HEAT TRANSFER AND HYDRAULIC RESISTANCE IN A SHORT PLANE-PARALLEL DUCT WITH ARTIFICIALLY ROUGHENED WALLS

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Experimental heat transfer and hydraulic resistance data are correlated by means of dimensionless relationships.

The results of investigation of the effect of artificial roughness on heat transfer and resistance in the conditions of the internal problem have been fairly thoroughly dealt with in the literature. There are no such data at present, however, for a short plane-parallel duct, and the results obtained for other ducts cannot be used owing to the difference in cross-sectional configuration, as was noted in [1].

This paper gives experimental heat transfer and resistance data for a duct of length l = 420 mm and height 2s = 10, 20, and 30 mm in which boundary-layer turbulence was artificially created by baffles on the heating surface. The baffles were semicylindrical projections of height h attached to the surface, transverse to the flow, at equal intervals t (Table 1). The baffles were made of ASTT plastic.

Heat transfer in the duct was produced by dc electric heating of the duct walls and cooling of them by an air flow. The air speed and temperature in the duct were measured with a thermoanemometer. The pressure drop along the duct was measured with Pitot tubes at the inlet and outlet of the duct. The heat transfer coefficient was determined by the steady heat flow method, and the dimensionless relationship for $\bar{\alpha}$ had the form

$$\overline{\mathrm{Nu}}_f = f(\mathrm{Re}_f,\ K_m,\ K_s),$$

where K_m = h/t and K_s = h/s are geometric roughness numbers [2].

The selected characteristic dimension in the similarity criteria was the duct length l, and the characteristic temperature was the arithmetic mean of the sum of the local flow and wall temperatures along the duct.

TABLE 1

Surface*	Roughness numbers		c_{exp}	$c_m c_s$	$\frac{c_{exp}}{c_m c_s}$
	K_m	K_s			
30/4,3 (t_1)	0,08	0,29	0,049	0,047	1,0425
30/4,3 (t_2)	0,054	0,29	0,052	0,056	0,93
30/3,8 (t_1)	0,07	0,25	0,052	0,0526	0,99
30/3,8 (t_2)	0,047	0,25	0,057	0,0566	1,0
30/1,0 (t_1)	0,019	0,067	0,048	0,0482	1,0
30/1,8 (t_1)	0,033	0,12	0,055	0,0548	1,0
20/4,3 (t_1)	0,08	0,45	0,044	0,0449	0,982
20/4,3 (t_1)	0,07	0,38	0,048	0,0509	0,943
10/3,8 (t_1)	0,07	0,76	0,043	0,0435	0,989

*Symbols denote ducts of height 10, 20, and 30 mm with baffles h = 4.3-1.0 mm at intervals t_1 = 54 mm, t_2 = 81 mm.

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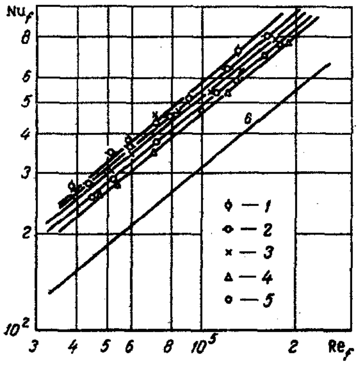

Fig. 1

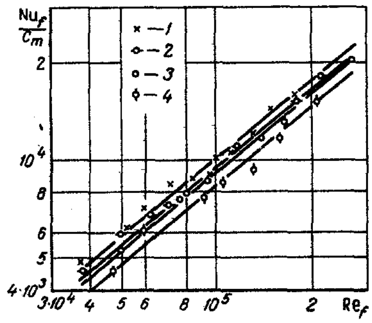

Fig. 2

Fig. 1. Correlation $\overline{Nu}_f = f(Re_f)$ for ducts with baffles on heating surface with K_s = const: 1) $K_m = 0.047$; 2) 0.033; 3) 0.07; 4) 0.018; 5) 0.08; 6) $K_m = 0$ (smooth-walled duct).

Fig. 2. Correlation $\overline{Nu}_f/C_m = f(Re_f)$ for different K_s: 1) $K_s = 0.25$; 2) 0.38; 3) 0.45; 4) 0.76.

