HEAT TRANSFER IN THE INITIAL SEGMENT OF A ROTATING TUBE IN THE CASE OF TURBULENT GAS FLOW

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On the basis of theory of local modeling we have determined the effect exerted by rotation on heat transfer in the initial segment of a tube in the case of turbulent gas flow. We have derived approximate working formulas which are compared with experimental data.

The cooling ducts of numerous contemporary engineering installations are subject to the effect of centrifugal forces.

It has now been established both theoretically and experimentally that the centrifugal forces exert a certain effect on the processes of heat transfer and friction in rotating objects, and that this effect is different for internal and external problems.

The theoretical investigations devoted to these questions proceed from the usual semiempirical theory of turbulence.

The complexity of hydrodynamic and thermal processes, particularly for the initial segment of a tube, in the case of diverse boundary conditions (tube length, temperature factor, variation of velocity in the flow core over the length of the tube, distribution of heat load, effect of centrifugal forces, etc.) makes it virtually impossible to determine the mechanism of the process so that it can be used to derive the working formulas. The theoretical investigations have therefore been primarily qualitative in nature up to this point.

Virtually no material has been published on the transfer of heat in the initial segment of a rotating tube, and as regards experimental research, this is always presented in the usual criterial form. The basic shortcoming of such treatment lies in the fact that consideration of all factors affecting heat transfer is associated with great difficulties. In certain cases, by choosing the exponents for the similarity criteria it is possible to group the experimental data about a single curve. However, here we have no assurance as to whether or not the effect of each criterion separately has been properly evaluated, nor whether it has been possible to avoid having these criteria offset one another.

In addition to the usual criterial treatment of experimental data, a new method has been determined for the investigation of heat transfer in the initial segment of a tube, and this procedure is based on the theory of local modeling.

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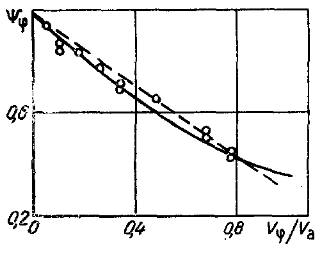

Fig. 1. Effect of tube rotation on the heat-transfer law applicable to a boundary layer. The dashed line denotes the linear approximation of the law.

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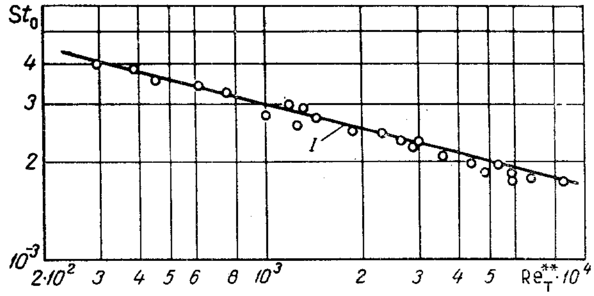

Fig. 2

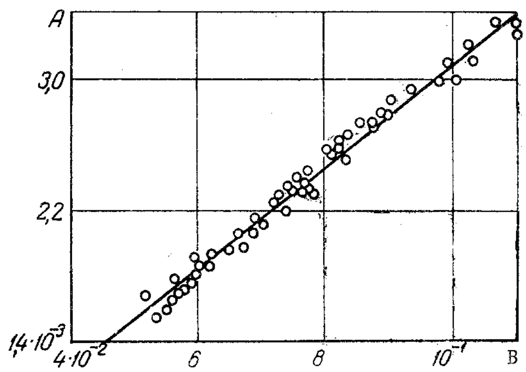

Fig. 3

Fig. 2. Results from the processing of experimental data for the transfer of heat in the initial segment of a rotating cylindrical tube: 1) heat transfer in the turbulent isothermal streamlining of a plate.

Fig. 3. Comparison of experimental data with theoretical data from Eq. (8): $A = St/(1-V_\varphi/V_a)^{0.8}$, $B = [Re_{D_1}^{0.2} Pr^{0.6} \bar{x}^{0.2}]^{-1}$.

The advantages of this method are discussed in [1-4], etc. With this method it is possible to determine the effect on the transfer of heat in the initial segment of a tube as exerted by such boundary conditions as the entry of the flow into the tube, the temperature factor, the distribution law for the heat load, the presence of a transverse flow of matter, and chemical reaction.

The method of local modeling has been extended to the region of stabilized flow in a tube, given an arbitrary distribution of the heat load and substantial nonisothermicity [5].

Here we will attempt to determine the effect of centrifugal forces on the heat-transfer laws applicable to the initial segments of a rotating tube, using the method of experimental data treatment that is based on the theory of local modeling.

