HEAT TRANSFER IN FLOW OVER STRAIGHT TRANSVERSE FINS

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An experimental investigation has been made of heat transfer in a large-scale model of a channel with fins transverse to a gas stream. Distributions of local heat transfer coefficient over fin height have been obtained as a function of the geometrical characteristics and the Re number, and from these data correction coefficients have been computed for the effectiveness of the fins, taking account of the nonuniformity of the distribution of α over fin height.

The widespread use of finned surfaces in modern heat exchangers makes increased demands on their computation. It is known that heat transfer coefficients may vary considerably from the base to the top of a fin. However, in view of the absence of systematic experimental data, it is assumed in computation that the heat transfer coefficients are constant over the entire surface of the fin, which may lead to substantial errors in certain conditions. Our knowledge of local heat transfer on straight transverse fins in longitudinal flow over a finned surface is very meager. Stynes and Myers [1] studied the distribution of local mass transfer coefficients for solid benzoic acid dissolved by water from the surface of a transverse fin, at Re = = 5800 and 12 000 and s/h = 0.31–2.3. Harris and Wilson [2] investigated the distribution of local heat transfer coefficient on a large-scale model at Re = $= 10^5$–$6 \cdot 10^5$ and s/h = 0.3. Thus, the available data either span a very narrow range of Re, or hold only for a single value of s/h. Therefore study of the heat transfer coefficient distribution with fin height over a wide range of the basic parameters, and computation of the efficiency on the basis of the data obtained, allowing for nonuniformity in distribution of α, are very timely.

Experimental equipment and method. The heat transfer investigation was conducted on a large-scale model of a channel with transverse fins in the range of variation of Re from $3 \cdot 10^3$ to $3.5 \cdot 10^5$. The experimental section (Fig. 1) was a rectangular plexiglas channel, 400 × 100 mm in cross section and 1020 mm long. The base of the channel had 64 slots of width 3.2, depth 8, and pitch 15 mm, in which were mounted micarta fins 3 mm thick, measuring 62 × 400 mm. Variation of the geometrical parameters was accomplished by varying the distance between fins.

Tests were performed with s/h = 0.222, 0.5, 1.06, 2.17, (h = 54 mm), and on a single fin (s = ∞).

Heat transfer was investigated by the method of local thermal modeling on a single fin, the other fins serving to model the hydrodynamic arrangement. The test fin was a micarta plate, on both sides of which were cemented constantan ribbons 0.2 mm thick with copper leads. After hot bonding under pressure, the ribbon was divided into individual strips in such a way as to form 5 strips 10 mm wide on each side of the fin. The space between strips was 1 mm, and was filled in with epoxy resin. On the lower part of the fin, that inserted into the slot, there was one ribbon strip 7 mm wide on each side, to compensate for loss by conduction to the base.

The surface temperature was measured by 24 copper-constantan thermocouples, which were laid in special slots of width 1 mm, and depth 0.7 mm previously milled in the micarta plate. After the thermocouples were put in, the slots were filled with epoxy resin in such a way that the heads of the thermocouples were flush with the surface of the backing piece. The air temperature was measured by the same thermocouples at the inlet to the experimental section, the thermocouple emf being measured with a PPTN-1 potentiometer. Self-heating of the strips was accomplished by passing alternating current through them supplied from the mains through a stabilizer and a voltage transformer. The heater power was controlled by an autotransformer on the high-voltage side. The current was measured with an astatic ammeter of accuracy class 0.5% and a current transformer. The resistance of each strip was measured beforehand by a potentiometric method.

To reduce the heat loss by radiation, the two fins adjoining the test fin were covered over with aluminum foil.

The air came to the experimental section from a high-head fan, first passing through a straightening grid and a stilling section. Mass flow rate was measured by a curtate Venturi tube, and at the maximum flow rates (Re ≥ $1.5 \cdot 10^5$)—with a Pitot tube.

In the tests the temperature* was maintained constant over the whole fin by controlling the heat power generated in the strips.

The heat flux density was determined from the equation

$$q_i = (Q_i - Q_r)/F/n, \tag{1}$$

where F/n is the heat transfer area ot a strip; Q_i = $= I^2R$ is the electrical power generated in the strip; Q_r is the heat flux due to radiation.

In calculating Q_r, the emissivity ε of the constantan ribbon surface was assumed to be 0.26. The magnitude of Q_r did not usually exceed 10% of the power generated.

The local heat transfer coefficients were calculated from the equation

*It was confirmed experimentally in [2] that the distribution of heat transfer coefficient was practically independent of the temperature distribution on the fin.

