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THERMAL EFFICIENCY OF MANIFOLD-HEAT-PIPE HEAT EXCHANGERS

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The thermal efficiency of manifold-heat-pipe heat exchangers is considered. A method for predicting the efficiency and an analysis of effects of various factors of the actual process are presented.

Heat-pipe heat exchangers (HPHE) are state-of-the-art heat exchangers that have found wide application owing to the possibility of separating the zones of heat supply and removal, minimal thermal resistance of heat pipes, and making the geometric shape of the latter more convenient in accordance with the characteristics of the heat-exchanging media. Among the variety of HPHE we should distinguish manifold-heat-pipe heat exchangers (MHPHE), which have recently become popular. One can become familiar with MHPHE in more detail in [1], which gives a review and particular results of investigating manifold heat pipes (MHP). This study also showed that issues related to characteristics and operation of MHPHE have not been explored adequately.

Equations obtained in [2, 3] permit determination of the thermal efficiency (hereinafter referred to as efficiency) of both heat-pipe and thermosiphon heat exchangers. The analysis presented is based on assumptions that regard HPHE as conventional heat exchangers, without account for internal transfer processes in heat pipes. Study [4] gave equations for predicting the efficiency of heat exchangers with forced circulation of an intermediate heat-transfer agent; on the basis of these equations optimal parameters of the heat exchangers were identified, in particular, circulation rates of the intermediate heat-transfer agent and distributions of the total heat-transfer surface between two heat exchangers.

In MHPHE, the circulation rate of the intermediate heat-transfer agent depends on the difference in liquid phase levels in the evaporator and condenser, thermophysical properties of the heat-transfer agent, length of the vapor and liquid channels, and other factors. Such heat transfer systems differ from forced-circulation heat exchangers in that they have no circulation pump, and the driving force is gravitation. Heat transfer is effected due to transfer of the latent heat of evaporation in reversible phase transitions (condensation and evaporation) (see Fig. 1). We resort to the approach to evaluating the efficiency of MHP heat exchangers proposed in [4] in reference to systems with an intermediate heat-transfer agent.

In most MHPHE, the cold and hot flows are uniphase, and external heat exchange between them and the MHP is due to convection and radiation. The heat transfer coefficients are appreciably lower in this case than those attained in boiling and condensation of the intermediate heat-transfer agent inside the MHP. The change in the temperature of the heat-transfer surfaces along the circulation path of the intermediate heat-transfer agent in the MHP is insignificant in comparison with the temperature drops over the operating external mass fluxes [1]. The above permits the thermal resistance of the MHP to be disregarded. This assumption makes it possible to consider the water equivalent of the intermediate heat-transfer agent in the MHPHE to be much greater than the water equivalent of the external heat-transfer agents ($W_{\text{int}} \gg W_h$, W_c), which is determined by phase transitions in the MHP. In view of the foregoing, the efficiency of the MHP heat exchanger can be determined as follows [4].

In the situation with counterflow, parallel flow, or cross flow:

for $W_h < W_c$

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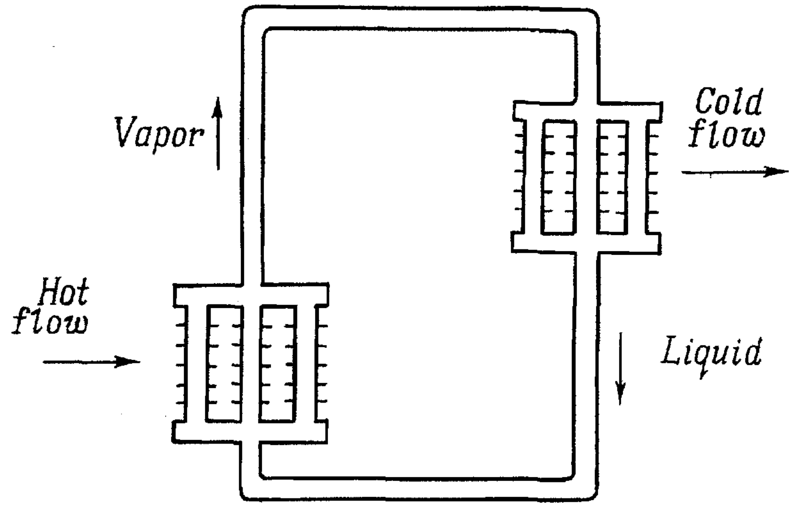

Fig. 1. Design of the MHP heat exchanger.

