HEAT TRANSFER AND DRAG OF A TURBULENT SWIRLING AIR STREAM IN THE ENTRANCE SECTION OF AN ANNULAR CHANNEL

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Results are presented of an experimental investigation of the average heat transfer and hydraulic resistance of air in turbulent flow in the entrance section of an annular tube with various angles of swirl.

In the design of heat exchangers (very common in contemporary energy conversion equipment) the problem of increasing the thermal loading of the heat transfer surface and correspondingly reducing the size of the heat exchanger is often encountered. One of the most promising methods for improving the technology of heat exchangers is the use of swirling flow as a means of intensifying heat transfer under internal flow conditions. The majority of published papers referring to this question relate to analysis of conditions for intensifying heat transfer using spiral strips with constant pitch along the channel length [1-3].

In some cases it is expedient to locate the swirl device at the channel entrance section. The literature has a small number of papers dealing with this topic [4, 5]. The diverse conditions under which the tests were conducted (construction and geometry of the swirl devices, the range of variation of initial flow swirl angle, etc.) do not allow extension of the results to other cases. For this reason an experimental investigation of the drag and convective heating in a swirling air flow at the entrance section of an annular tube was conducted at the Lenin Neva Works. The results of the first stage of the work, an investigation of local heat transfer, were published in [6]. The present tests were conducted on the same equipment, details of the construction of which were given in [6], together with the calibration, measurement techniques, and data reduction. A schematic of the test equipment is given in Fig. 1.

Air from an axial compressor reaches an annular channel, of a technical grade of smoothness, formed by the calorimeter tube 1, of internal diameter d = 200 mm, and a tube 2 with external diameter d = 130 mm. The length of the annular channel is L = 1850 mm, which is 26 calibers of the equivalent diameter of the annular channel, d_e = 70 mm, In the duct 3 ahead of the measuring section there are annular blade-type swirl generators 4 with solid cylindrical blades and a fairing 5. The tests were conducted with initial flow swirl angles of $\varphi_0 = 0°$; 50°30'; 63°30'; 77°50'. The tube 1 was heated at constant current by means of the Nichrome wire 6, mounted on the external surface of the tube. To reduce thermal losses the tube was insulated using asbestos cloth 7. To avoid longitudinal heat flow the surface temperature t_w of the calorimeter tube 1 was equalized along its length, i.e., the heat transfer was investigated under the condition t_w = const.

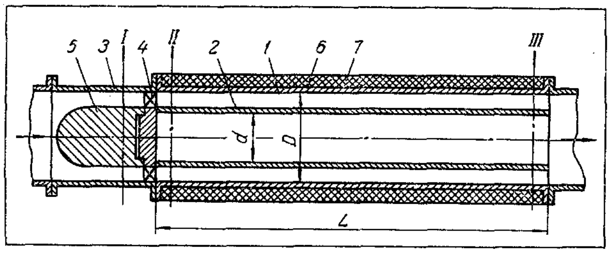

Fig. 1. Schematic of experimental equipment.

During the tests measurements were made of the air flow rate, the temperature and pressure at the entrance to the swirl device (section I), at the entrance and exit of the measurement section (sections II and III), of the surface temperature of the outer and inner tubes, and of the parameters of the heating

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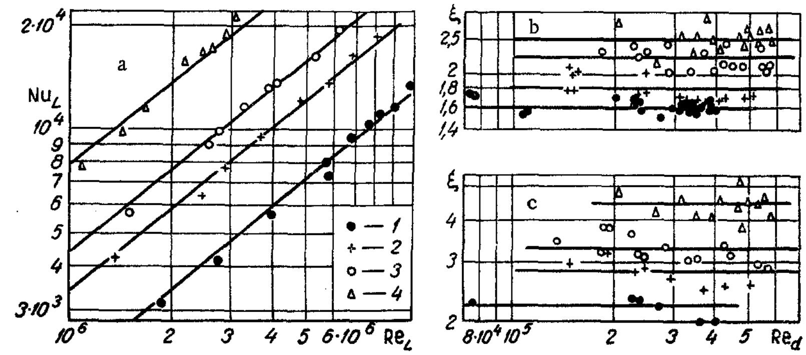

Fig. 2. Correlations obtained: a) $Nu_L = f(Re_L)$; b) $\xi = f(Re_d)$; c) $\xi_c = f(Re_d)$ for $\varphi_0 = 0°$ (1); 50° (2); 63° (3); 78° (4).

