region, and an increase in the width of this region worsens the temperature-stabilizing characteristics of the GCHP. The first independent models with a planar front to model the real situation were demonstrated in [9].

The cause of broadening of the interface region is energy and mass transfer in this region between the vapor and the gas and also axial transport of energy along the wall and the wick [10–12].

Articles [10, 13, 14] have studied the effect of axial heat conduction on the behavior of the vapor–gas front and the thermal stabilizing characteristics of the GCHP.

Kazakov et al. in [13] observed a deterioration in thermal stabilization due to transfer of heat along the pipe wall. To allow for axial heat conduction of the wall and the wick, the second factor in Eq. (3) has the form

$$\frac{\partial T_v}{\partial Q} = \left[\frac{x}{R_{ext} + R_{int}} + \frac{R_{int}\sqrt{\lambda F R_{ext}}}{(R_{ext} + R_{int})^2} \cdot \frac{\exp\frac{x}{\sqrt{\lambda F R_{ext}}} - \exp\frac{2L - x}{\sqrt{\lambda F R_{ext}}}}{\exp\frac{x}{\sqrt{\lambda F R_{ext}}} + \exp\frac{2L - x}{\sqrt{\lambda F R_{ext}}} - A(x)}\right], \tag{9}$$

where

$$A(x) = \frac{R_{int}}{R_{ext} + R_{int}}\left[\exp\frac{x}{\sqrt{\lambda F R_{ext}}} - \exp\frac{2L - x}{\sqrt{\lambda F R_{ext}}}\right] \operatorname{ctg} \sqrt{\frac{R_{ext} + R_{int}}{\lambda F R_{ext} R_{int}}} \cdot x. \tag{10}$$

It has been shown that the calculated values of S from Eq. (9) differ from the theoretical values by 15%, while the error may be as high as 100% if axial heat conduction is not accounted for.

The authors of [10] analyzed the heat- and mass-transfer processes in the gas—vapor front region. It was shown that axial heat conduction of the pipe body plays a more significant role in determining the wall temperature, and also in determining the heat-transfer characteristics in the condenser of gas-filled heat pipes, than does diffusion of mass in the axial direction. The diffusion of vapor mass through the noncondensible gas determines the rate of condensation of vapor on the cold reservoir walls, if $T_{sk} < T_{hp}$.

For a number of GCHP's of large diameter which use liquid metals as heat-transfer agents, the diffusion energy transfer can overcome the axial heat conduction of the walls, as was shown in [12, 14]. We are interested in a parameter to evaluate the mechanism for heat transfer in the vapor—gas front region [12]. The ratio of the heat transfer due to the latent heat of vaporization of the diffusing vapor to the conductive heat transfer along the wall has the form

$$\varkappa_1 = \frac{F_V \rho D r^*}{F_W L_W} \cdot \frac{dm}{dT}. \tag{11}$$

The ratio of the energy transmitted by diffusion to the energy transmitted by heat conduction of the vapor—gas mixture in the interface region is

$$\varkappa_2 = \frac{\rho D r^*}{\lambda} \cdot \frac{dm}{dT}. \tag{12}$$

Because the one-dimensional model does not describe the processes associated with interdiffusion of vapor and gas quite accurately, several attempts have been made [11, 12, 15] to solve the two-dimensional problem.

The first numerical solution of the two-dimensional problem of heat transfer in the vapor—gas front region was made by McDonald [11]. However, the model examined, which had a toroidal evaporator and condenser, was not put to practical use.

Results were presented in [15] of a solution of the equations of motion and diffusion for a boundary layer. Since the problem was solved for a model involving a one-directional permeability condition, i.e., diffusion of vapor into the gas volume was neglected, the solution has a limited area of use. A second assumption that the vapor—gas region is isothermal does not correspond to reality.

Results were presented in [12] of a numerical solution of the two-dimensional problem of steady heat and mass transfer in the vapor—gas front region. The results show that in some cases there are considerable axial and radial temperature gradients in the vapor—gas region. The assumption made that there is no conductive heat transfer along the wall limits the range of usefulness of the results. A large number of papers [16-26] have been devoted to experimental investigation of the parameters of gas-controlled heat pipes.

At present the main directions of experimental investigation are as follows: 1) investigation of specific processes of heat and mass transfer occurring in GCHP's; 2) the investigation of new materials; 3) the building of new devices; 4) investigation of the thermo-control parameters of heat pipes in various temperature ranges (with various heat-transfer agents). We shall give details on each of these topics below.