$$\varepsilon = \frac{1}{\frac{W_h}{W_c}\left[\frac{1}{1-\exp\left(-\frac{K_c F_c}{W_c}\right)}\right] + \left[\frac{1}{1-\exp\left(-\frac{K_h F_h}{W_h}\right)}\right]}, \tag{1}$$

for $W_h > W_c$

$$\varepsilon = \frac{1}{\frac{W_c}{W_h}\left[\frac{1}{1-\exp\left(-\frac{K_h F_h}{W_h}\right)}\right] + \left[\frac{1}{1-\exp\left(-\frac{K_c F_c}{W_c}\right)}\right]}, \tag{2}$$

and for $W_h = W_c$

$$\varepsilon = \frac{1}{\left[\frac{1}{1-\exp\left(-\frac{K_c F_c}{W_c}\right)}\right] + \left[\frac{1}{1-\exp\left(-\frac{K_h F_h}{W_h}\right)}\right]}, \tag{3}$$

If we write the number of heat-transfer units as $NTU = KF/W$ and the ratio of the water equivalents of the heat-transfer agents as $R = W_{min}/W_{max}$, Eqs. (1)-(3) can be written in the form

$$\varepsilon = f(R, NTU_h, NTU_c). \tag{4}$$

Figure 2 gives predictions from Eqs. (1)-(3) when $W_h \neq W_c$ and $R = 0.5$. Clearly, the efficiency of the MHP heat exchanger is a function of the ratio between the water equivalents of the "external" heat-transfer agents R and the numbers of transfer units of the external heat-transfer agents NTU_h and NTU_c in the evaporator and condenser.

The above predictions of the MHPHE efficiency are made with allowance for neglecting the internal thermal resistance. When the MHP operating mode is disturbed, for example, with heat transfer over a large distance etc., the internal thermal resistance of the MHP rises. In this case the internal thermal resistance of the manifold heat pipe should be taken into account.

Figure 3 presents an actual temperature distribution of heat-transfer agents in MHP heat exchangers. Generally, the direction of circulation of the intermediate heat-transfer agent in the MHP relative to the flow of the operating heat-transfer agents is cross. As Fig. 3a shows, the intermediate heat transfer agent in the MHP is

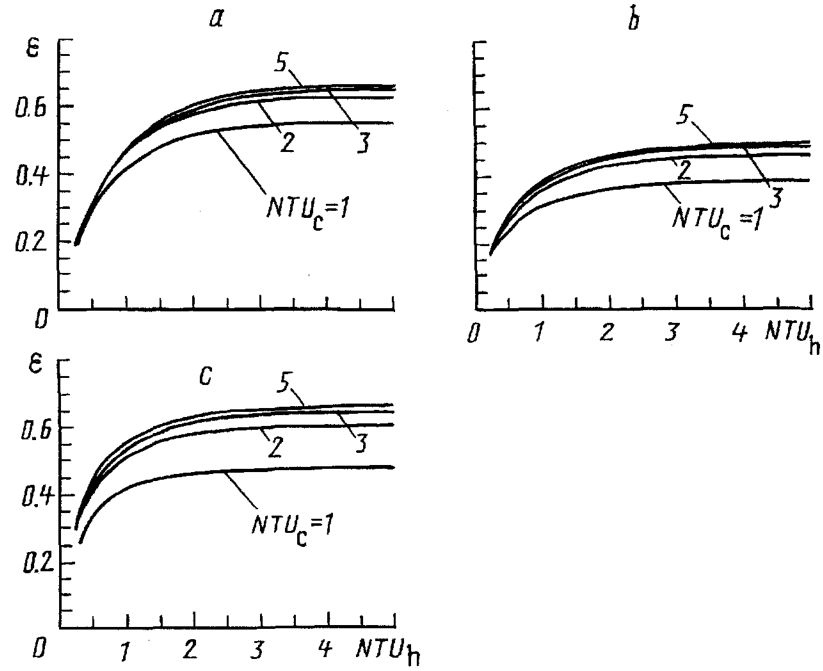

Fig. 2. Efficiency of the MHP heat exchanger: a) $W_c > W_h$; b) $W_c = W_h$; c) $W_c < W_h$.

