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EFFICIENCY OF AN AIR HEAT EXCHANGER

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Experimental and calculated dependences of the coefficient of thermal efficiency of a heat exchanger are presented for various effective diameters of the internal channels. It is shown that a region of a local maximum of the coefficient of thermal efficiency exists in the regime of a turbulent flow of the coolant.

Introduction. The thermal efficiency of a heat exchanger, i.e., its ability to dissipate the thermal energy of a coolant, is the main parameter of heat-exchange devices. Usually, this parameter is evaluated as the ratio between the thermal head ($T_{in} - T_{out}$) and the difference of the input temperature of the coolant (T_{in}) and the temperature of the cooling air (T_a), i.e.,

$$E = \frac{T_{in} - T_{out}}{T_{in} - T_a} . \qquad (1)$$

A special feature of heat exchangers with air used as the coolant consists in the fact that, in view of the substantially larger surface of contact with the cooling air compared to the surface of heat transfer in the internal channels, the thermal efficiency of the device is determined by the heat transfer in the internal channels. With a smaller heat-transfer surface of the internal channels, an increase in the heat flux to channel walls from the coolant can be achieved by means of intensification of the heat transfer. Methods for intensification of heat transfer in pipes by means of various inserts with elements affecting the turbulization of the boundary layer are well known [1, 2]. Intensification methods based on a change in the temperature gradient in the channels are much more poorly understood.

In what follows, we consider the possibility of increasing the thermal efficiency of a heat exchanger with constant external and internal surfaces by forming a higher temperature gradient at the internal wall of the channels. Experimental investigations are supplemented with a theoretical analysis of the heat-transfer process in the channels, which makes it possible to distinguish the governing geometric and hydrodynamic parameters.

Experiment. Measurements of the efficiency were carried out on a pipe-plate radiator fabricated from aluminum. The radiator is intended for cooling the supercharging air coming to an engine from a turbocompressor. Hot air is supplied to and removed from the internal channels of the heat exchanger by end fittings of the entrance and exit collectors. The internal diameter of the pipes was 8 mm. The heat-exchange plates were put on the pipes through flanged holes that controlled the distance between the plates. Thermal contact of the pipes with the plates was achieved by creating a guaranteed tightness of 0.35 mm. The effective diameter was changed by fitting in each pipe a (coaxial) rod 5 mm in diameter taking 90% of the pipe length, or a band swirler. The effective diameter of a pipe with a rod is determined from the ratio $d_e = 4F/\Pi$, where F is the area of the flow-passage cross section of the channel, and Π is the perimeter of the ring gap. In the case of a band swirler it was determined by the formula [1]

$$d_e = d_l\,(\pi d_l - 4\delta)/(\pi d_l + 2\,(d_l - \delta))\,,$$

where d_l is the internal diameter of the pipe and δ is the thickness of the band swirler.

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Investigations were carried out on a single radiator with various effective channel diameters: $d_e = 8$ mm (empty pipe), 3.2 mm (pipe with an internal rod), and 4.1 mm (pipe with a band swirler). The ratio between the area of the internal channels and the area of the surface of contact with the cooling air was equal to 0.033.

The radiator was cooled by external air using a ventilator. In the experiments, we measured the temperature of the air at the entrance to (T_{in}) and exit from (T_{out}) the radiator, the temperature of the cooling air (T_a) approaching the radiator, the flow rate of the hot air (G), and the pressure of the coolant at the entrance to (P_{in}) and exit from (P_{out}) the radiator.

Mathematical Model. As a first step toward modeling the heat transfer, we consider the heat transfer in a separate pipe of internal diameter R with a coaxial rod of smaller diameter R_1 fitted coaxially inside the pipe. We assume that the heat flux transferred from the coolant to the pipe walls and then to the heat-exchange plates is completedly removed by the approaching flow of cooling air. This assumption is in full agreement with the heat-exchanger design considered, in which the potentiality for removal of the heat flux by the cooling air substantially exceeds the magnitude of the heat flux released by the coolant on the channel walls. Then the equation of the steady-state convective heat transfer can be represented in the form [3]

$$v\partial_z T(z, r) = \Delta a T(z, r)\,, \tag{2}$$

where v is determined from the flow rate of the coolant and $T(z, r)$ is the temperature field. We will seek the solution of Eq. (2) subject to the boundary conditions

$$T(0, r) = T_{in}\,;\quad T(z, R) = T_a\,,\quad \nabla T(z, R_1) = 0\,. \tag{3}$$