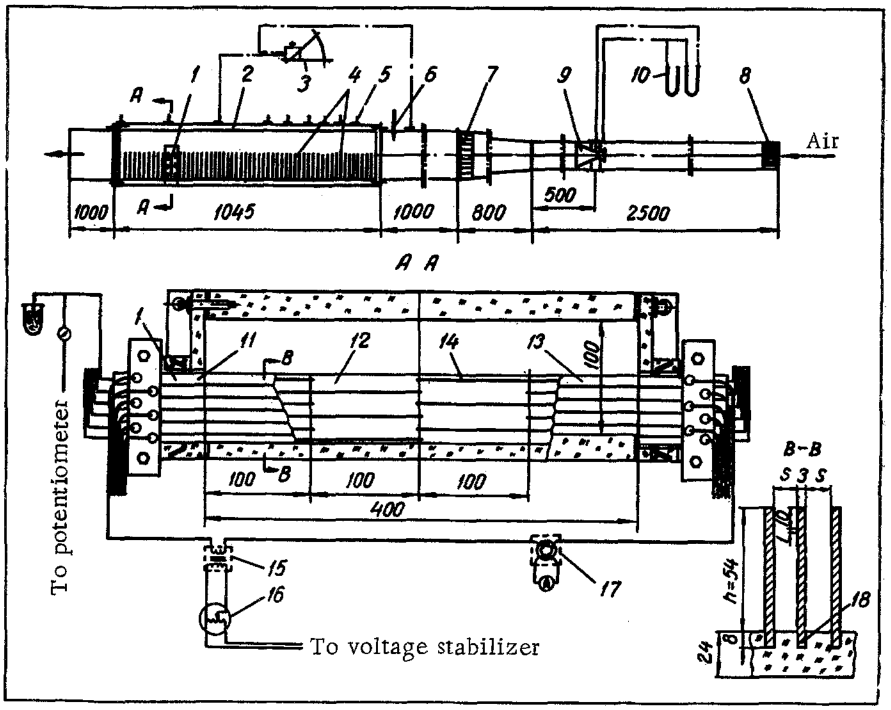

Fig. 1. Schematic of experimental apparatus: 1) test fin–calorimeter; 2) experimental section; 3) micromanometer; 4) modeling fins; 5) static pressure taps; 6) thermocouple; 7), 8) straightening grids; 9) Venturi tube; 10) U-tube manometer; 11) copper leads to plate; 12) micarta backing piece; 13) constantan plates; 14) channel for thermocouples along center of plate; 15) stepdown transformer; 16) voltage regulator; 17) current transformer; 18) compensating heater.

$$\alpha = q_i/(t_s - t_a). \tag{2}$$

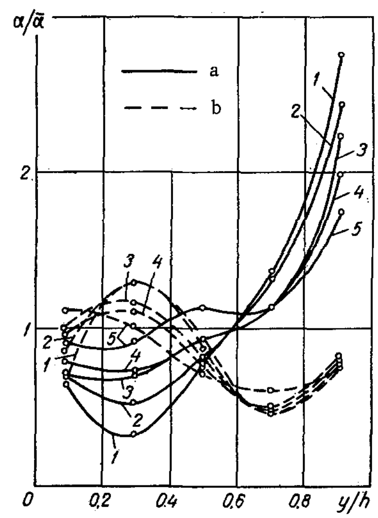

Fig. 2. Distribution of heat transfer coefficient over fin height for s/h = = 1.06 and Re = $3.53 \cdot 10^3$ (1), $19.2 \cdot 10^3$ (2), $51.1 \cdot 10^3$ (3), $95.2 \cdot 10^3$ (4), $336 \cdot 10^3$ (5) for the front (a) and rear (b) sides of the fin.

The mean heat transfer coefficient and the corresponding Stanton number were calculated from the equations

$$\bar{\alpha} = \frac{1}{10}\left(\sum_{i=1}^{10} q_i\right) \Big/ (t_s - t_a), \tag{3}$$

$$\overline{\mathrm{St}} = \bar{\alpha}/W\gamma c_p. \tag{4}$$

Results of the investigation and their generalization. Figure 2 shows a typical distribution of heat transfer coefficients α over the fin height. On the forward side of the fin α drops sharply from the top to the bottom.* On the rearward side the heat transfer coefficient distribution is more complex—there are minima close to the top and to the root of the fin, and maxima directly at the top and in the middle part of the fin. A distribution of this kind is due to the nature of motion of the fluid in the gap between fins (to the presence of two coupled vortices [2, 3]). The reduced heat transfer coefficient distributions obtained are in satisfactory agreement with the data of references [1, 2].