The working section of the experimental installation is made up of a calorimeter tube with an inside diameter of 97.6 mm and an overall length of 3000 mm; this tube is mounted on roller-bearing supports and set into rotation (0-1170 rpm). Air is pumped through the working section at a rate of 0.05-0.4 kg/sec. To achieve a uniform velocity profile for the inlet section of the tube, a fixed section 100 mm in length is set up in front of the tube, and the inlet of this fixed section is leminiscate in form. The working section is heated by means of an external electric heater made of a constantan strip that is wound uniformly about the tube.

To avoid the possibility of contact between the constantan strip and the tube surface, a reliable layer of mica and fiber glass insulation is introduced, and to reduce the heat losses to the ambient medium the working section is covered from above by a layer of asbestos insulation.

The experimental data were determined in regimes that were steady with respect to time.

The following parameters were measured during the test: the temperature of the ambient air; the temperature of the air at the inlet to the working section and at the outlet from the working section; the temperature of the inside tube surface, the parameters of the heating current, and the air flow rate.

The temperature of the tube wall was measured by means of copper–constantan thermocouples which were mounted at the following relative distances (x/d) from the inlet: 0.26, 0.77, 1.3, 2.0, 3.1, 4.6, 6.1, 7.8, 8.8, 8.8, 11.9, 13.9, 15.2, 16.2, 17.3, 19.3, 21.4, 22.6, 25.6, 25.7, 28.8. Particular attention was devoted to the manner in which the thermocouples were imbedded. The hot junctions of the thermocouples were soldered to the tube wall, flush with the inside surface of the tube. The thermocouple wiring led from the junction, first in the direction of the isotherms (about the circle formed by the lateral cross section of the tube), and then along the tube, to the measuring instruments through lip-ring wipers. After fabrication, all of the thermocouples were calibrated for a temperature range of 0-300°C. The emf was measured by means of a KP-59 potentiometer.

The air temperature at the inlet to the working section and at the outlet from the working section was measured by means of thermocouples and thermometers. In most of the experiments the measured flow temperature for the outlet cross section differed from the theoretical by no more than 2.5-3%.

To account for the heat losses to the ambient medium, we undertook a special calibration of the installation without a supply of air. The relative magnitude of the heat losses in the experiments came to 3-10% of the power evolved in the heater. The noncoincidence of the power calculated from the electric current (with consideration of the heat losses to the ambient medium) and from the enthalpy of the air flow did not exceed 5-6%.

The determination of the local value for the specific flow of heat over the surface segments of the calorimeter, bearing in mind the variable wall temperature and the corresponding heat losses to the ambient medium, demonstrated that the magnitude of the specific heat flow varied over the surface by no more than 2-3%. In processing the experiments we therefore assumed the specific flow of heat to be constant.

Evaluation of the accuracy of the experimentally derived quantities demonstrated that the maximum probable error in the determination of the Stanton number was about ±9%.

The equation of the thermal boundary layer for the initial segment of the tube, considering rotation, can be presented in the form

$$\frac{d\,\mathrm{Re}_t^{**}}{d\bar{x}} + \frac{\mathrm{Re}_t^{**}}{\Delta T}\,\frac{d(\Delta T)}{d\bar{x}} = \mathrm{Re}_D\,\mathrm{St}_0\psi^{0.5}\,\Psi_\varphi, \tag{1}$$

where $\Psi_\varphi = (\mathrm{St}/\mathrm{St}_0)_{\mathrm{Re}_t^{**}}$ is an unknown function by means of which we account for the effect of tube rotation on the relative change in the Stanton number when Re_t^{**} = idem.

Analysis of the experimental data demonstrated that the effect of tube rotation on the law governing heat transfer in the boundary layer can be taken into consideration only through introduction of the ratio of the circumferential velocity to the axial velocity, i.e., V_φ/V_a.

Figure 1 shows this relationship; these data are taken from the arithmetic mean of three cross sections, i.e., x = 4.6, 15, 28.8. As we can see from the figure, over the entire range of the investigated ratios V_φ/V_a = 0-0.8 we find very weak nonlinearity as regards the solid line showing heat transfer as a function of this parameter. It is probable that a further increase in the V_φ/V_a ratio must bring this function to the exponential law, i.e., at the limit $V_\varphi/V_a \to \infty$ the transfer of heat in the axial direction of the initial tube segment must tend toward zero, i.e., $\mathrm{St} \to 0$.

Within the limits of V_φ/V_a = 0-1 this function can be approximated linearly by

$$\left(\frac{\mathrm{St}}{\mathrm{St}_0}\right)_{\mathrm{Re}_t^{**}} = \Psi_\varphi = 1 - 0.715\,\frac{V_\varphi}{V_a}. \tag{2}$$

The derivation of the final working equations for heat transfer will not be adversely affected by further refinement of (2), as additional experimental and theoretical data are accumulated.

