AN INVESTIGATION OF THE PARAMETERS OF HEAT-EXCHANGING DEVICES USING THE VAPOR-LIFT MECHANISM

V. B. Eliseev, S. N. Ostapchuk,
and A. N. Spiglazov

UDC 621.643.2:536.24

An analysis has been made of the conditions of heat transfer which makes it possible to determine the optimum range of working temperatures.

One of the technical solutions for transferring heat in the direction of the force of gravity is the use of a loop with a vapor-lift mechanism for pumping the heat transfer medium such as is described in [1, 2]. No motors or pumps are used in the device for circulating the heat transfer medium, but many of the positive features of a heat transfer loop are retained. The purpose of the present paper is to study the heat transfer in such a device and to determine its most efficient conditions.

A sketch of the device is shown in Fig. 1. When heat is supplied into zone 1, vapor bubbles are generated, which enter the vapor-lift tube 2 and lift liquid with themselves and transfer it into zone 3. Here the vapor condenses, liberating the heat of vaporization at the walls of the condenser. The liquid transported by the vapor and the condensate flow downwards and increase the level of the liquid in the left-hand branch, which leads to motion of the liquid through the loop. The liquid is cooled by several degrees in the heat exchanger 4 with heat removal as a result of the heat capacity of the heat transfer medium.

In the vapor-lift tube the vapor bubbles occupy the entire cross-section of the tube, dividing the liquid into separate zones alternating with vapor zones. In order to counterbalance the liquid in the system the height of the vapor–liquid column must be greater than that of the column of continuous liquid. In order to raise some quantity of liquid through the height of the vapor-lift tube h_1 the total volume of the gas zones must equal the volume of this part of the tube, and for motion to begin it must somewhat exceed the latter.

From the point of view of the efficiency of pumping the liquid the volumetric ratio of the gas and liquid zones in the total stream is of great importance. This can be expressed in the form

$$G_L = nG_V. \tag{1}$$

The proportionality coefficient n depends on many factors: the characteristics of the heat transfer medium, the constructional features, and the operating conditions. For example, in the simplest case, such as is shown in Fig. 1, the coefficient n depends on the

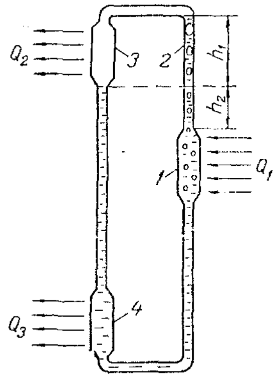
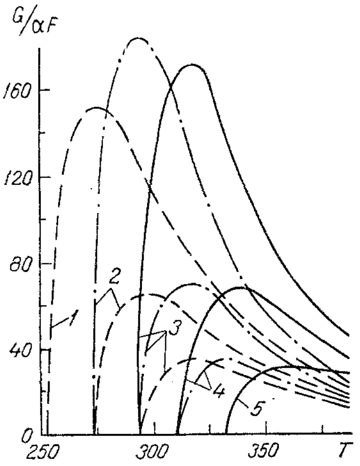


Fig. 1. Sketch of heat exchanging device with vapor-lift circulation of the heat transfer medium: 1) evaporator; 2) vapor-lift tube; 3) condenser; 4) heat exchanger; Q_1) input heat flux; Q_2 and Q_3) output heat fluxes transferred by the vapor and liquid, respectively.

All-Union Scientific-Research, Design-Construction, and Technological Institute for Power Sources, Moscow. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 56, No. 1, pp. 44-48, January, 1989. Original article submitted August 10, 1987.

 0022-0841/89/5601-0034$12.50



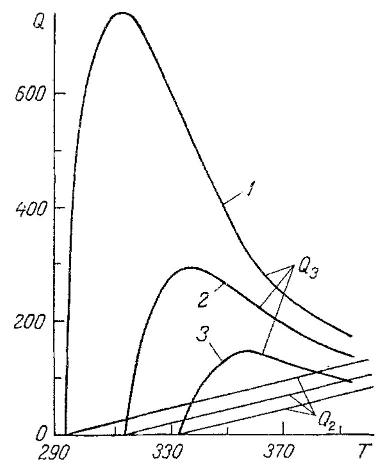


Fig. 2 Fig. 3

Fig. 2. Flow rate of the heat transfer medium G (cm^3/sec) as a function of the working temperature T (K) and various sink temperatures T_0 for αF = 1 W/K; heat transfer media: acetone, dashed lines; ethanol, dot-dashed lines; water, solid lines. 1) $T_0 = -20°C$; 2) $T_0 = 0°C$; 3) $T_0 = 20°C$; 4) $T_0 = 40°C$; 5) $T_0 = 60°C$.