1. The majority of gas-controlled heat pipes operate with a blocked region of temperature below the triple point of the heat-transfer agent. There is therefore a considerable danger of condensing vapor of the heat-transfer agent on the walls of the heat pipe in the blocked region, which may lead to drying out of the evaporator.

Marcus et al. [27] have investigated this phenomenon. Using an original experimental technique based on a weight method, considerable condensation rates were detected. For example, for a water gas-controlled heat pipe with a transfer power of 55 W and a sink temperature of 283°K, the rate of condensation of water vapor was 270 mg/h and, therefore, a heat-pipe design should provide for structural features to avoid diffusion of vapor through the gas. One answer is to use a high thermal conductivity wall material.

The work of Hessel [25, 28] is interesting: he used a laser to investigate the concentration field in a high-temperature GCHP (Na, Li, Ar). It was shown that there is a more pronounced separation of the components with a noncondensible gas present than without a gas.

The most promising heat pipes for transfer of large heat fluxes are those where the heat-transfer agent moves along arteries with low hydraulic resistance. An attempt to use arterial wicks in GCHP's has shown that the gas bubbles forming prevent the motion of liquid and decrease the transmitted thermal power. At present, ways are being sought for removing gas bubbles from heat-pipe arteries [29-31].

The appearance of bubbles is associated with the problem of gas solubility in the heat-transfer agent. When there is a drop in temperature the gas dissolved in the liquid is released and bubbles show up in the arteries and do not pass through the capillary structure in the vapor space, because of the surface tension at the phase interface. The time for the bubbles to disappear following their appearance is determined by the rate of solution of the gas in the liquid and the initial bubble radius. Thus, a reduction in the artery radius increases the reliability of its operation. In addition, the time for a bubble to be removed is proportional to the diameter $1/\alpha^*D$, where α^* is the Ostwald parameter, the ratio of the concentrations of dissolved substance in the liquid and gas phases.

The bubble elimination time can be as long as several days, and it is therefore difficult to use arterial heat pipes under dynamic operating conditions. The authors of [30] have suggested using cooling of the heat-transfer agent in the artery to remove bubbles. A reliable method of avoiding trapping of gas bubbles during servicing of an artery was proposed in a paper by Marcus [29]. The noncondensible gas is allowed to escape through a large number of apertures of capillary size in the artery wall, which is a thin foil. Because the menisci in this system flow together on the two sides of the thin foil, the gas bubbles will discharge easily from the artery. Experiments have shown that this is a highly efficient method of removing bubbles. Porous plates of metallized felt or cermet are recommended for the arteries of GCHP's; when mounted in the vapor channel and firmly compressed onto a screw thread on the evaporator and condenser, they are in no danger of being plugged up by bubbles of the noncondensible gas [32].

It was noted in [29, 33] that under certain conditions in GCHP's pressure fluctuations arise due to perturbations caused by convective dispersion of the gas front. This has been confirmed experimentally.

2. It was shown in [29, 4] that the release of gas in GCHP's leads to an undesirable increase in temperature. This matter has been discussed in [13], which presented results of an experimental investigation of the effect of different amounts of noncondensible gas on heat transfer in the condenser. It was shown that a very large amount of gas has a significant influence on the $T = f(Q)$ characteristic at low condenser temperature T_c.

Anderson and Petrick [34, 35] presented results of definitive tests on heat pipes. The main thrust of the work was an investigation of the liberation of gas (hydrogen) during interaction of the heat-transfer agent with the pipe wall and the wick. It was shown that methanol heat pipes made of stainless steel were promising for thermal control.

3. There are many different modifications of GCHP's (Fig. 3) designed for high thermal control levels. These fall into two groups. The first group concerns passive control heat pipes and the second, active control (with negative feedback).

As a rule, passive control GCHP's have a supplementary reservoir filled with noncondensible gas or a gas dissolved in a liquid.

