HEAT TRANSFER FROM CYLINDER DURING ITS IMPACT STREAMLINING BY AIR JETS

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A study is made of heat transfer from finned and smooth cylinders during impact streamlining by air jets. Criterial relations are derived for calculating the heat transfer in these cases.

Streamlining of a heat exchanger surface by high-velocity jets flowing normally to the surface is attended by an appreciable intensification of the heat transfer. This phenomenon can be explained by an erosion of the laminar boundary layer under the impact of jets on the surface, with a resulting decrease of the thermal impedance. Another factor contributing to intensification of the heat transfer is the appreciable turbulization of the boundary layer caused by the pressure gradient around the cylinder periphery. This phenomenon is widely utilized in heat exchangers. The intensity of heat transfer depends on the geometrical parameters of the convection surface and of the jet forming orifices, and also on the distance from the surface to the screen. A convection surface is, as a rule, a bundle of tubes. Therefore, pinpointing the peculiarities of heat transfer from a cylinder during its impact streamlining by air jets will facilitate the optimal design of tubular heat exchanger surfaces.

The heat transfer from a cylinder in the vicinity of the frontal stagnation point and the heat transfer from a plate during their streamlining by jets have already been studied [1-3]. The plate was assumed to be in a position perpendicular to the stream. Combining a cylinder and a plate should produce an efficient convection surface, one more intensely transmitting thermal fluxes during its streamlining by jets. The mechanism of heat transfer under these conditions has not yet been explored adequately enough. This study will deal with the heat transfer from a finned cylinder, its fins oriented perpendicularly to the jets, whereupon comparisons will be made with the heat transfer from a smooth cylinder [2] and from a plate [1, 3]. During discharge of a free jet from an orifice there takes place ejection of the substance occupying the ambient space as this substance diffuses into the jet. The parameters of a turbulent jet can be evaluated on the basis of the statistical theory of turbulent diffusion [4]. For a characterization of the scattering of particles around a straight trajectory we introduce the concept of dispersion

$$\sigma_{\tau}^2 = \frac{V_1^2}{V_2^2}\varkappa = \varepsilon^2\varkappa^2$$

with

$$\varkappa < 2\varkappa_0 < \frac{D_{\tau}}{V_{\varepsilon}^2}.$$

The solution to the equation of diffusion at the boundary between a jet and a quiescent medium is the velocity

$$V_\delta = V_0\Pi\left(\frac{d_v}{\delta_v};\ \frac{r}{\sigma_v}\right). \tag{1}$$

The function Π can be represented in the form

$$\Pi = \frac{1}{\Upsilon^2}\int_0^a \exp\left(\frac{r^2 - r_0^2}{2\sigma_v}\right) I_0\left(\frac{r}{\sigma}\right) r_0 dr. \tag{2}$$

Inasmuch as it is difficult to evaluate the dispersion precisely, it will be expedient to calculate the jet characteristics semiempirically. The velocity of an oncoming jet can be found by the expression [2]

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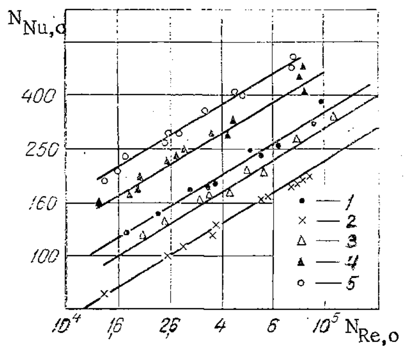

Fig. 1

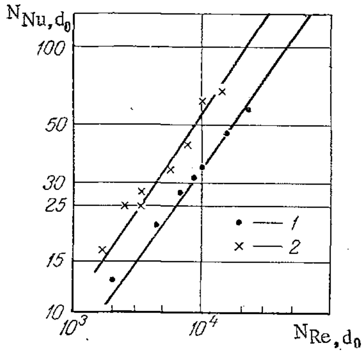

Fig. 2

Fig. 1. Dependence of $N_{Nu,o}$ on $N_{Re,o}$: 1) smooth cylinder, 2) $m_f = 2.24$, 3) $m_f = 1.68$; dependence of $N_{Nu,cr}$ on $N_{Re,cr}$: 4) $m_f = 1.68$, 5) $m_f = 2.24$.

Fig. 2. Dependence of N_{Nu,d_0} on N_{Re,d_0}: 1) $m_f = 1.68$, 2) $m_f = 2.24$.

$$V_\delta = 2V_0\left[\,1 + \left(1 + 0{,}278\left(\frac{\delta}{d_0}\right)\right)^{0.5}\right]^{-1}. \tag{3}$$

Accordingly, one can calculate the velocity of an oncoming jet either on the basis of expression (1), when function Π is known as defined by relation (2), or approximately on the basis of expression (3).