The experimental data on mean heat transfer coefficients, in the range $10^4 < \mathrm{Re} < 3.5 \cdot 10^5$, may be approximated, with an accuracy of ±10%, by the equation

$$\overline{\mathrm{St}} = 0.078\ \mathrm{Re}^{-0.25}(s/h)^{0.32}. \tag{5}$$

The exponent for Re is a mean for the whole range of s/h (it varies from −0.2 at s/h = 0.222 to −0.28 at s/h = 2.17).

*This effect increases with decrease of s/h.

For the value s = ∞ (solitary fin) we obtained

$$\overline{\mathrm{St}} = 0.272\ \mathrm{Re}^{-0.37}. \tag{5a}$$

Comparison of the test data on $\overline{\mathrm{St}}$ number of the present investigation with results of other papers on mean heat transfer on finned tubes [3–7] shows satisfactory agreement.

Efficiency of a straight fin when the heat transfer coefficient varies. The stepwise distribution of local heat transfer coefficients over the height of a fin obtained in the tests allows calculation of the actual efficiency of finning in the range of the basic parameters (s/h, Re) examined.

To determine the efficiency we make the conventional assumptions: 1) the heat transfer process is stationary; 2) the thermal conductivity of the fin is constant; 3) the fin thickness is small in comparison with its height, so that temperature gradients in a direction perpendicular to the lateral surface are negligibly small, and heat exchange with the ends may be neglected; 4) there are no internal heat sources in the fin.

With the above assumptions the differential equation of heat transfer for a straight fin of constant thickness has the form

$$\delta\lambda \frac{d^2\Theta_i}{dy^2} = 2\alpha_i\Theta_i, \tag{6}$$

where

$$\alpha_i = (\alpha_i' + \alpha_i'')/2;\ \Theta_i = t_i - t_a;$$

i = 1, 2, ..., n; t_i is the temperature of the fin; α_i', α_i'' are respectively the heat transfer coefficients on the front and rear sides of the fin on the i-th strip. In our tests n = 5.

The boundary conditions are

when $y = 0$

$$\Theta = \Theta_0;$$

when $y = i\Delta$

$$\Theta_i = \Theta_{i+1},\ \frac{d\Theta_i}{dy} = \frac{d\Theta_{i+1}}{dy};$$

when $y = n\Delta = h$

$$\frac{d\Theta_n}{dy} = 0. \tag{7}$$

We seek a general solution of the system (6) in the form

$$\Theta_i = C_i \operatorname{sh}\beta_i[y - (i-1)\Delta] + C_i' \operatorname{ch}\beta_i[y - (i-1)\Delta],$$

$$i = 1, 2, \ldots, n;\ \beta_i = \sqrt{\frac{2\alpha_i}{\delta\lambda}}. \tag{8}$$

Using (7), we obtain a system of equations for determining C_i and C_i':

$$C_1' = \Theta_0,$$

$$C_i \operatorname{sh}\beta_i\Delta + C_i' \operatorname{ch}\beta_i\Delta = C_{i+1}',$$

$$C_i \operatorname{ch}\beta_i\Delta + C_i' \operatorname{sh}\beta_i\Delta = C_{i+1}\frac{\beta_{i+1}}{\beta_i},\ i = 1, 2, \ldots, (n-1),$$

$$C_n \operatorname{ch}\beta_n\Delta + C_n' \operatorname{sh}\beta_n\Delta = 0. \tag{9}$$

From (9) we obtain recurrence relations for determining all the coefficients:

$$C_n = -C_n' \varphi_n, \quad \varphi_n = \operatorname{th} \beta_n \Delta, \quad \psi_n = 1,$$

$$\cdots\cdots\cdots\cdots\cdots\cdots$$

$$C_i = -C_n' \varphi_i, \varphi_i = \operatorname{ch} \beta_i \Delta \times$$
$$\times \left(\frac{\beta_{i+1}}{\beta_i} \varphi_{i+1} + \psi_{i+1} \operatorname{th} \beta_i \Delta \right),$$
$$C_i' = C_n' \psi_i, \quad \psi_i = \operatorname{ch} \beta_i \Delta \times$$
$$\times \left(\frac{\beta_{i+1}}{\beta_i} \varphi_{i+1} \operatorname{th} \beta_i \Delta + \psi_{i+1} \right). \quad (10)$$