We pass to the temperature $T(z)$ averaged over the pipe cross section, which is determined as follows:

$$T(z) = \frac{2}{R^2 - R_1^2}\int_{R_1}^{R} T(z, r)\, r dr\,. \tag{4}$$

By averaging Eq. (2) according to rule (4), we obtain, with allowance for the boundary conditions, the relationship

$$v\partial_z T(z) = \frac{2Ra}{R^2 - R_1^2}\nabla T(z, R)\,. \tag{5}$$

Following the definition, we will calculate approximately the value of the temperature gradient on the external boundary:

$$\nabla T(z, R) = \frac{T(z) - T_a}{R - R_1}\,. \tag{6}$$

By integrating (5) with allowance for (6), we obtain the following expression for the temperature drop $\Delta T(z) = T(z) - T_a$:

$$\Delta T(z) = (T_0 - T_a)\exp\left[\frac{2Raz}{(R^2 - R_1^2)\, v\,(R - R_1)}\right]. \tag{7}$$

Simple transformations in (7) yield a formula for calculation of the thermal-efficiency coefficient E:

$$E = \frac{\Delta T}{T_a - T_0} - 1\,. \tag{8}$$

For short heat exchangers, when the argument of the exponential in (7) is much less than unity, one can easily show that

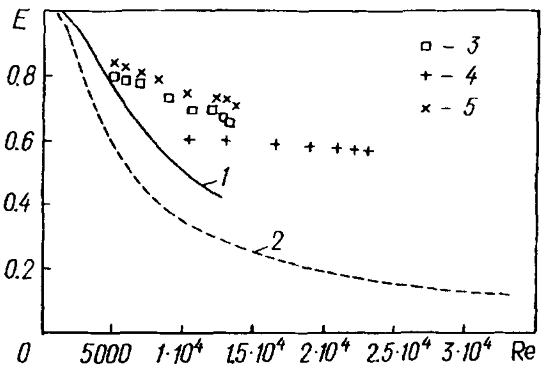


Fig. 1. Dependence of the coefficient of thermal efficiency E on Re and the effective channel diameter: 1, 2) calculation for a channel with diameter d_e = 3.2 and 8 mm, respectively; 3, 4, 5) experimental data for a channel with diameter d_e = 3.2, 8, and 4.1 mm, respectively.

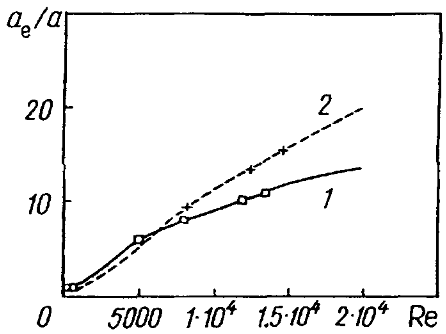
$$E = \frac{2Raz}{(R^2 - R_1^2)\, v\, (R - R_1)}, \tag{9}$$

where the expression $(R^2 - R_1^2)\pi v = Q$ is the air flow rate through a pipe of the heat exchanger.

It follows from (9) that for a constant volume flow rate of the coolant the quantity E is proportional to the length of the heat exchanger and its outer radius and is inversely proportional to the magnitude of the gap between the rod and the pipe. By decreasing the gap, we increase the temperature gradient and the efficiency of the heat exchanger. The value of the coefficient a entering into expression (9) equals the molecular thermal diffusivity of air. In the case of large flow rates of the coolant, a turbulent flow regime emerges in the channels (the Reynolds numbers determined from the effective channel diameter and mass-averaged velocity v exceed Re > 2500), for which the effective thermal diffusivity a_e cannot easily be determined without resorting to experimental data.

Results and Discussion. Experiments on evaluation of the quantity E were carried out with the same variation of the flow rate of the coolant for three different values of the effective pipe diameter. Experimental data are presented in Fig. 1 along with calculated E values obtained using values of the thermal-diffusivity coefficient a_e chosen so as to provide coincidence of the calculated E values with the experimental ones at the point Re = 5000 for d_e = 3.2 mm. This approach assumes that the mathematical model describes some arbitrary pipe whose efficiency already includes the effect of the inhomogeneity of the distribution of the coolant over the radiator pipes and the inhomogeneity of its cooling over its length. It should be noted that the chosen value of the coefficient a_e is also used for calculation of quantities E related to a pipe with an effective diameter of 8 mm. As is seen, the coefficient E decreases with increase in the Reynolds number. However, the calculations agree with the experimental values of E only at the point where the value of the effective thermal-diffusivity coefficient was chosen, i.e., at Re = 5000 for d_e = 3.2 mm. For an empty pipe (d_e = 6 mm), the calculated values of E are substantially smaller than the experimental ones, which points first of all to a dependence of the values of the effective coefficient a_e for channels of different diameters on the Reynolds number. Both the calculations and the experiment showed that the decrease in the effective pipe diameter due to the presence of an internal rod or a band swirler was accompanied by an increase in the efficiency of the heat exchanger. Twisting of the flow, even for a large pipe diameter, improved the efficiency of the heat exchanger by about 5% compared to the flow in a pipe with a rod. The observed increase in the discrepancy between the calculated and experimental E values can be explained by an increase in the effective thermal-diffusivity coefficient with increase in the Reynolds number.

