EFFECT OF HEAT-PIPE PARAMETERS ON THE EFFICIENCY OF A HEAT EXCHANGER

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The special features of designing a heat-pipe heat exchanger are considered. A procedure is proposed for selecting the parameters of a standardized heat pipe for heat exchangers.

At present heat exchangers based on heat pipes are finding widespread applications in various technological fields. Their use is particularly effective in heat utilization systems which are intended for operation in the areas of moderate and low temperatures. The advantages of heat exchangers based on heat pipes in such systems compared with other heat exchanger constructions have been dealt with in detail in [1, 2] and have been confirmed by usage tests.

A feature of the construction of this type of heat exchanger consists of the fact that it is designed on the basis of a single standardized element, which is the heat pipe. As a result, the design calculations must be directed towards the determination of the parameters which will satisfy the requirements with respect to the heat exchanger efficiency and which at the same time characterize the construction of the standardized heat pipes. The available procedures for designing recuperative heat exchangers, including heat exchangers based on heat pipes, do not make it possible to achieve these objectives.

In order to determine the efficiency of a heat exchanger it is necessary to have a relationship which establishes the dependence of this quantity on the parameters which characterize the construction of the heat pipe. These will include the lengths of the contact zones of the heat pipe with the heat transfer medium being cooled and with that being heated, the tube diameter supporting the ribbing, the rib coefficient, the internal diameter of the tube, and the geometric dimensions of the ribs, on which the outside heat transfer coefficients in the zones indicated above depend; the heat transfer coefficients for evaporation and condensation, which depend on the geometric shapes and dimensions of the internal surfaces of the pipe and on the choice of the working liquid for the evaporation—condensation cycle; and the thermal conductivities of the shell of the heat pipe and its ribbing.

The ensemble of parameters enumerated above makes it possible to uniquely determine the thermal resistance of the heat pipe of the heat exchanger:

$$\begin{gathered}
R_1=\frac{1}{\pi\alpha'_{\text{out}}l_e d\varepsilon},\qquad R_2=\frac{\ln(d_{\text{pi}}/d)}{l_e\lambda},\\
R_3=\frac{1}{\pi\alpha_e l_e d_{\text{pi}}},\qquad R_4=f(d_{\text{pi}},\ Q),\\
R_5=\frac{1}{\pi\alpha_c l_c d_{\text{pi}}},\qquad R_6=\frac{\ln(d_{\text{pi}}/d)}{l_c\lambda},\\
R_7=\frac{1}{\pi\alpha''_{\text{out}}l_c d\varepsilon},\qquad R=\sum_{i=1}^{7}R_i,
\end{gathered}\tag{1}$$

the value of which can be used for evaluating the efficiency of the construction of the evaporation—condensation device. Then by using for the individual rows, each consisting of m heat pipes, the concept of a temperature efficiency which was introduced in [3] for evaluating the construction of heat exchangers, it is possible after eliminating the temperature by using the heat transfer equation and the heat balance to obtain the following relationship for calculating the efficiency of the row of the heat exchanger:

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$$\eta_0 = \frac{2Mm}{2RMw_{\min} + (M+1)m}, \quad M = \frac{w_{\max}}{w_{\min}}. \tag{2}$$

An analysis of the heat transfer process in heat exchangers with the same efficiencies of all their n rows makes it possible to arrive at the relationship

$$\eta = 1 - \frac{M^{n-1}(1-\eta_0)^n}{(M-\eta_0)^{n-2}(M-\eta_0^2) + \eta_0\left[\sum_{i=1}^{n-2} M^i (M-\eta_0)^{n-i-2}\,(1-\eta_0)^{i+1}\right]}, \tag{3}$$

which, together with relationships (1) and (2), establishes a relationship between the efficiency of the heat exchanger and the parameters of the individual heat pipes. In the case where the water equivalents of the heat transfer medium being heated w_h and that being cooled w_c are equal, Eq. (3) can be simplified, and reduces to the relationship obtained in [4]:

$$\eta = \frac{n\eta_0}{1+(n-1)\eta_0}. \tag{4}$$

In the design calculations of the heat exchanger its efficiency and the permissible pressure drop in the heat exchange network are input quantities. Then if Eqs. (3) and (4) are supplemented by a relationship for calculating the pressure drop during the motion of the heat transfer medium through the heat exchanger

$$\Delta P = f(w_{\max}, m, n), \tag{5}$$

a system of two equations is obtained the solution of which with respect to the unknowns m and n makes it possible to find the required number of heat pipes of the given length and to determine their arrangement.