Experimental studies were made on the wind tunnel which had been described earlier [2]. The velocity of an oncoming jet was measured with a Prandtl tube. The readings were compared with theoretical values of the velocity according to expression (3). The calorimeter consisted of a cylinder with a heater inside, the latter comprising a nichrome coil wound around an asbestos–cement rod. In order to ensure a uniform temperature field on the cylinder surface, the gap between the heater and the cylinder was filled with fine-disperse quartz sand.

The emf of the thermocouples was measured with a Class 0.05 electronic potentiometer. Bundles of jets with circular cross section were formed by means of screens with holes. The spacing of these holes, namely the pitch a, was also varied during the experiment.

A preliminary estimate of heat losses yielded 7% of the calorimeter power. The heat transfer coefficient α for a cylinder was evaluated by the calorimetric method. The test data were processed by the standard method of statistical averaging and error calculation. The experimental error was 6%.

The criterial numbers N_{Nu} and N_{Re} were generally referred to the diameter of the holes in the screen and the jet velocity in the holes as characteristic quantities. For comparing the heat transfer from a finned cylinder and from a smooth cylinder we referred the heat transfer coefficients to the cylinder surface and to the total surface. As the characteristic dimension was in this case selected either the diameter d of the cylinder or the width of the cylinder with fins.

The dependence of the dimensionless heat transfer coefficient N_{Nu} on the Reynolds number N_{Re} is shown in Fig. 1 in a logarithmic anamorphosis for smooth and finned cylinders, the latter with a finning index $m_f = 1.68$ and 2.24, respectively.

An analysis of the experimental data has revealed that the magnitude of the heat transfer coefficients decreases as the finning index increases, if these coefficients are referred to the total surface. When they are referred to the tubular surface alone, however, then a noticeable intensification of the heat transfer seems to occur (Fig. 1).

For the purpose of determining the effect of a jet on the heat transfer from finned and smooth cylinders, tests were performed with streamlining of these cylinders under conditions of potential flow and by jets respectively (N_{Re} = idem, $V_\delta = 7$ m/sec).

It has been demonstrated [1] on the basis of Buckingham's Π-theorem that the heat transfer from a plate during its streamlining by jets normal to the surface depends on the ratios δ/d_0 and a/d_0 as well as on the Reynolds number N_{Re,d_0}.

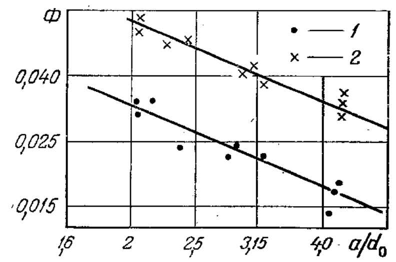

Fig. 3. Dependence of $\overset{\theta}{\varphi}$ on a/d_0: 1) m_f = 1.68, 2) m_f = 2.24.

For the purpose of determining the role of the Reynolds number N_{Re,d_0}, tests were performed with invariable geometrical parameters and with a variable velocity W_0. The test data were then processed in a logarithmic anamorphosis (Fig. 2). An analysis of the results has revealed that the Nusselt number N_{Nu,d_0} is proportional to N_{Re,d_0} 0.74.

Tests have also confirmed the results of that other study [1] pertaining to the dependence of the heat transfer on the ratio δ/d_0. An analysis of processed experimental data indicates that the Nusselt number N_{Nu,d_0} is proportional to $(\delta/d_0)^{0.09}$.

Experimental data on the heat transfer from a finned cylinder with the spacing of holes in the screen varied and with all other parameters fixed are shown in Fig. 3. These data have been processed in the coordinates

$$\Phi \equiv \frac{N_{Nu,\,d_0}}{(\delta/d_0)^{0.09}\;N_{Re,d_0}^{0.74}} - a/d_0$$

and the results indicate that the heat transfer coefficients are inversely proportional to the ratio a/d_0.