The expression for the efficiency

$$\eta = q/q_{\lambda=\infty} \quad (11)$$

may be put in the form of a product of the efficiency when

$$\alpha = \bar{\alpha} = \frac{1}{h} \int_0^h \alpha(y)\, dy$$

and a correction coefficient for variation of the heat transfer over fin height, i.e.,

$$\eta = \eta_c \zeta, \quad (12)$$

where $\zeta = q/q_{\bar{\alpha}}$

Using (6), we have

$$\eta = \frac{q}{q_{\lambda=\infty}} = -\frac{\delta\lambda (d\Theta/dy)_{y=0}}{2\bar{\alpha} h \Theta_0} = -\frac{C_1 \delta\lambda\beta_1}{2\bar{\alpha} h \Theta_0} =$$
$$= -\frac{C_1 \sqrt{\alpha_1/\bar{\alpha}}}{\bar{\beta} h \Theta_0}, \quad (13)$$

where $\bar{\beta} = \sqrt{2\bar{\alpha}/\delta\lambda}$.

From (10) we find

$$C_1 = -\frac{\varphi_1}{\psi_1} \Theta_0.$$

Using (13) and the well-known expression for $\eta_c = \operatorname{th} \bar{\beta} h/\bar{\beta} h$, we finally obtain

$$\zeta = \frac{\varphi_1}{\psi_1} \frac{\sqrt{\alpha^1/\bar{\alpha}}}{\operatorname{th} \bar{\beta} h}. \quad (14)$$

Figure 3 shows values of ζ calculated from the above equations for all the systems investigated. These graphs permit us in engineering calculations, to take account directly of the distribution of heat transfer coefficient over the height. For example, from the known characteristics of finning, s/h, and the parameters of the heat transfer agent, Re, we find the $\overline{St}$ number from (5), and then, using Fig. 3, we obtain

$$St_r = \overline{St} \eta_c \zeta, \quad (15)$$

from which we may determine the heat flux or the temperature at the base of the fin.

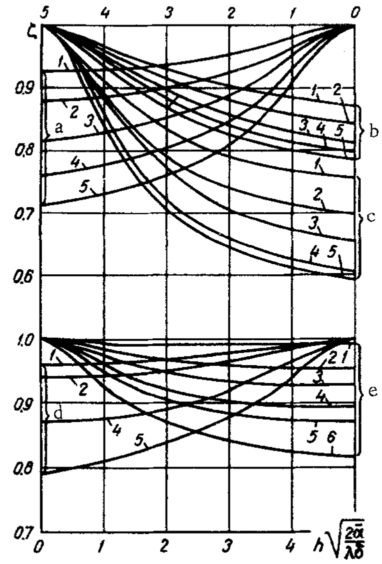

Fig. 3. Dependence of correction coefficient ζ on the quantity $h(2\bar{\alpha}/\delta\lambda)^{1/2}$ for Re = $3 \cdot 10^5$ (1), 10^5 (2), $5 \cdot 10^4$ (3), $2 \cdot 10^4$ (4), 10^4 (5), $3 \cdot 10^3$ (6) and s/h = 0.5 (a), ∞ (b), 0.222 (c), 2.17, (d), 1.05 (e). Values of the abscissa $h\sqrt{2\bar{\alpha}/\delta\lambda}$ for (b), (c), and (e) are on the lower scale, and for (a) and (d)—on the upper scale.

NOTATION

h—fin height; y, s—distances from base and between fins; δ—fin thickness; H and b—height and width of channel; $f = b(H - h)$—free flow section; $d_e = 2b(H - h)/(b + H - h)$—equivalent diameter with respect to free flow section; W—velocity of heat transfer agent (determined from cross section f); t_0—temperature at base of fin; t_s and t_a—temperature of surface and air; q—heat flux through base of fin; $q_{\lambda=\infty}$—heat flux through base under conditions of infinite thermal conductivity of fin material; $\eta = q/q_{\lambda=\infty}$—efficiency; α, $\bar{\alpha} = \frac{1}{h}\int_0^h \alpha\, dy$, $\alpha_r = q/(t_0 - t_a)$ —local, mean, and reduced heat transfer coefficients; λ, ν, c_p—respectively, thermal conductivity, kinematic viscosity, and specific heat of air; $St = \alpha/W\gamma c_p$, $St_r = \alpha_r/W\gamma c_p$—respectively, mean and reduced Stanton numbers; $Re = Wd_e/\nu$—Reynolds number.

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