Heat pipes can be classified as follows in terms of type, purpose, and role of the reservoir in the thermal control:

a) cold reservoir, with walls covered by a porous structure to reduce condensate;

b) hot reservoir (at constant temperature) with a wick on the walls;

c) a reservoir without porous structure (with a short fat connecting tube to ensure fast diffusion);

d) without a porous structure (with a long narrow connecting tube to achieve slow diffusion of vapor through the layer of noncondensible gas);

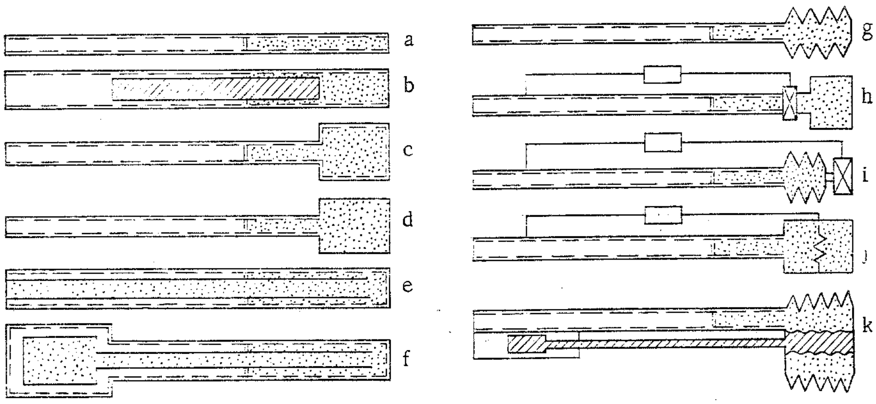

Fig. 3. GCHP structures: a) traditional GCHP without a gas reservoir [6]; b) a GCHP with a displaced volume [37, 38]; c) with a gaseous wick reservoir [7]; d) with a wick-free gas reservoir; e) with a displaced gas volume, playing the role of gas reservoir [7]; f) GCHP (same as e) with an increased gas reservoir volume; g) with a self-adjusting gas reservoir volume in the form of a bellows [39]; h) GCHP for active control with negative feedback determined by the amount of noncondensible gas [17]; i) active control with negative feedback controlled by the noncondensible gas pressure [17]; j) active control with temperature negative feedback [39]; k) using a hydraulic system [42].

e) a sequence of heat pipes with gaseous reservoirs;

f) a reservoir filled with liquid which has high solubility for the noncondensible gas (Ostwald number >10); reservoirs filled with liquid may be several times smaller than gas reservoirs [36].

Negative feedback in active control heat pipes can be accomplished by a mechanical action (expansion or contraction of the reservoir in proportion to change of evaporator temperature) or by electrical heating of the reservoir. Since the technique for investigating different GCHP structures is general, we shall give only a short description of the basic constructions.

The traditional construction (Fig. 3a) contains a noncondensible gas for thermal control, as proposed in [6], and has a low value of the temperature sensitivity factor. It is therefore not widely used. To increase σ^* one must decrease F_v/V_g [4]. As a constructive proposal to decrease the cross section, Shlitt [37, 38] proposed introducing a displaced volume (Fig. 3b) in the vapor channel.

An analogous effect can be achieved by using a supplementary gaseous reservoir. Such a construction was suggested by Bienert in [7] (Fig. 3c), and the gaseous reservoir may have a wick on the inside wall, or this may be absent (Fig. 3d). Bienert also proposed to combine structural designs to reduce F_v (Fig. 3b) and to increase V_g (Fig. 3c) in a single GCHP (Fig. 3e). A defect of this construction is the limited volume of the internal reservoir, and therefore a modification of the previous variant has been proposed (Fig. 3f). In the HP construction examined earlier the gas was enclosed in a constant-volume reservoir. The control capability can be enhanced by allowing the volume to be self-adjusting, i.e., to use an elastic bellows in practice. This type of construction was proposed by Wyatt in [39] (Fig. 3g). A similar effect can be attained by using a spirally wound pipe instead of the bellows, as proposed in [40]. Previously, we considered passive control heat pipes, whose main advantages are simplicity and reliability. In some cases one needs a higher degree of thermal control than is possible with a passive control system. Then controlled heat pipes with inverse feedback are used, i.e., active controlled heat pipes. The active control system consists of two parts: the control element, a passive control heat pipe, and the automatic system control element. The first uses of negative feedback for active control were in [16] (Fig. 3h). The active control is accomplished by means of a regulator located between the heat pipes and the gas reservoirs which regulates the amount of gas, depending on changes in temperature and pressure. The construction (Fig. 3i) is a variant of the active control heat