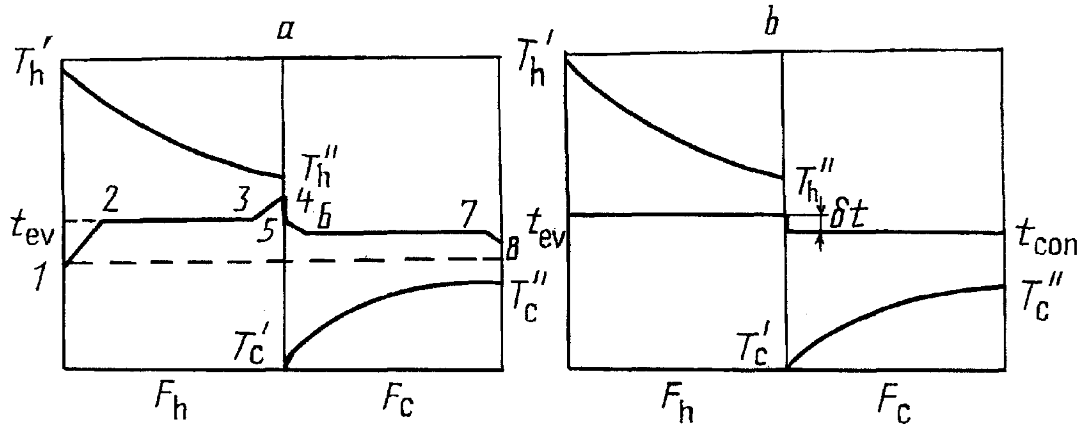

Fig. 3. Temperature distribution in the MHP heat exchangers: a) actual; b) model under consideration.

transported in the following way: first, supercooled heat-transfer agent is heated in the evaporator to saturation (1-2); the liquid evaporates and boils, transforming completely into a vapor (2-3); saturated vapor escapes the evaporation zone, with subsequent heating in some cases (3-4); afterwards the vapor flows along the channel, where partial heat loss may occur (4-5); in the case of superheated vapor, the latter enters the condenser and is cooled to saturation (5-6), giving off heat and transforming completely into a liquid phase (6-7); next, the heat-transfer agent is supercooled and leaves the condenser (7-8); liquid is returned to the evaporator via the channel with partial heat loss (8-1). Regardless of the superheating, supercooling, or heat losses along the liquid and vapor channels, the main heat transfer in the MHP is effected by phase transitions; therefore, it is sometimes assumed that there is no temperature variation in the evaporator and the condenser and there are no heat losses in the channels. Actually, there is a temperature difference ($\delta t = t_{ev} - t_{con}$) between the evaporator and the condenser owing to internal thermal resistances in the MHP (Fig. 3b).

Hence, the efficiency of the MHP condenser may be represented in terms of the condenser temperature by the following equation:

$$\varepsilon_c = \frac{T_c'' - T_c'}{t_{con} - T_c'}. \tag{5}$$

Using the evaporator temperature t_{ev} we write expression (5) as

$$\varepsilon_c' = \frac{T_c'' - T_c'}{t_{ev} - T_c'}. \tag{6}$$

From Eqs. (5) and (6) we may obtain a relationship between ε_c and ε_c':

$$\varepsilon_c' = \frac{\varepsilon_c}{1 + \dfrac{\delta t}{t_{con} - T_c'}}. \tag{7}$$

We now present an analysis of MHPHE efficiency for the case $W_h > W_c$. Other cases can be determined similarly:

$$\varepsilon' = \frac{T_c'' - T_c'}{T_h' - T_c'}. \tag{8}$$

With a view to Eq. (5) we get

$$t_{con} - T_c' = (T_h' - T_c')\frac{\varepsilon'}{\varepsilon_c}. \tag{9}$$

On the other hand [4], the efficiency of the MHP heat exchanger is determined by the following equation:

$$\varepsilon' = \frac{1}{\dfrac{W_c}{W_h}\dfrac{1}{\varepsilon_h} + \dfrac{1}{\varepsilon_c'}} = \frac{1}{\dfrac{W_c}{W_h}\dfrac{1}{\varepsilon_h} + \left(1 + \dfrac{\delta t}{T_h' - T_c'}\right)\dfrac{1}{\varepsilon'}}. \tag{10}$$

Substituting ($t_{con} - T_c'$) from Eq. (9) into Eq. (10) results in

$$\varepsilon' = \frac{1}{\dfrac{W_c}{W_h}\dfrac{1}{\varepsilon_h} + \dfrac{1}{\varepsilon_c} + \left(1 + \dfrac{\delta t}{T_h' - T_c'}\right)\dfrac{1}{\varepsilon'}}. \tag{11}$$

We rearrange Eq. (11), denoting $\Delta T = T_h' - T_c'$:

$$\varepsilon' = \left(\frac{1}{\dfrac{W_c}{W_h}\dfrac{1}{\varepsilon_h} + \dfrac{1}{\varepsilon_c}}\right)\left(1 - \frac{\delta t}{\Delta T}\right). \tag{12}$$

Taking into account that the first factor in the expression derived represents the efficiency of the MHP heat exchanger ε without account for the internal thermal resistance of the MHP, we may write relation (12) as

$$\varepsilon' = \varepsilon\left(1 - \frac{\delta t}{\Delta T}\right). \tag{13}$$

For the cases $W_h < W_c'$ and $W_h = W_c$ we may also obtain an equation similar to Eq. (13).

