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HEAT PIPES WITH HEADER AND ARTERY SYSTEMS

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Abstract—A novel form of artery and header system for use in heat pipes is described, then thermal/hydraulic design procedures developed. The artery system is simple and robust, and self-priming without the use of capillary arteries. Operation with single-phase fluids in the artery, when acting against gravity, is discussed. When in this orientation the axial load can exceed that in the horizontal plane, and be more than ten-fold the unit without an artery.

NOMENCLATURE

A, wick cross-sectional area;
E, coefficient defined by equation (46);
F, friction pressure ratio, equation (17);
F_f, modified friction pressure ratio defined by equation (44);
F^*, friction pressure ratio between zero flow points defined by equation (29);
$\bar{F}$, modified friction pressure ratio defined by equation (23);
g, gravitational acceleration;
K, permeability;
L, specific latent heat;
$\dot{m}$, mass flowrate;
$\dot{m}_H$, mass flowrate in horizontal pipe with no artery and vapour shear;
$\dot{m}_0$, mass flowrate in horizontal pipe of length equal to capillary rise with no artery and vapour shear;
$\dot{m}_V$, mass flowrate in vertical pipe with no artery and vapour shear;
$\dot{m}_1$, mass flowrate downward in wick;
$\dot{m}_2$, mass flowrate upward in wick;
n, ratio of total heat-transfer length to evaporator length;
p_G, vapour pressure;
p_L, liquid pressure;
Δp_c, maximum capillary pressure difference;
$\Delta p_{c,2}$, capillary pressure difference between zero flow points;
Δp_G, overall pressure drop in vapour;
Δp_L, pressure drop in wick;
$\frac{dp_f}{dz}$, friction pressure gradient in wick;
$\frac{dp_G}{dz}$, friction pressure gradient in vapour;
Q, axial heat load;
Q_{cf}, axial heat load for pipe of length cf;
Q_H, axial heat load for horizontal pipe with no artery;
Q_0, axial heat load corresponding to $\dot{m}_0$;
Q_V, axial heat load for vertical pipe;
q, length ratio $\frac{z_c}{z}$;
r, length ratio $\frac{z_2}{z_1}$;
r_3, length ratio $\frac{z_3}{z}$;
v_G, specific volume of vapour;
v_L, specific volume of liquid;
w, length ratio $\frac{z_4}{z}$;
z, heat pipe length;
h_c, capillary rise, equation (4);
$z_{1,2,3,4}$, lengths as shown in Fig. 2.

Greek symbols

β, pressure ratio $\frac{(p_G - p_L)}{\Delta p_c}$;
β_1, pressure ratio β at point c;
μ_G, absolute viscosity of vapour;
μ_L, absolute viscosity of liquid;
ρ_G, density of vapour;
ρ_L, density of liquid.

1. INTRODUCTION

THE PERFORMANCE of heat pipes [1, 2] are greatly improved by the provision of arteries [2] which provide paths of low flow resistance to allow the condensate to return from condenser to evaporator.

Major problems in the design of heat pipes [3] are the requirement of ensuring successful priming of the artery and of preventing vapour or gas blockage of the artery. In conventional artery systems the artery is primed by the action of capillary forces, which puts an upper limit on the possible bore size of the artery. Vapour blockage is prevented by placing the artery remote from the heated wall, and gas blockage by ensuring thorough degassing of the device.

This paper discusses an artery system which does not rely on capillary forces for the priming of the artery, and avoids vapour and gas blockage by a header system in which vapour and gas may collect without reduction in heat-pipe performance. Patent protection has been obtained for the arrangement.

2. THE ARTERY/HEADER SYSTEM

Figure 1 shows the system. Along the wall of the heat pipe (1) there is a wick (2) to which are sealed at both ends header plates (3, 4). There is no access between the headers (5, 6) and the vapour space (7) except through the wick. Connecting the header plates (3, 4) and sealed to them is the artery (8).

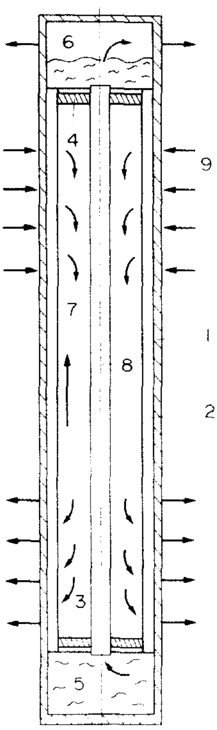

FIG. 1. Heat pipe with header and artery system.