TABLE 1. Characteristics of Gas-Controlled Heat Pipes

Geometrical parameters $L_s + L_{a.z} + L_c$ $d_{hp} \times d_v$	Filter and wall material	Heat-transfer agent and noncondensible gas	$\alpha°$	$T \pm \Delta T$, °K	Q, W	V_g, cm^2	Literature source
46+25+43 cm 0,95×0,5 cm	Stainless steel —	C_2H_5OH Ar	0		0—30	66	[7]
8+23+15 cm 1,85×0,3 cm	Stainless steel Stainless steel × 8 mesh	H_2O, Ar	0	340∓10	30,0—320,0	0	[21]
10+20+20 cm 1,25×0,7 cm	Monel metal Monel metal 2 layers × 8 mesh	H_2O, 53% Ne 47% N_2	0	293∓1	55	0	[8]
60+42+63 cm 1,25×0,7 cm	Stainless steel mesh arteries, channels	NH_3, C_2H_5OH, Ar, He	0	310	100	56	[29]
1,4×1,2 cm	Stainless steel—mesh	CH_4—N_2	0	315	24	—	[48]
—	—	Li, K, Na, Rb, Cs, Ar	90	700—1500	—	—	[17, 18]
60,0 0,9×0,8	Stainless steel	Na, Ar	90	700—900	—	0	[51]
50 0,9×0,8	Tantalum	Pb, Ar	0	1500—1900	2300	0	[16]
38,0 3,0×2,7	Channels	K, Ar	0	1100	1000	50	[37, 38]
90 1,0—1,6	Mesh	Na, H_2	0	1000—1200	600	0	[4]
100,0	Steel—arteries	NH_3, He	0	300—310	30	—	[46]

pipe [16]. Control is achieved by varying the bellows volume for a constant amount of gas, and active control of the thermal resistance of the pipe is achieved by varying the gas temperature, using the system shown in Fig. 3j [39].

An active control system, used as a control element of a hydraulic system, is shown in Fig. 3k. Many papers have studied this system, because of the high level of temperature control and the absence of electronic elements [41-43].

4. The main interest in experimental studies of GCHP's is an investigation of their thermal control characteristics. Table 1 gives the main parameters of experimental GCHP's. It can be seen that a large portion of the work deals with investigation of high-temperature GCHP's using liquid metals [16, 18, 19, 44, 45], and also medium-temperature heat pipes (250-400°K), applied to thermal control of spacecraft [22, 23, 24, 46-50]. The GCHP's have the following features.

1. Presence of Interfaces. The first paper applying a gas-controlled heat pipe used it to determine the saturation pressure of liquid metals at high temperature [17]. The problem of measuring the pressure in liquid metals is complex, since it is difficult to use manometers under these conditions, and the direct methods used are complicated and have a large measurement error. For this purpose the authors used the test metal as the working liquid in a heat pipe. Because of the fact that $P_v - P_g$ and that there was practically no mass flux through the interface, it was possible to measure $P_v = f(T_v)$ using an ordinary manometer at room temperature (Fig. 4a).

In [18] the presence of a vapor—gas interface is used to determine the surface-tension coefficient. A schematic diagram of the equipment is shown in Fig. 4b. The presence of noncondensible gas makes it possible to insert a moving probe into the gas-controlled heat pipe and to measure the pressure difference in the probe and in the heat pipe. Using this method the authors measured the surface tension of the metal Li, Na, K, Rb, Cs over the temperature range 300-1600°K.

2. Movement of the Interface. Figure 4c shows a diagram of a heating device located in a spacecraft, where the latter rotates relative to the heat source (the sun, the earth,