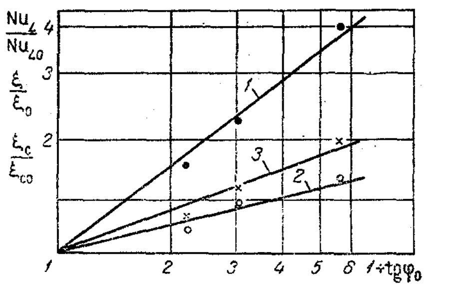

Fig. 3. Effect of initial swirl angle on mean heat transfer (1), hydraulic resistance of the test section (2), and total hydraulic resistance (3), for constant fluid flow rate.

current. The heat losses through the insulation were determined by prior calibration as a function of the temperature drop through the insulation, and amounted to 10-12% of the heat given by the tube to the air stream via convection. The imbalance between the heat generated by the electrical heater, allowing for losses, and the heat received by the air did not exceed 2-5%, for the majority of the tests.

The average heat transfer coefficient was found from the expression

$$\alpha = \frac{Q_0 - Q_l - Q_r}{f(t_w - t)}.$$

Following [7], we sought a correlation for the average heat transfer in the form $Nu_L = f(Re_L)$. Here the physical constants and the characteristic velocity were determined from the mean air parameters in the measurement section. The results of processing the test data in this way are shown in Fig. 2a. The experimental points separate out as a function of the initial flow swirl angle, φ_0, and for φ_0 = const they are described well by the relation:

$$Nu_L = c\,Re_L^{0.8}.$$

The quantity c increases with increase of the swirl angle, and for $\varphi_0 = 78°$ the average heat transfer, for the same air flow rate, is greater by a factor of 4 than for straight-line motion of the air.

The test data on drag of the measurement section for each swirl angle were processed to fit a relationship $\xi = f(Re_d)$. As follows from Fig. 2b, the drag coefficient in the range tested does not depend on the Re number, and increases sharply with increase of swirl angle. For nonswirling flow the experimental data are in good agreement with the results of [8] for turbulent fluid motion in the entrance section of a tube with a "sharp edge" at the entrance.

The drag coefficient ξ does not take account of pressure drop over the swirl generator. Nevertheless, the loss of head in the swirl generator appreciably increases the total drag of the measurement section including the swirl generator. The test data for this case are shown in Fig. 2c. The drag coefficient ξ_c, giving the total head loss, was calculated from the total pressure drop between sections I and III and the dynamic head at the entrance to the measurement section.

The heat transfer and drag in swirling flow depends both on the axial component, and on the rotational component of velocity. Therefore, following [6], we choose the characteristic parameter to be the ratio of these velocities at the entrance section of the measurement section (at the exit from the swirl generator), $W_{\varphi_0}/W_{x_0} = \tan\varphi_0$. The effect of swirl can be explained by comparing the dimensionless coefficients Nu, ξ, and ξ_c with their corresponding values for axial flow. Figure 3 shows that these ratios depend on a function of the form