The number of heat pipes required for the construction of a heat exchanger with specified thermotechnical characteristics and their arrangement depend on the diameter, length, and internal resistance of the heat pipes, and the height, thickness and pitch of their ribs. The weight of the heat pipes, and hence the mass of the entire heat exchanger, also depend on these parameters. Thus, if a change is made in the parameters characterizing the construction of the heat pipes while keeping the thermotechnical characteristics of the heat exchanger unchanged, then the number of heat pipes in the heat exchanger will be changed, and so will their weight, i.e., the quantities which determine the cost of the heat exchanger.

An analysis which has been carried out has shown that the costs of manufacturing the heat pipes of a heat exchanger with specified thermotechnical characteristics can be described by the parameter

$$Z = mn\left(\frac{C_{\text{тт}}}{C_{\text{м}}} + P\right), \tag{6}$$

which depends on the weight of the heat pipes P and the ratio of the cost of their manufacture C_{TT} to the cost of unit weight of the shell material C_M, and which should be a minimum for the optimum design.

The calculation procedure can be discussed by the example of selecting a heat pipe construction for the heat exchanger of a ventilation system calculated for an air interchange of 3.7 kg/sec. It will be assumed that a heat exchanger is required with a temperature efficiency of 0.55 and with a permissible pressure drop in the heat transfer network of 150 Pa. The lengths of the evaporation and condensation zones of the heat pipes, which are manufactured from aluminum, are taken to be equal to 1 m.

Let us carry out a comparison of heat exchangers which differ in the diameter and rib parameters of the heat pipes. To do this, the arrangement of the heat pipes in the pipe layout, the total number of tubes, and their weight are determined from the relationships given above. After this, an evaluation is made of various constructions. The parameter Z is used as the criterion for the evaluation; this criterion should have a minimum value. It is obvious that such a comparison will lead to large numbers of computations, and that it is necessary to use computer techniques.

An analysis of the results of calculations of numerous constructions of heat exchangers differing in the heat pipe parameters showed that as the pitch of the ribs increased, the total number of heat pipes in the heat exchanger also increased. In this case the nature of

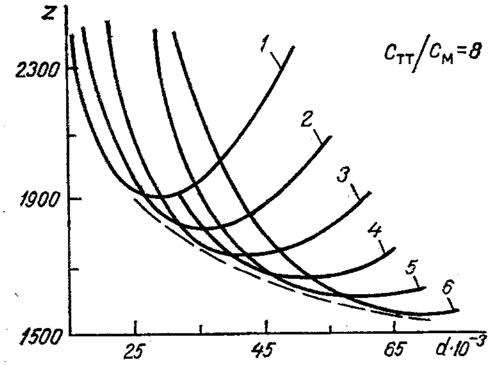

Fig. 1

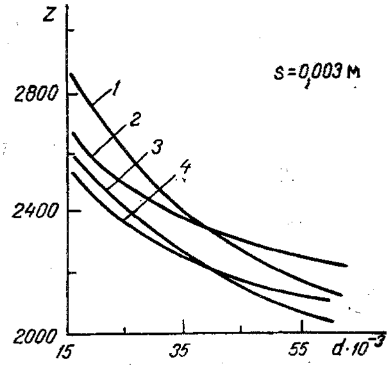

Fig. 2

Fig. 1. The parameter Z as a function of the tube diameter supporting the ribs for various values of the ratio of the internal to the external diameters of the ribs: 1) 0.3; 2) 0.35; 3) 0.40; 4) 0.45; 5) 0.50; 6) 0.55. The value of d is given in meters.

Fig. 2. The parameter Z as a function of the heat pipe diameter supporting the ribs for various values of the rib thickness: 1) 0.0001 m; 2) 0.0005 m; 3) 0.0002 m; 4) 0.003 m.

the change in the weight of the heat tubes was such that in all cases the costs of the heat exchangers increased as the rib pitch increased. The minimum value of the pitch is determined in practice by the technological possibilities in manufacturing the ribbing and the usage conditions, and varies in the range of 1.5-3.0 mm for the most widely encountered types of ribbed surfaces.