The wick (2), the header plates (3, 4), and the artery (8) form a sealed unit. Provided good contact can be achieved between the wick and wall in the evaporator (9) the unit can be assembled prior to insertion within the heat pipe. When it is assembled there are two spaces connected only through the wick: the space (7) between the headers; and the header space (5, 6), the artery, and the space (if any) between the wick and wall.

Consider that sufficient liquid has been added to saturate the wick and the remainder of the available space except the space (7) between the headers. The wick length is less than the capillary height so the wick will become saturated in any orientation relative to gravity. With the heat pipe standing vertically, consider the situation where the upper header (6) contains vapour; the lower part of the vapour space (7) must in that case contain liquid. If the evaporator is now heated the pressure will rise in the vapour space (7) between the headers. This will force the liquid in the lower part of the vapour space through the wick into the header/artery system. Vapour will not pass into the upper header (6) owing to the action of the capillary forces in the saturated wick. Thus liquid is displaced into the header system; the unit can be designed so that either an upper header is provided larger than the volume of the compressed vapour in the upper header (this is the situation shown in Fig. 1), or the vapour can be condensed by heat transfer from the upper header. The latter is a more convenient method if the device operates at temperatures above ambient.

By this method the artery is filled with liquid against gravity without the use of capillary forces within the artery. The artery can be then made sufficiently large that flow losses in it can be neglected.

The performance of such header/artery systems are analysed in the remainder of this paper.

3. PERFORMANCE WITH NO ARTERY

The axial heat load is related to the circulation rate and latent heat in the expression

$$Q = \dot{m}L. \tag{1}$$

It is convenient to take, as a reference flowrate, $\dot{m}_0$ that of a horizontal heat pipe of length equal to the capillary height (z_c), with no isothermal length and with uniform heat flux in both evaporator and condenser; the pressure drop in the vapour phase is assumed to be negligible. In a porous wick with laminar flow

$$-\frac{dp_f}{dz} = \frac{\mu_L \dot{m}}{\rho_L K A} \tag{2}$$

hence

$$\Delta p_c = \frac{\mu_L \dot{m}_0 z_c}{2\rho_L K A} \tag{3}$$

where Δp_c is the maximum capillary pressure difference. From the definition of the capillary height

$$\Delta p_c = g\rho_L z_c. \tag{4}$$

Also define

$$q = \frac{z_c}{z}. \tag{5}$$

For a horizontal pipe of length z and isothermal length z_3

$$\Delta p_c = \frac{\mu_L \dot{m}_H (z - z_3)}{2\rho_L K A} + \frac{\mu_L \dot{m}_H z_3}{\rho_L K A}. \tag{6}$$

Using equations (3)–(5) and the length ratio definitions described and defined in the nomenclature and Fig. 2,

$$\frac{Q_H}{Q_0} = \frac{\dot{m}_H}{\dot{m}_0} = \frac{q}{1 + r_3}. \tag{7}$$

For vertical flow against gravity

$$\Delta p_c = g\rho_L z + \frac{\mu_L \dot{m}_V (z - z_3)}{2\rho_L K A} + \frac{\mu_L \dot{m}_V z_3}{\rho_L K A} \tag{8}$$

then using equations (3), (4) and (8)

$$\frac{Q_V}{Q_0} = \frac{\dot{m}_V}{\dot{m}_0} = \frac{q - 1}{1 + r_3}. \tag{9}$$

In the following sections the performance with the header/artery system is analysed.

4. CONDENSATE FLOW AGAINST GRAVITY

Figure 2 illustrates schematically the variation of capillary pressure along the wick, as a plot of ratio of the difference between the gas and liquid pressure ($p_G - p_L$) and the maximum capillary pressure difference Δp_c. Length ad represents the evaporator, de the isothermal length, and eh the condenser. Point a is immediately adjacent to the upper header plate and point h adjacent to the lower header plate.

Through the artery and headers the pressure gradient is assumed to be due to gravity only. The capillary pressure difference between points a and h is then attributable to gravity and the slope of line jk in Fig. 2 is, using equations (4) and (5),

$$\frac{g\rho_L z}{\Delta p_c} = \frac{z}{z_c} = \frac{1}{q}. \qquad (10)$$

This governs the relation between points j and k in Fig. 2. With no heat transfer the capillary pressure variation along the heat pipe is given by line jk; the system is in equilibrium with the headers and artery filled with liquid.