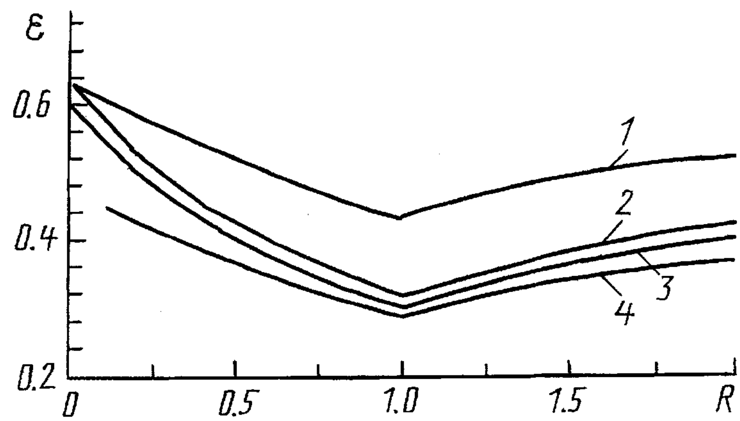

Fig. 4. MHPHE efficiency vs the ratio of the water equivalents of the heat-transfer agents: 1) traditional heat exchanger; 2) MHP heat exchanger at $\delta t/\Delta T = 0$; 3) MHP heat exchanger at $\delta t/\Delta T = 0.05$; 4) heat exchanger with forced circulation of an intermediate heat-transfer agent.

Knowing the internal thermal resistance, we can calculate the coefficient $(1 - \delta t/\Delta T)$ from Eq. (13). Comparing the obtained equations with results of [4], we may conclude that, if the value of W_c/W_{int} (or W_h/W_{int}) is approximately equal to $\delta t/(T_c'' - T_c')$ and $T_c'' - T_c' = \varepsilon(T_h' - T_c') = \varepsilon\Delta T$, Eq. (13) can be derived.

As is evident from Eq. (13), a rise in thermal resistance of the MHP will decrease the efficiency of the MHP heat exchanger. We consider the assumptions adopted in predicting the efficiency of the MHP heat exchanger with allowance for the internal thermal resistance.

1. Supercooling of the liquid arriving at the evaporator.

From the expression $\varepsilon_h = (T_h' - T_h'')/(T_h' - t_{cond})$ it follows that the evaporator efficiency deteriorates with decreasing t_{ev}, which lessens the efficiency of the MHP heat exchanger.

2. Superheating of the vapor entering the condenser.

It follows from the expression $\varepsilon_c = (T_c'' - T_c')/(t_{con} - T_c')$ that the condenser efficiency will decrease with rising t_{con}; the efficiency of the MHP heat exchanger also decreases.

3. Account for the heat losses in the vapor and liquid channels.

With increasing heat loss in the vapor and liquid channels the efficiency of the condenser and the MHP heat exchanger will diminish.

4. Account for the distance of thermal energy transport (the distance between the evaporator and the condenser).

The hydrodynamic drag rises as the heat transfer distance increases. Accordingly, the pressure drop also rises. This leads to an increase in the temperature difference between the evaporator and the condenser δt. Here, as follows from expression (13), the MHPHE efficiency deteriorates.

Apart from the noted features, with account for expression (4), the efficiency of the MHP heat exchanger is affected also by the ratio of the water equivalents of the external flows of the heat-transfer agents $R = (W_{min}/W_{max})$. Figure 4 presents the efficiency of the MHP heat exchanger vs the ratio between the water equivalents of the external heat-transfer agents R and the coefficient $\delta t/\Delta T$ with allowance for the internal thermal resistance.

As is seen from Fig. 4, from the standpoint of efficiency MHP heat exchangers rank below classical heat exchangers but have advantages over heat exchangers with forced circulation of an intermediate heat-transfer agent. The minimum efficiency for all types of heat exchangers corresponds to equality of the water equivalents of the heat-transfer agents, i.e., $R = 1$.

In conclusion we should note that the analysis presented is based on the normal operating mode of the MHP, when the heat fluxes do not reach critical values. Under such conditions the predictive method presented can be employed for determining MHPHE efficiency.

NOTATION

ε, thermal efficiency of the heat exchanger; W, water equivalent of the heat-transfer agent; K, total heat-transfer coefficient; F, total heat-transfer surface on one side of the heat exchanger; R, ratio of the water equivalents of the heat-transfer agents; NTU, number of heat transfer units; T', T'', entry and exit temperatures of the external flows; t, temperature of the internal heat-transfer agent. Subscripts: h, hot side; c, cold side; int; intermediate heat-transfer agent; con, condenser; ev, evaporator.

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