Using experimental values of E and Eq. (9), we plotted dependences of the effective thermal-diffusivity coefficient on the Reynolds number for channels with diameters d_e = 8 and 3.2 mm (Fig. 2). The character of variation of the coefficients a_e and their values in the range of Re < 6000 are quite similar in both cases. Taking into account the fact that the considered dependences for a_e were obtained using two independent sets of experimental data, their approximate coincidence points to the fact that for values not exceeding Re = 6000, the heat transfer in a channel is localized in a region located near the internal channel surface. When the Reynolds



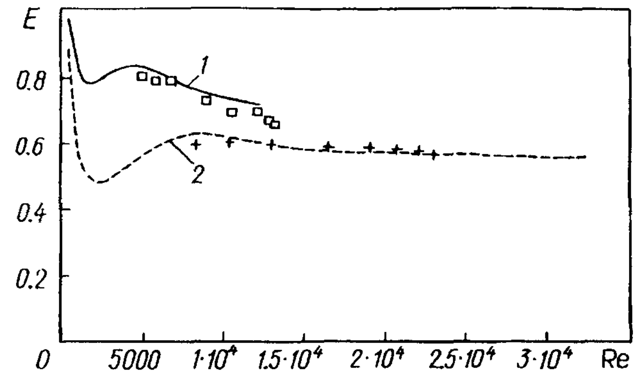


Fig. 2. Dependence of the dimensionless effective thermal diffusivity on Re: 1, 2) spline interpolation [4] of experimental data for channels of d_e = 3.2 and 8 mm, respectively.

Fig. 3. Dependence of E on Re and the effective channel diameter: 1, 2) calculations and experimental data for d_e = 3.2 and 8 mm, respectively.

number is increased, large-scale transverse modes develop in a channel with a large effective diameter, which intensifies the heat transfer. As a result, the coefficient a_e grows faster for an empty pipe.

Values of E calculated with allowance for the dependences of the coefficient a_e presented in Fig. 2 are plotted in Fig. 3.

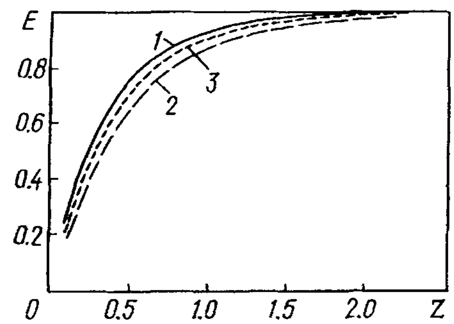
First of all, we should note a basic change in the character of the dependence of the efficiency coefficient E on the Reynolds number. The values of the quantity E have a maximum in the region of small Re (laminar regime of coolant flow), then decrease, and reach a minimum at Re = 1000. In the intermediate region of coolant flow, the efficiency increases, becomes maximum in the flow region corresponding to Re = 5000 for a pipe with a rod and Re = 8000 for an empty pipe, and then decreases. The shift in the Reynolds number between the maximum values of the efficiency coefficient E in the region of turbulent flow of the coolant is most likely determined by differences in the flow geometry. It can be assumed that an increase in the intensity of the heat transfer in the ring gap takes place until the scale of the transverse modes approaches the gap size. The character of the variation in the magnitude of the effective coefficient of heat transfer demonstrates a sharp decrease in the growth exponent upon surpassing the values Re > 5000 for a pipe with a rod. On the other hand, in the case of an empty pipe, the growth of the coefficient a_e with Re determines the small variation of the coefficient E of the heat exchanger when Re > 10,000. Inasmuch as the experimental data for the empty pipe were obtained in the range of Re > 8000, the behavior of the quantity E at smaller values of Re requires further investigation.

The growth of the efficiency of the heat exchanger with increasing channel length is a predictable result, but the character of the variation of the quantity E for different velocities of the coolant in the channels is nontrivial (Fig. 4). The efficiency of the heat exchanger at a velocity v = 30 m/sec is lower than at v = 40 m/sec. This is caused by the behavior of the effective thermal diffusivity, whose value is smaller at a velocity of 30 m/sec (Re = 3700) (Fig. 3).