On heating the unit, the capillary pressure difference increases in the evaporator, and decreases in the condenser until the maximum capillary pressure difference is achieved at point b. The minimum pressure difference occurs at g in the condenser.

5. POINTS OF ZERO CONDENSATE FLOW

Consider now the location of the points of zero condensate flow (c and f). Condensate flows upward from point f towards point c, and also downward from point f, through the artery, to point c. As the gravitational force on the liquid in the artery balances that on the liquid in the wick, it follows that the friction pressure drop in each direction must be the same. This is regardless of any vapour shear.

This can be expressed, neglecting friction in the headers and artery, as

$$\frac{\mu_L \dot{m}_1 n z_1}{2\rho_L K A} = \frac{\mu_L \dot{m}_2 n z_2}{2\rho_L K A} + \frac{\mu_L \dot{m}_2 z_3}{\rho_L K A}. \qquad (11)$$

If uniform heat flux is assumed in both evaporator and condenser

$$\frac{\dot{m}_1}{z_1} = \frac{\dot{m}_2}{z_2} = \frac{\dot{m}}{z_1 + z_2}$$

or

$$\dot{m}_1 = \frac{\dot{m}_2}{r} = \frac{\dot{m}}{1+r}. \qquad (12)$$

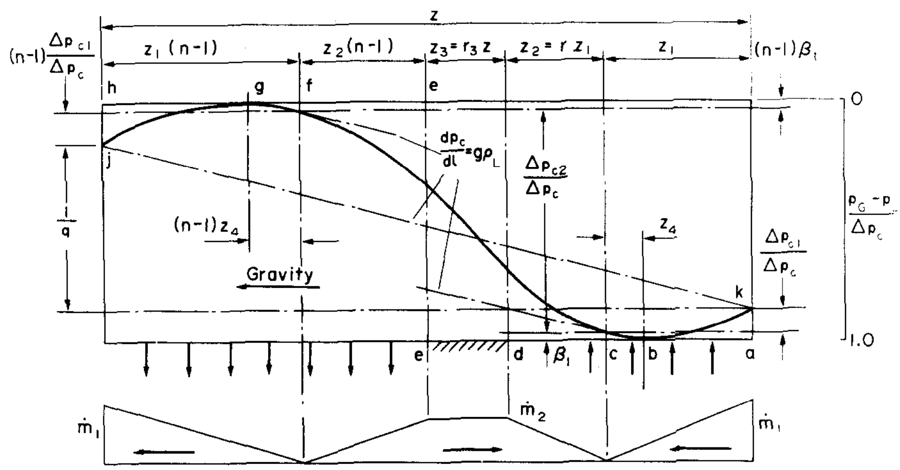

FIG. 2. Distribution of capillary pressure difference acting against gravity.

Also from the various length ratio definitions (see Fig. 2)

$$z = z_1(1+r)n + r_3 z$$

or

$$\frac{z_1}{z} = \frac{1-r_3}{(1+r)n}. \qquad (13)$$

From equations (11)–(13)

$$r = \frac{1-r_3}{1+r_3} \qquad (14)$$

or

$$1+r = \frac{2}{1+r_3}. \qquad (15)$$

6. CAPILLARY PRESSURE DIFFERENCE AT ZERO FLOW POINTS

The solution requires the evaluation of the distance between the points of zero capillary pressure gradient (points b and g) and the points of zero condensate flow (points c and g).

Define

$$w = \frac{z_4}{z_1} \qquad (16)$$

and

$$F = \frac{\Delta p_G}{\Delta p_c} \tag{17}$$

where Δp_G is the overall vapour pressure drop over the heat pipe. For the heat pipes without arteries, F has been assumed to be zero.