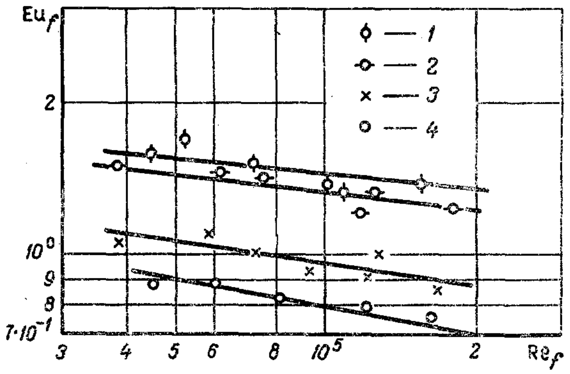

Fig. 3. Resistance in duct with 2s = 30 mm and baffles on the heating surface, K_s = const: 1) $K_m = 0.08$; 2) 0.054; 3) 0.047; 4) 0.033.

The effect of K_m on the heat transfer rate in the channel is shown in Fig. 1. As this graph shows, correlation of the results $\overline{Nu}_f = f(Re_f)$ in a logarithmic anamorphosis with $Re_f = 3 \cdot 10^4 - 3 \cdot 10^5$ gives straight lines with a gradient determined by the exponent n = 0.8 of the Reynolds number. Thus, the stratification of the experimental data for K_m and K_s is determined entirely by the proportionality factor

$$C_0 = f[C_m(K_m),\ C_s(K_s)],$$

which for K_s = const is given fairly accurately by the relationship

$$C_m = -1.04K_m - 10.4K_m^2.$$

The proportionality factor $C_s = f(K_s)$ is determined from the correlation of the results $\overline{Nu}_f/C_m = f(Re_f)$ in Fig. 2, which shows that the experimental results are interpolated by straight lines parallel to one another. The exponent of the Reynolds number here is 0.8, and the proportionality factor $C_s = f(K_s)$ is given by the expression

$$C_s = 1.029 - 0.07K_s - 0.26K_s^2.$$

Thus, the dimensionless equation for heat transfer in the investigated duct with baffles on the heating surface with $0.78 > K_s > 0.25$ and $0.08 > K_m > 0.019$ has the form

$$\overline{Nu}_f = (0.03 + C_m C_s)\, Re_f^{0.8}. \tag{1}$$

When h = 0 the results of this equation agree with the experimental data for a plane-parallel duct with no roughness on the heating surface.

The hydraulic resistance, in the form of the dimensionless number Eu_f, for the investigated duct is shown as a function of Re_f in Fig. 3. This figure shows that the distribution of the experimental data depends on the geometric roughness numbers. When K_s = const, K_m determines not only the increase in resistance in the duct, but also the change in the relationship $Eu_f = f(Re_f)$: with increase in K_m the slope of straight lines in Fig. 3 decreases. The value of K_s affected only the magnitude of the resistance in the duct.

Hence, the expression for the resistance in the investigated duct is represented by the relationship:

$$Eu_f = \left(C_0 \frac{l}{s} + C_1\right) Re_f^{-n_0 + m'},$$

where $Eu_f = C_0(l/s)Re_f^{-n_0}$ is the relationship for a duct with no roughness on the heating surface; $C_1 = C'_m C'_s$ is a proportionality factor which determines the effect of K_m and K_s on Eu_f.

Proceeding in the same way as in the determination of Eq. (1) we obtain for C'_m and C'_s

$$C'_m = 1.22K_m^{-0.4},$$
$$C'_s = 0.37 + 2.61K_s - 1.68K_s^2.$$

The final equation for the resistance in a plane-parallel duct with baffles on the walls has the form

$$Eu_f = \left[0.067\left(\frac{l}{s}\right)^{0.9} + C'_m C'_s\right] Re_f^{-0.21+1.89K_m^{1.13}}. \quad (2)$$

When the relationships given here are applied to baffles with a different configuration from those investigated the resistance of the baffles must be taken into account, since in the general case it affects the efficiency of the duct heating surface.

The relative error of the presented results was 4.6% in Eq. (1) and 5.2% in (2).

LITERATURE CITED

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2. V. G. Pavlovskii, Proc. of Scientific Conference of Young Scientists of the Lithuanian SSR [in Russian], Tekhnika, AN LitSSR, Vil'nyus (1967).