Having selected the rib pitch to be equal to 3 mm, the results of the calculations of various heat pipe heat exchangers were used to build up the dependence of the parameter Z on the diameter supporting the ribs for various ratios of the internal to the external diameters of the ribs (the case of individual circular ribs (fins) is considered). As can be seen from Fig. 1, this relationship is characterized by the presence of a minimum, the position of which depends not only on the ratio of the internal to the external diameters of the ribs but also on the cost of constructing the heat pipes. The presence of the minimum is related to the opposite changes in the weight and number of the heat pipes as the diameter supporting the ribs is varied. An analysis of these relationships developed for a large number of heat exchangers with differing constructions of the heat pipes made it possible to establish the following points: a staggered arrangement of the heat pipes in the heat exchanger permitted a lower cost of its construction to be reached if the geometric dimensions of the pipes and their ribbing are correctly selected; for a staggered arrangement and a value of the parameter $C_{TT}/C_M > 2$ the costs of manufacturing the heat pipes of the heat exchanger become smaller the larger the diameter of the heat pipe supporting the ribs is made. The ratio of the internal to the external rib diameters for which the cost for manufacturing the heat exchanger is a minimum is determined after selecting the diameter of the pipe supporting the ribs from the relationships represented in Fig. 1.

When the thickness of the ribs changes, the nature of the relationship between the parameter Z and the heat pipe diameter remains unchanged, as shown by numerous calculations. Consequently, if the pipe diameter and the height and pitch of the ribs are selected, then in order to determine the thickness of the ribs it is sufficient to analyze the nature of the change of the minimum value of the parameter Z as a function of the heat pipe diameter. Such a relationship has been set up for heat exchangers with ribs of different thicknesses (see Fig. 2). This relationship can be obtained after setting up the curves of the type shown in Fig. 1 for each value of the rib thickness. For this purpose the envelope of the minimum values has been found, which is the required relationship (dashed line in Fig. 1).

As can be seen from Fig. 2, for a staggered tube arrangement the heat exchanger of the air conditioning system being considered has a minimum construction cost if the rib thickness is about 0.2 mm for tube diameters greater than 35 mm or is about 0.3 mm for tubes of diameter less than 35 mm.

The recommendation for selecting the rib thickness, as shown by the analysis, is connected with the particular value of the rib pitch. This is explained by the fact that the value of the thickness depends on the pitch at which the construction costs of the heat exchanger are at a minimum.

Thus, the relationships given in the present paper for calculating the heat exchanger efficiently make it possible to select the parameters of standardized heat pipes which ensure a minimum cost for the heat exchanger construction. Simultaneously the arrangement of the heat pipes is determined which satisfies the requirements with respect to the efficiency of the heat exchanger and the permissible pressure drop in the heat exchange layout. In this case the procedure for the design calculations includes the following steps: the selection from the working conditions of the working fluid for the heat pipes the maximum possible length for the heat pipes, and the pitch of the ribbing; setting up for various ratios of the internal to the external diameters of the ribs relationships between the cost of constructing the heat exchanger and the diameter of the tubes supporting the ribs; and for the diameter selected, determining the rib heights, then the rib thickness and the spacing between the ribs.

NOTATION

l_e, length of the evaporation zone of the heat pipe; l_c, length of the condensing zone of the heat pipe; d, diameter of pipe supporting ribs; d_{pi}, internal diameter of pipe; ε, ribbing coefficient; α'_{out}, α''_{out}, external heat-transfer coefficients in the evaporation and condensation zones, respectively; α_e, heat-transfer coefficient in evaporator; α_c, heat-transfer coefficient in condenser; λ, thermal conductivity of shell of heat pipe and its ribbing; P, weight of heat pipe; C_{TT}, manufacturing cost of an individual heat pipe; C_M, cost of unit weight of the shell material.

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LIMITS OF APPLICABILITY OF THE THEORY OF A SINGLE-PHASE BOUNDARY LAYER USING CONDITIONS OF FREE-CONVECTIVE FROSTING

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The conditions of applicability of the approximation of a single-phase boundary layer with the formation of a condensate in the form of frost on a vertical flat surface are determined.

I know of no published studies of the limits of applicability of the theory of a single-phase boundary layer in the analysis of heat and mass transfer under conditions of precipitation of a solid condensate (frost) from a gaseous binary mixture (containing predominately inert components) on an isothermal surface. One of the main reasons for this is the uncritical use of correlations for the Nusselt and Sherwood numbers. The analytical basis for them is the theory of a single-phase (multicomponent) boundary layer, in spite of the difference in the temperatures (medium—isothermal surface) under which it is formed. Thus, for example, almost all theoretical approaches to the determination of the coefficient of heat and mass transfer under the conditions of formation of frost from moist air on isothermal surfaces were constructed for single-phase models [1]. On the other hand, in many experiments it was found that the water phase changes spontaneously even before the frost layer forms on it [2].

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