Figure 2 shows the heat-transfer law for the initial segment of a rotating tube, with consideration given to the effect of rotation on the basis of Eq. (2). As we can see from the figure, the experimental points are grouped about a single curve, describing the heat-transfer law for the case of the streamlining of a isothermal plate, with a simultaneous increase in the thermal and dynamic boundary layers. For the region of variation in Re_t^{**} from 10^3 to 10^4, this law can be given by the formula

$$\mathrm{St}_0 = \frac{0.0128}{\mathrm{Re}_t^{**0.25}\mathrm{Pr}^{0.75}}. \tag{3}$$

The increase in the velocity in the flow core in the case of subsonic flow is taken into consideration by means of the following formula, given in [1]:

$$\mathrm{Re}_D = \mathrm{Re}_{D_1} + 5.2\psi\,\mathrm{Re}_T^{**}, \tag{4}$$

while nonisometricity is accounted for by means of the Kutateladze limiting heat-transfer law, which for the region of variation in ψ from 1 to 5 is approximated with sufficient accuracy by the simple exponential formula

$$\left(\frac{\mathrm{St}}{\mathrm{St}_0}\right)_{\mathrm{Re}_t^{**}} = \psi^{-0.5}. \tag{5}$$

We thus have a system consisting of the five equations (1), (2), (3), (4), and (5), with the five unknowns Re_t^{**}, Re_D, ψ, St, Ψ_φ. This system can be solved in various ways, depending on the formulation of the problem.

In many cases, we can neglect the temperature factor and the change in the velocity in the flow core (the comparatively small temperature heads and the low flow rates for the subsonic flow). In this case, from Eq. (1) we can derive the simple formulas for the determination of Re_t^{**} for any law governing the supply of heat.

For example, when

$q_{wall} = \mathrm{const}$

$$\mathrm{Re}_t^{**} = \mathrm{St}_0 \mathrm{Re}_{D_1} \bar{x} \left(1 - 0.715 \frac{V_\varphi}{V_a}\right), \tag{6}$$

$q_{wall} = q_0 \exp a\bar{x}$

$$\mathrm{Re}_t^{**} = \mathrm{St}_0 \cdot \mathrm{Re}_{D_1} \cdot \frac{1}{a} \left(\frac{\exp a\bar{x} - 1}{\exp a\bar{x}}\right) \left(1 - 0.715 \frac{V_\varphi}{V_a}\right). \tag{7}$$

From Eqs. (2), (3), (6), or (7) we derive the following approximate working formulas:

$q_{wall} = \mathrm{const}$

$$\mathrm{St} = \frac{0.0306 \left(1 - 0.715 \dfrac{V_\varphi}{V_a}\right)^{0.8}}{\mathrm{Pr}^{0.6} \mathrm{Re}_{D_1}^{0.2} \bar{x}^{0.2}}, \tag{8}$$

$q_{wall} = q_0 \exp a\bar{x}$

$$\mathrm{St} = \frac{0.0306 \left(1 - 0.715 \dfrac{V_\varphi}{V_a}\right)^{0.8}}{\mathrm{Pr}^{0.6} \mathrm{Re}_{D_1}^{0.2}} \left(\frac{a \exp a\bar{x}}{\exp a\bar{x} - 1}\right)^{0.2}. \tag{9}$$

Figure 3 shows a comparison of experimental data with the results from the calculation of the Stanton number by means of the approximate formula (8). As we can see from the figure, the coincidence of the experimental data with the theoretical function is quite satisfactory. The scattering of the experimental points lies within the experiment's limits of accuracy.

NOTATION

$\mathrm{St} = \alpha / c_p \rho_0 V_{a0}$	is the Stanton number;
$\mathrm{Re}_T^{**} = \rho_0 V_{a0} \delta_T^{**} / \mu_0$	is the characteristic Reynolds number for the thermal boundary layer;
$\delta_t^{**} = \int_0^{\delta_t} (\rho V_a / \rho_0 V_{a0})[1 - (T_w - T) / (T_w - T_0)](1 - y/R) dy$	is the size of the energy loss;
δ_t	is the thickness of the thermal boundary layer;
T	is the absolute temperature;
V_a	is the flow velocity in the axial direction;
V_φ	is the rotational velocity for the tube;
$\psi = T_w / T_0$	is the temperature factor;
$\mathrm{Pr} = \mu c_p / \lambda$	is the Prandtl number;
$\Psi_\varphi = (\mathrm{St}/\mathrm{St}_0)_{\mathrm{Re}_t^{**}}$	is the relative change in the Stanton number in the case of tube rotation;
$\bar{x} = x/d$	is the relative distance from the tube inlet;
R and d	are, respectively, the radius and diameter of the tube;
$\mathrm{Re}_D = \rho_0 V_{a0} d / \mu_0$	is the Reynolds number, on the basis of the parameters outside of the boundary layer;
$\mathrm{Re}_{D_1} = \rho_{01} V_{a01} d / \mu_{01}$	is the Reynolds number with respect to the gas parameters at the tube inlet.