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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 ($W_{int} >> W_h$, 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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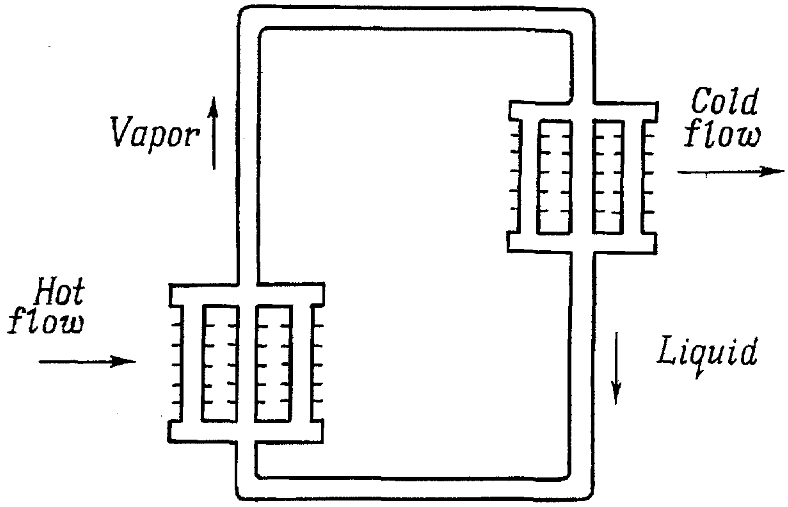

Fig. 1. Design of the MHP heat exchanger.

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

for $W_h > W_c$

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

and for $W_h = W_c$

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

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