Assuming laminar flow of the vapour it can be shown [4], using the above definitions, that the vapour pressure gradient at b is

$$-\frac{dp_G}{dz} = Fg\rho_L q(1-w). \tag{18}$$

The condensate pressure gradient due to friction at b is

$$-\left(\frac{dp_f}{dz}\right)_b = \frac{\mu_L \dot{m}_1 w}{\rho_L KA}. \tag{19}$$

Combining equations (3), (4) and (19)

$$-\left(\frac{dp_f}{dz}\right)_b = 2\frac{\dot{m}_1}{\dot{m}_0} wg\rho_L. \tag{20}$$

At the point b using equations (18) and (20), as the pressure gradients in the vapour and liquid must be the same as there is zero capillary pressure gradient, and neglecting gravitational effects on the vapour,

$$g\rho_L[1+Fq(1-w)] = 2\frac{\dot{m}_1}{\dot{m}_0} wg\rho_L \tag{21}$$

from which

$$2\frac{\dot{m}_1}{\dot{m}_0} = \frac{\bar{F}}{w} - Fq \tag{22}$$

where

$$\bar{F} = 1+Fq. \tag{23}$$

The pressure drop in the vapour between points b and c is

$$\Delta p_{G,bc} = \tfrac{1}{2}Fg\rho_L q(2-w)z_4. \tag{24}$$

A force balance over length bc gives

$$z_4 g\rho_L[1+\tfrac{1}{2}Fq(2-w)] = \beta_1 \Delta p_c + \frac{\mu_L \dot{m}_1 z_4^2}{2\rho_L KAz_1} \tag{25}$$

where $\beta_1 \Delta p_c$ is the capillary pressure difference at point c.

Using equations (3), (4), (16), (22) and (25), then rearranging, gives

$$\beta_1 = \frac{z_1}{z_c}\frac{w}{2}\bar{F}. \tag{26}$$

A similar analysis in the condenser results in the conclusion that the capillary pressure difference at the zero flow point (point f) is $(n-1)\beta_1$. The difference in capillary pressure between the zero flow points is then

$$\Delta p_{c,2} = \Delta p_c - n\beta_1 \Delta p_c$$

$$= \Delta p_c\left(1 - \frac{nz_1}{z_c}\frac{w}{2}\bar{F}\right). \tag{27}$$

7. PRESSURE BALANCE BETWEEN ZERO FLOW POINTS

It can be shown [4] that the vapour pressure drop between the zero flow points is

$$\Delta p_{G,2} = F^*\Delta p_c \tag{28}$$

where

$$F^* = F\left[1 - \frac{1-r_3}{(1+r_3)(1+r)^2}\right]. \tag{29}$$

Consideration of the pressure change between the zero flow points through the artery, assuming negligible friction in the artery, gives

$$\Delta p_{c,2} = \frac{\mu_L \dot{m}_1 n z_1}{2\rho_L KA} + (z-nz_1)g\rho_L + F^*\Delta p_c$$

$$= \Delta p_c \frac{\dot{m}_1}{\dot{m}_0}\frac{nz_1}{z_c} + (z-nz_1)g\rho_L + F^*\Delta p_c. \tag{30}$$

Combining equations (27) and (30) gives

$$w^2 - 2w\left\{\frac{2[q(1-F)-1]}{(1+Fq)(1-r_3^2)} + 1\right\} + 1 = 0. \tag{31}$$

This can be solved for w. Then from equation (22)

$$\frac{\dot{m}_1}{\dot{m}_0} = \frac{1}{2}\left(\frac{\bar{F}}{w} - Fq\right). \tag{32}$$

Combining equations (12), (15) and (32)

$$\frac{Q}{Q_0} = \frac{\dot{m}}{\dot{m}_0} = \left(\frac{\bar{F}}{w} - Fq\right)\frac{1}{1+r_3}. \tag{33}$$

Combining equations (7) and (33)

$$\frac{Q}{Q_H} = \frac{\dot{m}}{\dot{m}_H} = \left(\frac{\bar{F}}{w} - Fq\right)\frac{1}{q}. \tag{34}$$

For the case of a horizontal pipe ($q = \infty$) equation (31) reduces to

$$w^2 - 2w\left[\frac{2(1-F)}{F(1-r_3^2)} + 1\right] + 1 = 0 \tag{35}$$

and using equation (21) with no gravitational term

$$\frac{\dot{m}_1}{\dot{m}_0} = \frac{Fq}{2}\left(\frac{1}{w} - 1\right). \tag{36}$$

Combining equations (7), (12) and (36)

$$\frac{Q}{Q_H} = \frac{\dot{m}}{\dot{m}_H} = F\left(\frac{1}{w} - 1\right). \tag{37}$$