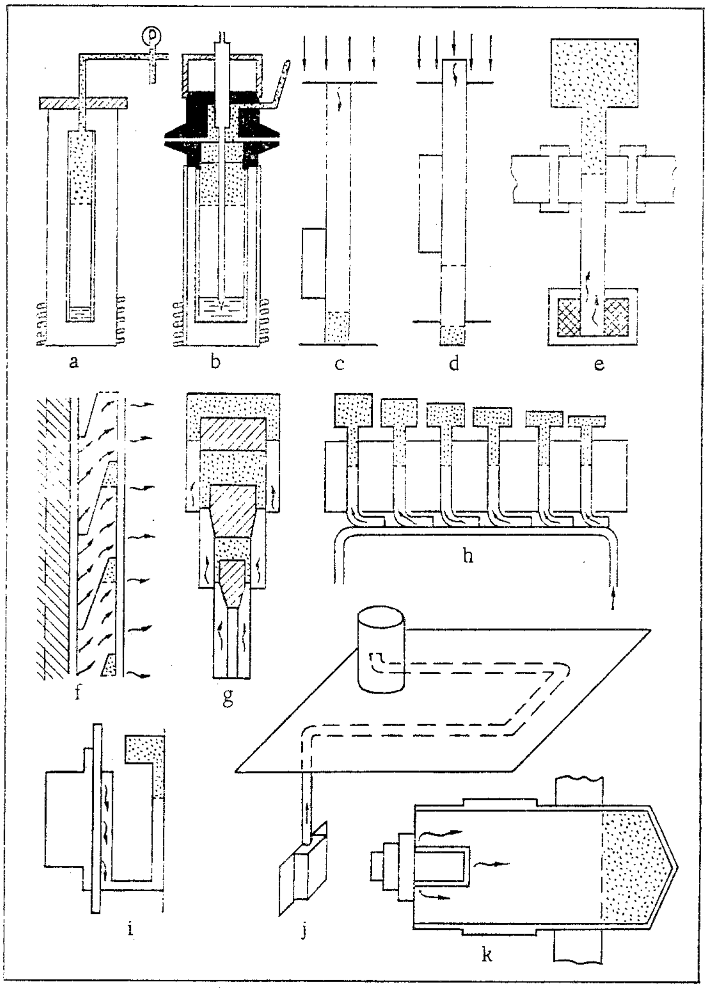

Fig. 4. Uses of GCHP's: a, b) devices for measuring P = f(T) on the saturation line [17], and for measuring σ [18]; c, d) heaters for electronic instruments [6]; e) cooler for a radioisotope generator [53]; f) thermal control device for a fuel element [51]; g) radiator for a thermionic converter [37]; h) radiator-receiver [54]; i, j, k) cooler for the orbiting astronomical observatory (OAO) system [46], for the lunar surface magnetometer [48], and for a high-voltage instrument [55].

etc.). The system is a controlled heat pipe to which a thermostat element is attached. The tube itself terminates at both ends in radiators. During the spacecraft flight alternatively one radiator and then the other is illuminated. The presence of a certain amount of gas in the tube inhibits heat transfer from the instrument to the radiator located in the shadow and also inhibits transfer of heat from the illuminated radiator to the shaded one. When the spacecraft rotates the opposite radiator toward the sun, the picture is repeated, but in terms of the second radiator. The system (Fig. 4d) allows not only heating of an instrument, but also rejection of undesired heat into space. This is possible because part of the tube is located outside the shaded radiator. Under normal operation one of the radiators is illuminated, and the vapor–gas interface is located between the instruments and the shaded radiator. While a radiator is receiving undesired excess energy (e.g., a solar flare) or while the instrument is giving out excess energy, the vapor–gas interface moves beyond the shaded radiator, and thus thermal energy can be rejected. A similar effect occurs during launch of heat pipes in which the heat-transfer agent is initially in the solid state or has a high viscosity [20, 26, 52]. The displacement of the vapor–gas interface due to melting at launch of a gas-controlled heat pipe is probably the only method of solving this problem.

3. Thermal Control. This parameter is the basic one, and the region of application is very wide [20, 37, 44, 46, 48, 51-56, 65-79]. The traditional fields of application of GCHP's are nuclear energy, space technology, and the radio and electronics industries. Controlled heat pipes are used to control thermal conditions for the cathodes of thermo-emission radioisotope generators [53] (Fig. 4e). As the fuel drops and the excess energy decreases, the vapor-gas interface moves toward the edge of the thermo-emission converter cathode, reducing the excess heat rejection section. Therefore, this type of heat pipe can be considered as a unique safeguard against external thermal energy or overheating, which may occur if a thermo-emission converter goes unserviceable or if the external circuit is disconnected. An interesting case is the use of a bank of coaxial GCHP's for thermal control of fuel elements [51]. A schematic of this device is shown in Fig. 4f.

Lee and Werner [44] investigated the possibility of thermal control of an irradiated capsule using a gas-controlled heat pipe. The thermal control efficiency is higher, the higher the heat flux density. It was therefore proposed [37] to use a series heat-pipe construction as a radiator for the thermo-emission converter, the first pipe having low heat flux density in the evaporator and high heat flux in the condenser. The irradiation scheme for the three pipes is shown in Fig. 4g. Gas-controlled heat pipes have found wide applications in space technology, because of their thermal control properties. They are used in radiator-coolers, to cool equipment and instruments, and also to equalize temperatures of structural elements in spacecraft. In particular, GCHP's are used in NASA spacecraft systems: in the orbiting astronomical observatory (OAO), in the Skylab, in the Shuttle, etc. Figure 4h shows the radiator schematic [54]. The radiator construction is such that different GCHP's have different reservoir volumes, i.e., the larger gas reservoir volume is located closer to the heat-transfer agent outlet. The authors of [46] proposed the use of GCHP's for thermal control of the OAO. A scheme for thermal control of a typical spacecraft instrument is shown in Fig. 4i. Marcus [48] proposed to use a thermal control effect for the lunar surface magnetometer (LSM). The magnetometer system is shown in Fig. 4j. A passive gas-controlled heat pipe with a cold wick-type gas reservoir was used.

