UNIT FOR STUDYING HEAT AND MASS TRANSFER AND FRICTION IN A RECTANGULAR CHANNEL AT REDUCED PRESSURE

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A description is given of a unit which can be used to obtain a hot gas flow in a rectangular channel and to study heat and mass transfer and friction on the surface of different materials. Results from an experimental investigation are presented.

Units designed for testing high-temperature materials in a frontal flow are now being widely used [1]. The study of plate models in tangential gas flows requires a substantial increase in the dimensions of the gas stream compared to the case of frontal flow and, thus, a substantial increase in the power consumed by the electric-arc heater.

The present article proposes a design of unit which makes it possible to study heat and mass transfer and friction on the surface of specimens of thermal-insulating materials made in the form of plates. The unit is characterized by relatively low unit power (Fig. 1). A distinguishing feature of the unit is the use of a micronozzle grid 3 and a movable element in a holder 4 with a tensometric transducer 7 placed in a vacuum chamber 6.

The working medium is air 9, which after heating by the electric-arc heater 1 enters the prechamber 2. It then rotates 90° and enters a rectangular channel 100 × 10 mm in cross section and 11 diameters in relative length. We took the channel height h as the diameter. The lower wall of the channel consisted of two parts: a statinary part, and the movable holder 4. The holder holds the specimen 5, which is in the form of a ground ceramic slab.

Uniform distribution of the flow over the cross section ahead of the channel inlet is ensured by the nozzle grid, made of 2-mm-diam. Nichrome wire with a 2.5-mm mesh. This design makes it possible to obtain a gas flow in the channel with a stagnation temperature as high as 1500°K. The use of two air heaters with counter-directed flows and hollow water-cooled grids makes it possible to raise the temperature of the gas in the prechamber to 6000°K and to significantly expand the range of parameters that can be investigated.

Flow parameters were measured for seven operating regimes of the unit. Air flow rate in the electric-arc heater was changed within the range 1.4-2.8 g/sec, while the flow rate in the prechamber ranged within 4.4-17.7 g/sec. The consumed power was 28 kW for regimes 1-3 and 45 kW for regimes 4-7.

The character of the distribution of static and dynamic pressure along the channel was studied for regimes 1, 2, 4, and 6. For this purpose, we drilled six 1-mm-diam. holes in the lower wall of the channel. The holes were then connected to U-shaped water manometers. The first hole was located 1.5 diameters from the channel inlet, while the rest of the holes were spaced 1.9 diameters from each other.

It is evident from Fig. 2a that the dependence of ΔP on x/h is complex in character as a result of the presence of the nozzle grid ahead of the channel inlet. Due to the structural complexity of the inlet part of the channel, it was not possible to measure static pressure at distances less than 1.5 diameters from the inlet. The zone over which the nozzle grid exerted an effect extended roughly 5 diameters from the inlet. Such a character of change in static pressure is due to redistribution and agitation of the flow as it leaves the grid and enters the channel. Static pressure decreases beginning at distances greater than 5 diameters and continues to decrease up to the channel outlet. Here, the character of the pressure drop is similar to that seen in the flow of gas in the initial section of channels.

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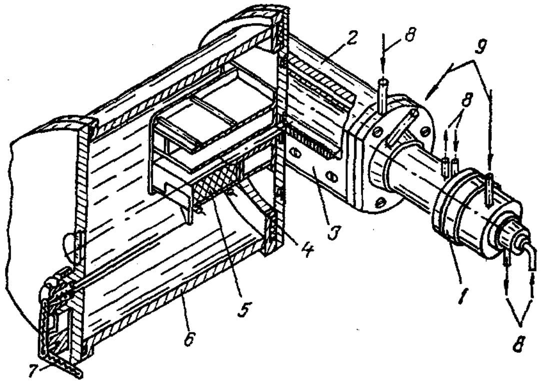

Fig. 1. Diagram of unit.

The stagnation pressure and temperature were measured simultaneously with a combination probe which included a Chromel–Alumel thermocouple and a Pitot tube. The probe took the form of a stainless-steel tube with an inlet hole 0.4 mm in diameter. The probe was moved over the height of the channel by means of special traversing equipment with an accuracy of ±0.05 mm.

Depending on the operating regime of the unit, stagnation temperature on the channel axis changed from 900 to 1500°K. Stagnation pressure ranged within $(7.8\text{–}20)\cdot 10^3$ N/m^2. The completed measurements show that stagnation temperature on the channel axis decreases with an increase in gas velocity. However, it remains nearly constant along the entire channel for a given operating regime. This fact indicates that there is no joining of the thermal boundary layers in the test channel. Neither did we observe joining of the dynamic boundary layers, evidenced by the dynamic pressure profiles obtained in the section 8.5 diameters from the inlet (Fig. 2b). It is apparent from the figure that the thicknesses of the dynamic boundary layer for regimes 1, 2, 4, and 6 average 0.25h. This is evidence of the presence of a potential core.

In studying heat transfer, we faced the inside surface of the walls of the channel with ground ceramic slabs. Ten heat-flux transducers [2] were embedded in the upper wall of the channel. The sensitive element of the transducers was press-fit into a copper jacket. Uniform heating of the transducer and jacket prevented heat flow from the sensitive element. The thermocouples were installed at a distance equal to $\sqrt{3}/3\Delta$ (Δ is the height of the transducer) from the lower end of the transducer. Heating temperature was recorded on a K-2 loop oscillograph. The heat-flux measurement error was roughly 15%. The temperature of the channel wall for different operating regimes ranged from 400 to 700°K. Since the radiative component was no more than 5% of the total heat flux, which is one-third of its measurement error, its value was ignored.