Fig. 3. The temperature dependence of the heat fluxes transferred by water (Q_3) and by steam (Q_2), Watts, (with cooling of the water in the heat exchanger by $T - T_2 = 1$ K and with αF = 1 W/K): 1) with $T_0 = 20°C$; 2) with $T_0 = 40°C$; 3) with $T_0 = 60°C$.

location at which the bubbles are introduced into the vapor-lift tube h_2. It is assumed that in order for motion to occur vapor bubbles are introduced into the tube in a quantity slightly larger than h_1. If they are introduced not far below the surface of the liquid ($h_2 < h_1$) the volume of liquid rising with them will be small. Introducing the same bubbles at a lower level increases the volume of liquid transferred ($n > 1$ when $h_2 > h_1$). Thus, theoretically it is possible to have pumping with $n > 1$, but in practice this occurs only in the early stages when the liquid begins to boil with the smooth liberation of bubbles. With the onset of violent boiling the mechanism of liquid transfer changes.

The column of liquid h_2 is rapidly thrown into the condenser by the excess of vapor bubbles. Because of the large inertia, the equilibrium of the liquid in the system is established with some delay (from fractions of a second to several seconds). During this time it will have been possible to transfer from the evaporation zone to the condensation zone a quantity of vapor which exceeds in volume the quantity of liquid transferred by tens or hundreds of times.

However, by taking suitable constructional steps it is possible to use the intensive conditions more effectively. For example, if the evaporator is placed horizontally or at some angle then an accumulation of vapor will occur in the evaporator up to some excess pressure (fixed by h_1), followed by efflux of the liquid. The vapor bubble completely frees the evaporation zone of liquid, and further evaporation does not occur. It is not difficult to see that the coefficient n will be close to unity here. The mechanism for the carryover of the liquid in this example is somewhat different, but in principle, the path to its analysis remains similar.

The heat flux Q_1 which is introduced in the heating zone 1 leaves through the condenser 3 (Q_2) and the heat exchanger 4 (Q_3): $Q_1 = Q_2 + Q_3$.

It is of interest to consider the relationship between the components. The assumptions are made that the vapor obeys the ideal gas law, the physical properties of the heat transfer medium are independent of the temperature, etc., all of which are customary for calculating heat pipes.

From the equality of the heat supplied and removed at the walls of the condenser it is found that

$$Q_2 = r\rho_V G_V = \alpha F (T - T_0). \quad (2)$$

The temperature of the condenser wall T is assumed to be equal to the vapor temperature.

The heat flux transferred in the heat exchanger is given by

$$Q_3 = c\rho_L G_L (T_1 - T_2). \qquad (3)$$

The temperature T_1 is assumed to be equal to T.

By taking the ratio of Eq. (3) to Eq. (2) and making use of Eq. (1), it is found that

$$\frac{Q_3}{Q_2} = \frac{nc\rho_L (T - T_2)}{r\rho_V}.$$

It is assumed that the temperature of the liquid passing through the heat exchanger 4 is reduced by a total of 1 K, i.e., that $T - T_2 = 1$ K. In this case (with n = 1) numerical evaluations show that the value of Q_3 in some cases exceeds that of Q_2. Depending on the density of the vapor, if it is not large but is sufficient for pumping the liquid, Q_3 can be more than ten times larger than Q_2.

Let us now determine the dependence of the volumetric vapor flow rate G_V on the vapor temperature, assuming that it remains constant over the path from the evaporator to the condenser.

By using the equation of state for an ideal gas it is found that

$$\rho_V = \frac{PM}{TR}. \qquad (4)$$

The dependence of the saturated vapor pressure P on the temperature T can be obtained from the Clapeyron equation in the form

$$P = 101{,}33 \exp \frac{rM}{R}\left(\frac{1}{T_K} - \frac{1}{T}\right) \text{ [kPa]}. \qquad (5)$$

From Eqs. (2), (4), and (5) the temperature dependence of the vapor flow rate G_V can then be represented as

$$G_V = 9{,}87 \frac{\alpha F (T - T_0) TR}{rM \exp \frac{rM}{R}\left(\frac{1}{T_K} - \frac{1}{T}\right)} \text{ [cm}^3\text{/sec]}. \qquad (6)$$

By carrying out an analysis of the function (6) ($dG_V/dT = 0$) it is found that the maximum value of G_V occurs at the temperature

$$T_{max} = \frac{\left(T_0 + \frac{rM}{R}\right) - \sqrt{\left(T_0 + \frac{rM}{R}\right)^2 - 8\frac{rM}{R}T_0}}{4}. \qquad (7)$$

The function (6) is represented in the form

$$\frac{G_V}{\alpha F} = 9{,}87 \frac{(T - T_0) TR}{rM} \exp \frac{rM}{R}\left(\frac{1}{T} - \frac{1}{T_K}\right) \qquad (8)$$

in Fig. 2.