The use of GCHP's for thermal control of electronic equipment has been described in [55]. One of the main defects of heat pipes for use in high-voltage electronic instruments and transformers is the low dielectric strength. This stems from the low dielectric parameters of the vapor at low pressure, e.g., when starting up or under conditions of low heat flux.

This problem is easily solved by means of a gas-controlled heat pipe in which the noncondensible gas pressure can increase the electrical strength by several orders of magnitude. A variant of this use of the properties of GCHP's was proposed in [20, 56], with reference to cooling of a traveling-wave tube. A schematic of this tube is shown in Fig. 4k.

II. Heat Pipes with Controlled Vapor Flux

The thermal resistance of heat pipes can be varied by controlling the vapor flux by means of a throttle or a valve located in the vapor channel [57]. The valve is mechanically connected to a sealed reservoir filled with liquid, which expands or contracts under the action of the heat flux from the heat source. The variation in the pressure drop in the heat pipe causes a variation in the evaporator temperature [58]. A thermal sensitive reservoir, filled with liquid, can be located inside the heat pipe evaporator or can be attached to it at one side. A second method of mechanical control of vapor flux is used to dry the porous wick in the heat-pipe evaporator [59]. When the vapor channel is covered there is a sharp increase in the vapor pressure in the evaporator and liquid is discharged from the porous wick of the evaporator into the condenser. When the heat load decreases the valve opens and equalizes the pressure along the vapor channel. The liquid passes to the evaporator along the porous wick. It is appropriate to use a heat pipe with a controlled vapor flux when the temperature difference between the evaporator and the condenser is small (in this case GCHP's have low efficiency).

Concurrent GCHP's with active negative feedback (electrical heating of the gas reservoir) are heat pipes with controlled vapor flux using a fan or in a throttle. A defect of heat pipes with vapor flux control is that the condenser temperature must always be higher than the freezing temperature of the liquid, as otherwise the liquid cannot return to the evaporator. The maximum temperature difference between the evaporator and the condenser in this kind of heat pipe is determined by the capillary potential of the wick. If the pressure drop between the evaporator and the condenser is greater than the capillary head, the vapor passes through the porous structure and the tube becomes uncontrolled. Therefore, for tubes with vapor flux control it is expedient to use a liquid with a low partial vapor pressure. On the other hand, to increase the capillary head of the wick it is desirable to use porous structures with small pore size (which has a negative effect on the output of the capillary pump). Therefore, it is especially effective in these cases to use a finely porous structure with arteries, and the absence of noncondensible gas allows a wide class of applications. The use of finely porous wicks (cermets, felt, or fine mesh) limits the heat-pipe length. It is desirable to make them short so that they can be located between the two ordinary pipes for control of the thermal system as a whole.

Marcus and Eninger [59] described the construction of a water heat pipe made of Monel metal with vapor flux control. The heat-sensitive liquid chosen was methanol (14.9 cm^3 volume). The outer diameter of the pipe was 2.54 cm, and the length was 61.9 cm. The pipe consisted of two identical sections, separated by a valve of Monel metal. A porous wick (four layers), including a single artery, was made of 118 mesh grid. The liquid could only pass through the body of the valve along the artery. The valve was fully closed at a temperature $T = 23.9°C$. During transmission of power up to 100 W the heat pipe controls the thermal resistance, maintaining the evaporator temperature in the limits 19.5–21.7°C. An increase in power above 100 W causes complete opening of the valve and the tube then behaves normally.

Heat pipes using a mechanism for discharging liquid from the evaporator filter to the condenser have fully as much promise for application in thermal control systems as heat pipes with vapor flux control. It is also attractive to use them jointly with two ordinary heat pipes, located on both sides of the controlled pipe.