This gives the ratio of circulation rates with and without an artery in the horizontal plane. When F is zero

$$\frac{Q}{Q_H} = \frac{\dot{m}}{\dot{m}_H} = \frac{4}{1-r_3^2}. \tag{38}$$

8. THE VAPOUR FRICTION RATIO

Assume that the liquid and vapour pressure drops are identical in magnitude, and that these and the gravitational component together equal the maximum capillary pressure difference. Then

$$2\Delta p_G + g\rho_L z = \Delta p_c. \tag{39}$$

Using equations (4), (5) and (39)

$$\Delta p_G = \frac{\Delta p_c}{2}\frac{q-1}{q}. \quad (40)$$

From the definition of F (equation 17) then

$$F = \frac{q-1}{2q}. \quad (41)$$

9. PREDICTED PERFORMANCE RATIOS

Figure 3 shows the performance ratios Q/Q_H to a base of the isothermal length ratio $r_3(z_3/z)$ for the device operating against gravity. The ratio of the circulation rates is, of course, equivalent to the ratio of the axial heat-transfer rates. Performance throughout is better than the horizontal pipe without the artery, particularly with larger isothermal lengths.

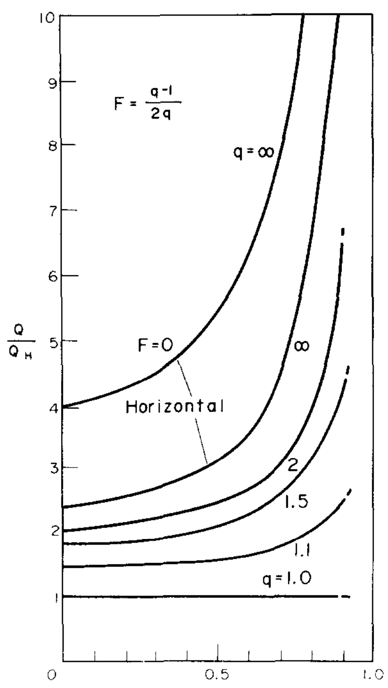

FIG. 3. Performance ratio Q/Q_H to a base of isothermal length ratio.

In Fig. 3 gas friction has been introduced on the basis discussed in the previous section, equation (41) being used for F. In many cases the unit can be designed to give less gas friction in which case the performance ratios will be increased above those in Fig. 3.

By assuming $F = 0$ for the reference device with no artery (Section 3) the performance ratios are underestimated.

10. PERFORMANCE WITH CONDENSER FLOODING

When acting against gravity, if the axial load is increased beyond the conditions corresponding to Fig. 2, the condenser will flood or overflow over the length gh.

The condensate pressure drop in that case over gh is

$$\Delta p_{L,gh} = g\rho_L(n-1)z_1(1-w)+\Delta p_{G,gh} \quad (42)$$

and it can be shown (4) that in this case equation (31) becomes

$$w^2+2w\left\{\frac{[q(1-F)-1](1+r)n}{F_f L_R(1-r_3)}+1\right\}+1 = 0 \quad (43)$$

where

$$F_f = 1+\frac{2Fq}{(1+r)(1+r_3)}. \quad (44)$$

The length ratio, r, is required to solve equation (43); with condenser flooding equation (15) is not applicable. This is obtained taking the force balance round the liquid path. The equations obtained are

$$r = \frac{1-r_3-E}{1+r_3+E} \quad (45)$$

where

$$E = \frac{\dot{m}_0}{\dot{m}_1}\frac{1-r_3}{(1+r)^2}\frac{n-1}{n}(1-w) \times\left[\frac{\dot{m}_1}{\dot{m}_0}(1+w)-1-\frac{(1-w)Fq}{(1+r)(1+r_3)}\right]. \quad (46)$$

An iterative process is necessary to solve these equations; however, convergence is rapid if E is assumed to be zero for the first iteration.

Having calculated w the performance ratio is obtained (4) as

$$\frac{Q}{Q_H} = \frac{\dot{m}}{\dot{m}_H} = \frac{1}{w}\left[\frac{(1+r)(1+r_3)}{2q}+F\right]-F. \quad (47)$$

Figure 4 presents the performance ratios for the case of condenser flooding. Performance with flooding is a sensitive function of the value of n. An interesting aspect of this figure is that it indicates that the performance ratio can be greater than for the horizontal pipe with an artery.