The expression $G_V/\alpha F$ is the ratio of the volumetric flow rate of the vapor to the heat transfer capability of the condensation zone, i.e., the characteristics of the heat sink. The curves in Fig. 2 were calculated for various temperatures T_0 and various heat transfer media with $\alpha F = 1$ W/K. The functions are characterized by clearly marked maxima. T_{max} depends only on the sink temperature T_0 since the remaining terms in Eq. (7) are constant. The product rM is the molar heat of vaporization. For most heat transfer media in the range 0-100°C the values of this product are similar. Thus, for example, for water, ethanol, acetone, etc., as the heat transfer media T_{max} is displaced by 20-25 K from T_0 (Fig. 2).

Making use of Eqs. (1) and (8) the heat flux transferred by the heat exchanger is found to be

$$Q_3 = \rho_L cnG_V(T - T_2) = 9{,}87\rho_L cn\alpha F (T - T_2)(T - T_0)\frac{TR}{rM} \exp\frac{rM}{R}\left(\frac{1}{T} - \frac{1}{T_K}\right).$$

The heat transfer conditions both in the condenser and in the heat exchanger influence Q_3. This function also has a maximum which shifts towards higher temperatures as T_2 is decreased. For example, when $T_2 = T_0$ the maximum of the function occurs for water in the region of T = 373 K.

With the cndition $(T - T_2) = 1$ K, the function $Q = f(T)$ will differ from $G_V/\alpha F = f(T)$ only by a constant multiplier. The function is given in Fig. 3 for water with n = 1 and various values of T_0. For comparison, Eq. (2) is given on the same plot. Under the conditions of maximum pumping (T_{max}), Q_3 exceeds Q_2 by several orders of magnitude.

As the working temperature T increases the fraction of the heat transferred by the liquid decreases, while that transferred by the vapor increases. At some point the transfer of heat by the vapor begins to predominate and the device undergoes a transition to the operating conditions of a heat pipe.

The heat exchanger 4 (see Fig. 1) can be connected separately to the condenser 3 to any heat sink, by which it can be cooled even below T_0 ($T_2 < T_0$). In this case the ratio of Q_3 to Q_2 becomes greater than 100, i.e., the main heat flux is transferred by the circulation of the liquid phase of the heat transfer medium.

NOTATION

G, volumetric flow rate; Q_1, Q_3, Q_3, heat fluxes in the evaporator, condenser, and heat exchanger; r, latent heat of vaporization; α, heat transfer coefficient of condenser; F, surface area of condenser; c, heat capacity; ρ, density; T, T_1, T_2, T_0,* temperature of the condenser walls, the surrounding medium, and the liquid at the inlet and outlet of the heat exchanger; P, pressure; M, molecular weight; R, universal gas constant; n, proportionality coefficient of the volumetric contents of vapor and liquid in the vapor-lift tube; h_1, h_2, height of the vapor-lift tube above the liquid level in the system and from the evaporation zone to the liquid level. Subscripts: V, vapor; L, liquid; K, boiling; max, maximum.

LITERATURE CITED

1. Heat Pipe: USSR Patent No. 637615, International Patent Classification [MKI] F28D15/00.
2. "Thermal system for circulating a fluid heat transfer medium," European Patent Application No. 0043227, International Patent Classification [MKI] F24J3/02.

THERMAL DIFFUSION RATIO OF SATURATED VAPOR–GAS MIXTURES

M. Mamedov

UDC 533.271:536.23

A theoretical method of determining the temperature dependence of the thermal diffusion ratio of saturated vapor – gas mixtures is proposed.

The known experimental methods of determining the value of k_T cannot be used for saturated vapor–gas mixtures. The relationship provided by the rigorous molecular kinetics theory cannot be used reliably for calculating the value of k_T without checking its adequacy experimentally.

Our aim was to develop a theoretical method for determining the thermal diffusion ratio of saturated vapor–gas mixtures on the basis of the phenomenological linear equations of the thermodynamics of irreversible processes.

An experimental method for determining the thermal diffusion ratio of saturated vapor–gas mixtures was proposed in [2]. Its essence consists in securing a process of simultaneous evaporation of a liquid into a vapor–gas medium and condensation of its vapor in an enclosed volume under diffusion conditions. The flat evaporation and condensation surfaces are parallel to each other. A Stefanian flow is contemplated. The theoretical premise of the method involves the use of linear phenomenological equations:

$$J_{1z} = -\frac{\mu_1}{\mu}\frac{\rho D}{1-x_1}\left(\frac{dx_1}{dz} - \frac{k_T}{T}\frac{dT}{dz}\right), \qquad (1)$$

Solntse Scientific and Industrial Association, Academy of Sciences of the Tadzhik SSR, Ashkhabad.

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 56, No. 1, pp. 48-54, January, 1989. Original article submitted July 16, 1987.

0022-0841/89/5601-0037 $12.50