Heat pipes with liquid expulsion have lower thermal resistance than pipes with controlled vapor flux. The main limitation on the transmitted power of such pipes is the transmission capability of the capillary pump. In these the thermal resistance control is achieved not by ensuring a difference in the saturation temperatures in the evaporator and condenser regions, but by a variation in the heat-transfer process in the evaporator filter. When the valve closes owing to an increase in pressure in the evaporator, the liquid is expelled into the condenser and the heat transfer sharply deteriorates. When the heat flux is reduced the valve opens, equalizing the pressure, and the liquid again reaches the evaporator.

Such heat pipes use structures with high permeability and large pores, a system of channels or arteries, and the liquid must have a high partial pressure (e.g., ammonia).

Marcus and Eninger [59] described a system of three heat pipes of length 155 cm. The middle heat pipe of diameter 2.54 cm and length 31 cm has a thermal resistance control system. To its ends are attached ordinary heat pipes of external diameter 1.27 cm. The length of the inlet pipe is 36 cm and that of the exit pipe is 88 cm. The working liquid is ammonia. The volume of the heat-sensitive liquid (perfluoropentane) is 2.3 cm^3. The inlet and controlled pipes are made of stainless steel, and the exit pipe is made of aluminum. The heat-sensitive volume is built into the aluminum body of the heater of the inlet heat pipe. Part of the external surface of the inlet and exit pipes, located within the controlled pipe, has a screw thread. The heat pipes are joined together by a porous artery. The thermal conductivity of the tube system with the valve fully open is 12.6 W/°C, and with the valve fully closed it is less than 0.018 W/°C. For comparison, the thermal conductivity of a heat pipe with vapor flux control of the first pipe is 15 W/°C with the valve open and 0.36 W/°C with it closed.

III. Heat Pipes with Liquid Flux Control

Control of the thermal resistance of a heat pipe by changing the liquid content in the evaporator can be accomplished in several ways. The first way is to remove the liquid from the evaporator (drying the wick) into a supplementary reservoir located behind the condenser [60]. A second method is to introduce excess of liquid into the evaporator and to block it off (a heat pipe operating as a diode) [61].

By partially evacuating the evaporator or partially blocking off the liquid one can vary the thermal resistance of the heat pipe in proportion to a change in heat flux or temperature.

The original method of control of thermal resistance is the method of a measured supply of liquid to the evaporator using an "airlift." The essence of this effect is that in boiling of a liquid in a tube or channel of small diameter the vapor bubbles formed force the liquid out of the channel, like a piston.

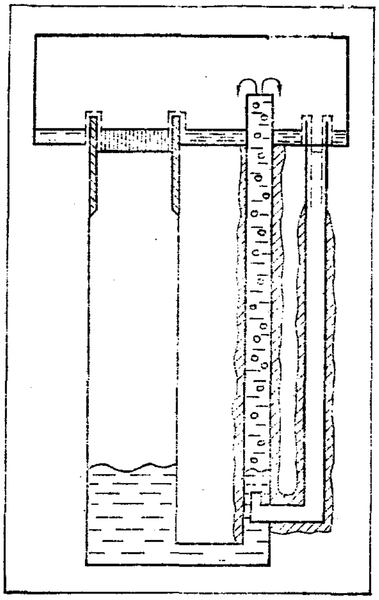

Fig. 5. Construction of a controlled heat pipe using the airlift principle.

This expulsion force arises from the difference in the liquid and vapor density. The phenomenon can be used to raise the liquid by a certain height against gravity, if the lower part of the tube is heated. The pipe walls must be thermally insulated. The number of bubbles arising during liquid boiling in a narrow channel or pipe is proportional to the heat flux density at the wall. By increasing the heat flux we obtain a large yield of the pump when pumping the liquid, and the pump output decreases when the heat flux decreases. The drops of liquid, passing from the top end of this supply tube, reach the porous structure of the heat-pipe evaporator and are smeared over it because of capillary forces, thus cooling the evaporator surface. The vapor formed in the evaporator moves toward the cold part of the heat pipe, where it condenses, releasing heat. The liquid from the heat-pipe condenser reaches the supply pipe, where there is boiling of the liquid afresh, and a rise in the two-phase liquid—vapor mixture against the gravity force.

The pump here has the following requirements. The expenditure pumping a liquid should be small compared with the thermal power transmitted by the heat pipe; the thermal capacity of the pump should be a minimum, and the necessary amount of liquid must be constantly supplied to the evaporator.