$$(1 + \mathrm{tg}\,\varphi_0)^n. \qquad (1)$$

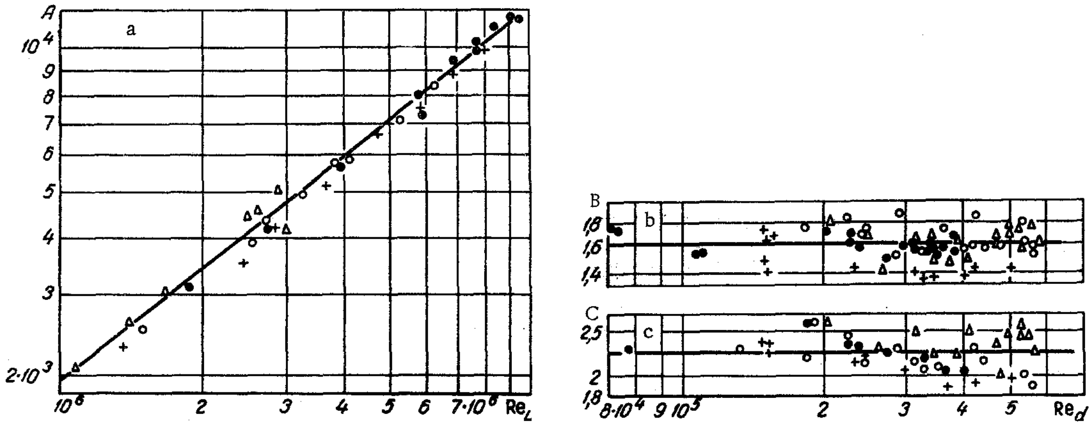

Fig. 4. Correlation of test data on heat transfer (a) and drag (b and c) in swirling flow in the entrance section of an annular channel (for notation, see Fig. 2): $A = Nu_L/(1 + \tan \varphi_0)^{0.77}$; $B = \xi/(1 + \tan \varphi_0)^{0.25}$; $C = \xi_c/(1 + \tan \varphi_0)^{0.35}$.

A parametric equation which is a satisfactory correlation of the test data on average heat transfer in swirling flow at the entrance section of the annular channel ($L \le 25\, d_e$), with heating of only the outer tube, has the form

$$Nu_L = 0.0319\,(1 + \mathrm{tg}\,\varphi_0)^{0.77}\,Re_L^{0.8}. \qquad (2)$$

The maximum scatter of the experimental data does not exceed 12% (Fig. 4a).

The correlation obtained permits possible limiting transition; for $\varphi_0 = 0$ (axial flow) Eq. (2) gives the heat transfer of a plate washed by an infinite turbulent stream without pressure gradient [9].

The drag of the initial section of the annular tube with swirling flow can be determined from the formula

$$\xi = \xi_0\,(1 + \mathrm{tg}\,\varphi_0)^{0.25}; \; \xi_0 = 0.0162. \qquad (3)$$

For calculating the pressure drop in the swirl generator, the total head loss over the entire length of the measurement section ($L = 26 d_e$) can be described by

$$\xi_c = \xi_{c_0}\,(1 + \mathrm{tg}\,\varphi_0)^{0.35}; \; \xi_{c_0} = 0.0223. \qquad (4)$$

The maximum scatter of the test data on drag amounts to 15% (Fig. 4b and c).

NOTATION

D, d, d_e, L, f, and F	are the outer, inner, and equivalent diameters, length, cross-sectional area, and heat transfer area for the annular channel;
L_1	is the distance between sections I and III;
$T_w = t_w + 273$ and T_{in}	are the absolute temperature of outer and inner surface of annular channel;
G	is the flow rate;
P, t, and W_x	are the average pressure, average temperature, and average flow rate of air in the measurement section;
φ_0, γ_0, W_{x_0}, W_{φ_0}, $W_0 = G/3600\,\gamma_0 \cos\varphi_0 f$, $P_p = W_0^2\gamma_0/2g$	are swirl angle, specific gravity, axial, tangential, and total velocity of flow, and dynamic head at exit from swirl generator;
Q_0, Q_l, $Q_r = 4,9\,\varepsilon F[(t_w/100)^4 - (t_{in}/100)^4]$	are the heat liberated by electric heater, heat loss through the outside insulation to the surrounding medium, heat loss by radiation;
$\varepsilon = 0.6$	is the reduced emissivity;
α, λ, and ν	are heat transfer coefficient, thermal conductivity, and kinematic viscosity;
$Nu_L = \alpha L/\lambda$, $Re_L = W_x L/\nu$, and $Re_d = W_0 d_e/\nu$	are similarity parameters;
Δp_c and Δp	are the drop in total pressure between sections I and III and between sections II and III;

$\xi_c = (\Delta p_c / P_r)(d_e/L)$ and $\xi = (\Delta p/P_r)(d_e/L_1)$ are the drag coefficient of measurement section, with and without pressure drop in the swirl generator accounted for.

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