The results of test stand experiments with finned tubes can be generalized by the criterial relation

$$N_{Nu,\,d_0} = 0.11\,(\delta/d_0)^{0.09}\,(a/d_0)^{-0.77} N_{Re,\,d_0}{}^{0.74} N_{Pr}{}^{0.33}. \tag{4}$$

The finning index was m_f = 1.68 in these tests. As the finning index is increased to m_f = 2.24, the heat transfer diminishes and the critical relation becomes

$$N_{Nu,\,d_0} = 0.094\,(a/d_0)^{-0.77}\,(\delta/d_0)^{0.09} N_{Re,\,d_0}{}^{0.74} N_{Pr}^{0.33}. \tag{5}$$

Tests were performed with the finning index up to m_f = 5. An analysis of the results has revealed that as the finning index m_f is increased, the coefficient in the criterial relation decreases proportionally to $m_f^{-0.55}$.

The results of this study were compared with those of study [3] on streamlining of blunt bodies by jets (intermediate-intensity heat transfer). Since the Reynolds number $N_{Re,o}$ in the criterial relations there [3] had been referred to the width of the plate and the striking velocity at the jet axis, we recalculated the velocity here according to relation (3). The comparison revealed a discrepancy of not more than 10% for the test with a/d_0 = 13.5, δ/d_0 = 1.63, and w_0 = 27.7 m/sec. This discrepancy is due to the limited applicability of the criterial relations in [3] within the ranges $2 \le \delta/d_0 \le 8$ and $15 \le a/d_0 \le 60$ only, so that the values of these two ratios in this study are smaller than the respective limit values of $\bar{h}$ and $\bar{\varkappa}$ in [3].

Therefore, for calculating the intermediate-intensity heat transfer from blunt bodies with $a/d_0 < 15$ and $\delta/d_0 < 2$ one can recommend the criterial relations (4) and (5).

This evaluation of the dependence on the geometrical parameters has made it possible to determine the optimum design dimensions of a convection surface consisting of finned cylinders. The construction of such a heat exchanger surface has been already described [5]. It consists of a bundle of tubes jointed by two rows of perforated continuous fins. The first row of fins acts as a screen which forms jets and at the second row of fins is impacted by these jets. A construction with $a/d_0 \ge 15$ and $\delta/d_0 \ge 2$ is feasible.

NOTATION

$\varkappa$, a Cartesian coordinate; σ_T^2, dispersion of turbulent transverse deviation of jet volume; δ, distance from a hole in the screen to the plate (cylinder); r, jet radius; D, diffusion coefficient; ε, intensity of turbulence; I_0, zeroth-order Bessel function of an imaginary argument; a, spacing (pitch) of holes in the screen; d_0, initial jet diameter; m_f, finning index; b, plate width; $\bar{\varkappa}$, dimensionless distance from the jet source to the plate; V_0, jet velocity at a hole; V_δ, jet velocity at distance δ from the source; Y^2, dispersion; N_{Nu}, Nusselt number; N_{Re}, Reynolds number; and N_{Pr}, Prandtl number.

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A CIRCULAR TURBULENT JET IN A CROSSFLOW

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An integral method is proposed for calculating a circular turbulent jet propagating in a crossflow. The jet parameters obtained by a numerical method for different values of $\overline{q}$ were compared with experimental data. Satisfactory agreement between the sets of data was found.

The interaction and mixing of jets with a crossflow is a complex form of jet flow, and the study of the propagation of such jets is important for planning and designing equipment and devices in which mixing takes place. Several works by foreign and domestic authors have been devoted to the theoretical [1-5] and experimental [6-13] study of the laws of mixing and propagation of turbulent jets in a crossflow. As a rule, the theoretical studies [1-5] are based on integral methods, assume an increase in jet width, and make various other assumptions regarding the conditions of momentum conservation. Most of the investigations have focused on determining the jet axis, and only certain studies have examined laws of change in width, axial velocity, apparent additional mass, and other parameters.

It was shown in [5] that a jet propagating in a crossflow does not possess the property of similitude. This is evidenced first of all by the fact that, in the construction of lines of equal velocity, the cross sections of the jet change from a circular to a horseshoe shape. Such a change in jet development along its length leads to problems in analytically describing profiles of velocity, temperature, and concentration in its cross sections. In connection with this, it was proposed in [11] that the jet flow region be broken down into three sections, within each of which the flow could be assumed to possess the property of similitude. Meanwhile, according to the data in [11], the determining change in the cross-sectional shape of the jet occurs in the initial section. However, this was not confirmed by the experiment in [7]. In the present work, we attempt to analytically determine the above jet parameters within a broad range of values of the hydrodynamic parameter $\overline{q}$ ($4 \leq \overline{q} \leq 400$). The method of calculation is based on several assumptions: the jet axis is the locus of the points where the velocity for each section normal to the direction of the jet is maximal; the jet is bounded by the surface on which the excess velocity in the direction of the axis decreases to less than a specified low value.

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