11. DESIGN EXAMPLE

A heat pipe [5] has a wick of 9.5 mm O.D. and 4.75 mm I.D. It contains water at 100°C and its length characteristics are

$$z = 368.5\,\text{mm}$$
$$n = 1.67$$
$$r_3 = 0.31$$
$$z_c = 735\,\text{mm}.$$

Hence

$$q = 735/368.5 = 2.$$

Under test against gravity

$$Q_V = 98\,\text{W}.$$

Using equations (7) and (9)

$$Q_H = \frac{q}{q-1}Q_V = 196\,\text{W}.$$

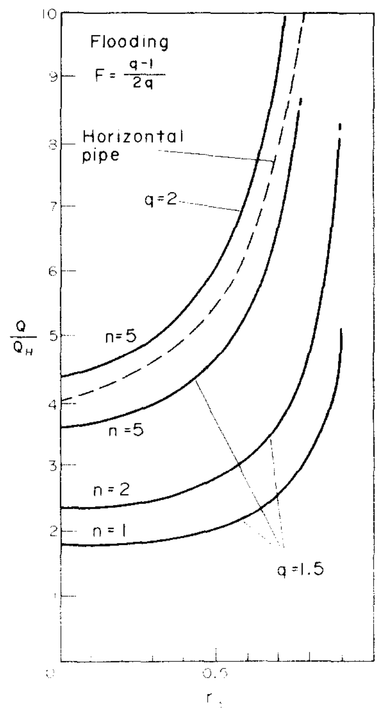

FIG. 4. Performance ratio Q/Q_H to a base of isothermal length ratio: influence of flooding.

Using Fig. 3, the artery system will increase the axial heat load to

$$Q = 196 \times 2.1 = 412 \text{ W}.$$

This is for $F = 0.25$. A check on the vapour pressure gradient shows that F is less than 0.25 so the design is conservative if 412 W is adequate. Where F is larger than given by equation (42) it is necessary to evaluate F using the actual friction factor, and estimate the hydraulic diameter allowing for the presence of the artery.

Using Fig. 4, if the condenser is allowed to flood, the performance will only be marginally improved for these conditions.

12. SUMMARY AND CONCLUSIONS

A novel header/artery system for heat pipes has been described. The advantages of the structure are: (a) simple and robust construction; (b) vapour and gases are automatically swept to the header; (c) priming of the artery is not dependent on the artery capillary radius; (d) the system is amenable to analysis, and; (e) a five-fold and more improvement in performance can be obtained.

A design procedure has been developed, and figures to facilitate preliminary design presented. Their use has been illustrated. Interestingly, performance against gravity can be in excess of that in the horizontal plane.

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CALODUCS AVEC DES SYSTEMES DE PILOTAGE ET D'IRRIGATION

Résumé—On décrit une nouvelle forme de système de pilotage et d'irrigation utilisable dans les caloducs, puis on développe les points thermiques et hydrauliques de leur conception. Le système artériel est simple et robuste et sans réseau capillaire. On discute le cas des fluides monophasiques dans le système artériel opérant contre la pesanteur. Dans cette orientation, la charge axiale peut dépasser celle dans le plan horizontal et elle peut être supérieure à dix fois la charge sans système artériel.

WÄRMEROHRE MIT VERTEILER- UND ARTERIEN-SYSTEMEN

Zusammenfassung—Es wird eine neue Form von Arterien- und Verteilersystemen zur Anwendung in Wärmerohren beschrieben und das thermisch/hydraulische Auslegungsverfahren abgeleitet. Das Arteriensystem ist einfach und robust, selbstansaugend ohne Verwendung von Kapillar-Arterien. Der Betrieb mit einphasigen, entgegen der Schwerkraft in den Arterien strömenden Flüssigkeiten wird diskutiert. Bei dieser Orientierung kann die aciale Transportleittung größer als im horizontalen Fall sein und mehr als das Zehnfache eines Aggregats ohne Arterien betragen.

ТЕПЛОВЫЕ ТРУБЫ, ИМЕЮЩИЕ СИСТЕМЫ КОЛЛЕКТОР-АРТЕРИЯ

Аннотация — Описан новый вид системы артерия-коллектор для тепловых труб и разработаны термогидравлические методы расчёта. Артерия является простой и прочной по конструкции и самозаполняющейся без помощи капиллярных трубок из сетки. Рассматривается работа тепловой трубы при течении в артерии однофазных жидкостей против сил тяжести. В этом случае аксиальная нагрузка может превосходить нагрузку в горизонтальной плоскости и более чем в десять раз превышать нагрузку в трубе без артерии.