The lower end of the supply pipe can be either closed or open. With a closed end the liquid is supplied upward, and the heating cycle is pulse-type. When the heating of the lower part of the pipe is stopped, the liquid is stored. When the heater comes on the liquid boils and is ejected into the evaporator. If the lower end of the pipe is open and joined to the heat-pipe condenser, the tube heating process may be continuous, with the liquid arriving in the evaporator steadily.

Robert [62] has described the parameters of a heat pipe with an airlift pump. The pipe is made of copper, and the heat-transfer agent is water. The condenser is cooled by forced-air convection. An infrared method is used for monitoring the operation and for controlling the thermal resistance of the pipe. The airlift pump power is 2 W for a transmitted thermal power of 70 W along the pipe. The diameter of the supply pipe plays an important part in the thermal control. Pipes of small diameter allow fine control of the thermal resistance, but the pump output is then small. Pipes of large diameter allow large pump output, but the control of thermal resistance is coarser. Therefore, it is desirable to use several supply pipes of different diameters.

At the A. V. Lykov Institute of Heat and Mass Transfer of the Academy of Sciences of the Belorussian SSR several CHP's based on the airlift principle [66] have been developed. Figure 5 shows a schematic of one heat pipe. The liquid from the heat-pipe condenser is supplied by an airlift pump to an auxiliary condenser, located above the heat-pipe evaporator, and separated from it by a porous hydraulic valve. Under the action of capillary forces and the gravity field the liquid reaches the evaporator via the hydraulic valve. By varying the rate of heating of the lower part of the airlift pump supply tube, one can control supply of liquid to the heat-pipe evaporator. The hydraulic valve between the evaporator and the upper end of the airlift pump pipe allows the liquid to be raised against gravity to a large height, compared with the heat-pipe constructions described above.

IV. Electrohydrodynamic and Electro-Osmotic Controlled Heat Pipes

In electrohydrodynamic heat pipes the control of the flow of liquid from the condenser to the evaporator is accomplished by means of an electric field. If the heat pipe body is made of metal (e.g., aluminum), the ends are made of a dielectric (Plexiglas), and electrodes are drawn through the ends (wires, strips, etc.) parallel to the pipe walls, then an electric potential applied between the electrode and the body can transport a dielectric liquid (Freon, diatherm, etc.) [63].

The space between the electrode and the pipe body plays the role of an artery. Jones [63], the author of this type of construction, has calculated that in an electrohydrodynamic pipe one can transport liquid (e.g., diatherm) against the force of gravity to a height of 50 cm and more. Electrohydrodynamic heat pipes have very fast response. By varying the electric potential according to some specific pattern, one can provide for a pulsed or steady supply of liquid to the evaporator, and can quickly suppress it at a specified time. Electrohydrodynamic heat pipes can be used for partial evacuation of the evaporator, and the thermal resistance can be varied smoothly over a wide range.

In electro-osmotically controlled heat pipes, in contrast with electrohydrodynamic types, a low-voltage electric field is used to transport the liquid. If electrodes are located between the condenser and the evaporator in the porous wick of a heat pipe, made of a porous dielectric, and an electric potential (20–50 W) is applied to these, then a dielectric or weakly polar liquid can be moved along the porous wick. The liquid head in a particular case can be substantially greater than the capillary head. For example, the authors of [64] were able to transmit 2.4 kW of energy with an electric potential of 20 V in a pipe of length 41 cm and diameter 5 cm, with a wick made of glass spheres, saturated with water. Further developments of forms of construction based on this principle can probably improve the parameters of electro-osmotic controlled heat pipes substantially.

NOTATION

T, temperature; V, volume; P, pressure; L, length; F, area; Q, heat flux; q, heat flux density; m, mass; M, molecular weight; k, heat-transfer coefficient; ρ, density; σ, surface tension; n, molar volume; r^*, latent heat of evaporation; h, height; σ^*, temperature sensitivity coefficient; D, diffusion coefficient; R_0, gas constant; g, acceleration of gravity; k', elasticity constant; λ, thermal conductivity; τ, time; α^*, Ostwald coefficient; R, radius. Indices: v, vapor; g, gas; r, reservoir; sk, heat sink; s, heat source; c, condenser; a, active zone of condenser; d.v., displaced volume; int, interior volume; tr, heat flux transformer; st, straight flux; d, thermal diode; con, control heat flux; t, thermal diffusion; tot, total pressure; 0, initial value; a.z., adiabatic zone.

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