As is evident from Fig. 4, the coefficient E is almost insensitive to the flow rate of the coolant for both short (z = 0.5 m) and long (z = 1.5 m) channels. The strongest dependence of the quantity E on the flow rate of the coolant is observed for channels with a length ranging from 0.5 to 1.5 m.

It is evident that introduction of a rod into a channel reduces the flow-passage cross section and enhances the pressure drop over the channel length accordingly. The measured pressure drop over the coolant path in the radiator $\Delta p = P_{in} - P_{out}$ (Fig. 5) shows that the hydraulic resistance of the radiator increases with the Reynolds number to a greater extent than the observed coefficient of thermal efficiency E.

In the case of Poiseuille flow in a pipe with a cirular cross section, the pressure drop is determined by the expression [3]



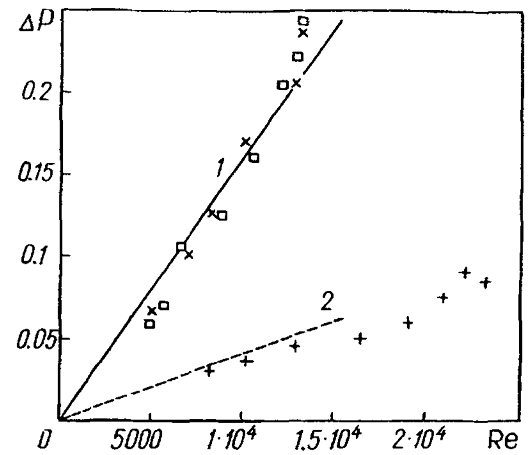


Fig. 4. Dependence of E on the length of the heat exchanger at various mass-averaged velocities of the coolant: 1) $v = 40$ m/sec; 2) 90; 3) 30.

Fig. 5. Dependence of the pressure drop along the coolant path on the Reynolds number: 1, 2) calculation for channels of $d_e = 3.2$ and 8 mm, respectively. The experimental data are the same as in Fig. 1. Δp, Pa·10^5.

$$\Delta p = v\,(R^2 - R_1^2)\,8\nu z \left[R^4 - R_1^4 - \frac{(R^2 - R_1^2)^2}{\ln (R/R_1)}\right]^{-1}, \tag{10}$$

where ν is the kinematic viscosity.

In the region of turbulent flow (Re > 2500) we retain Eq. (10) and introduce the effective viscosity ν_e. We determine its value from the condition of matching of calculated values of Δp to experimental data at Re = 5000. In this case $\nu_e = 4.25\nu$. It should be noted that one and the same value of the effective viscosity $\nu_e = 4.25\nu$ makes it possible to describe the change in the pressure in pipes with different effective diameters at various Reynolds numbers.

Conclusions. Experimental investigations have shown that the thermal efficiency E of the heat exchanger at constant internal and external heat-exchange surfaces can be increased by installing axial rods or band swirlers in the channels. Maximum values of the coefficient E were observed at Reynolds numbers of 5000 to 6000. We proposed a mathematical model for describing heat transfer in the channels of a heat exchanger that employs the concept of a Reynolds number-dependent effective thermal diffusivity a_e. The dependences $a_e = f(\mathrm{Re})$ obtained on the basis of a spline interpolation of experimental data are quite universal with respect to the channel geometry when Re < 6000. Calculations have shown that the thermal efficiency increases monotonically with the length of the channels. When the parameters of the construction satisfy the condition $2Raz/((R^2 - R_1^2)\pi v(R - R_1)) < 0.5$, the coefficient E depends linearly on the length of the channels.

The effective viscosity determined from experimental values of the pressure drop is a parameter on the basis of which one can calculate with sufficient accuracy for engineering purposes pressure drops in heat exchangers irrespective of the effective channel diameter and the flow regime of the coolant.

NOTATION

T_{in}, coolant temperature at the entrance to the heat exchanger, °C; T_{out}, coolant temperature at the exit from the heat exchanger, °C; T_a, temperature of the cooling air, °C; r, current radius of an internal channel of the heat exchanger, m; R, radius of an internal channel, m; R_1, radius of the rod set up in an internal channel, m; z, channel length, m; v, mass-averaged velocity of the coolant, m/sec; d_e, effective diameter of a channel, cm; Re = vd_e/ν; ν, kinematic viscosity, m^2/sec; ν_e, effective kinematic viscosity; a, molecular thermal diffusivity, m^2/sec; a_e, effective thermal diffusivity, m^2/sec. Indices: in, parameters at the entrance to the heat exchanger; out, parameters at the exit from the heat exchanger; e, effective parameters.

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