The empirical data on heat transfer was analyzed in criterional form (Fig. 3):

$$\mathrm{Nu}_x = \frac{q_w x}{\lambda (T_{\infty_0} - T_w)}, \quad \mathrm{Re}_x = \frac{\rho_\infty u_\infty x}{\mu_\infty}. \tag{1}$$

Comparison of the experimental values of numbers Nu_x (points) with the calculated values (curve 1) obtained with a relation for laminar flow [3] showed that some disagreement between the data is seen in the range of Reynolds numbers Re_x from $5\cdot 10^3$ to $3\cdot 10^4$. The transitional flow regime evidently begins with a further increase in the number Re_x.

The number Re_δ, calculated from the formula

$$\mathrm{Re}_\delta = \frac{\rho_\infty u_\infty \delta}{\mu_\infty} \tag{2}$$

using experimental values of the thickness of the boundary layer δ, was within the range $1\cdot 10^3\text{–}3\cdot 10^3$. This corresponds to the region of transition from laminar flow to turbulent flow on a flat plate.

Surface friction was measured with a movable (floating) element, which made it possible to sufficiently simply determine τ_w with different operating regimes. The method of measuring surface friction was as follows. A movable element consisting of the holder 4 and specimen 5 (see Fig. 1) was installed on supports. Translation of the element by a rod was communicated to the tensometric transducer 7. The signal from the transducer, passing through a "Topaz" tensometric station, was recorded on a KSP-4 potentiometer.

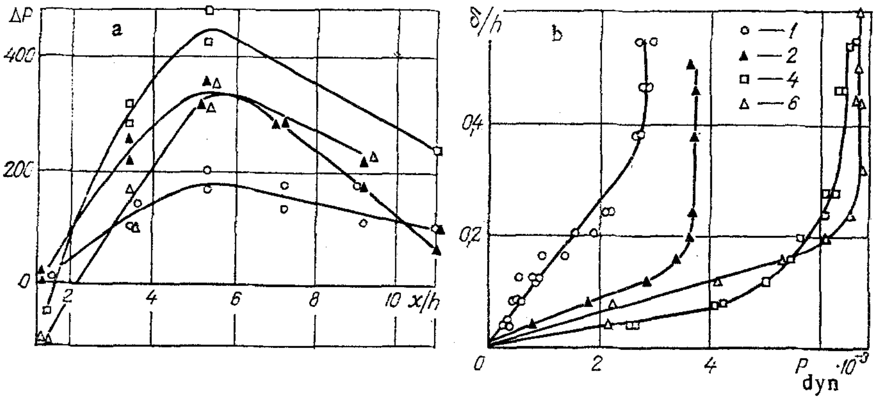

Fig. 2. Distribution of static and dynamic pressure in chamber: a) change in static-pressure drop along channel: points denote test results (ΔP, N/m²); b) profiles of dynamic pressure in the section x/h = 8.5: points denote experiment; 1, 2, 4, and 6 denote regimes of operation of the unit (P_{dyn}, N/m²).

During the experiments the movable element was moved by friction on the side of the incoming flow, the static-pressure drop on the specimen surface, and inertial forces. This movement was opposed by the weight of the element, friction in the supports, and the force due to the difference between the atmospheric pressure and the pressure in the vacuum chamber. This force acted on the end of the rod. Allowing for the effect of these factors in the given system was complicated, since they are variable and depend on the displacement of the movable element. Thus, before any measurements were made, we calibrated the system in the vacuum using a set of small weights. This allowed us to consider the total effect of all forces (except for inertia) and, with a known pressure drop on the specimen, to determine the shear stress on the wall.

The shear stress was calculated from the test data by the formula [4]

$$\tau_w = \frac{F - \Delta P S_{\text{end}}}{S}. \tag{3}$$

The mean value of τ_w along the specimen calculated by Eq. (3) was, for example, 18.6 N/m² for regime 2.

The value of τ_w can also be obtained from test data on heat transfer. If we set Pr = 1 and reason that the Reynolds analogy is valid on the channel section where τ_w was measured, then

$$\tau_w = \frac{q_w u_\infty}{c_p (T_{\infty_0} - T_w)}. \tag{4}$$

The value of τ_w calculated from (4) for regime 2 is 21.5 N/m².

The following are among the important results discovered in analyzing the test data. Foremost, measurements of velocity profiles in the boundary layer in a section located a distance x/h = 8.5 from the inlet showed that the boundary layers on opposite sides of the channel do not join together. Also, it was found by approximation of these velocity profiles by the power relation

$$\frac{u}{u_\infty} = (y/h)^n \tag{5}$$

that the superscript n in (5) approaches the value 1/7 for the flow regimes studied in the tests. Finally, the stagnation temperature on the flow axis can, within the limits of accuracy of the experiment, be considered constant along the entire channel. All this makes it possible to solve problems on heat transfer in a turbulent gas flow in the initial section of the channel.

There are presently a large number of different empirical methods of solving such problems [5-8, etc.]. Here, we will use the method of Kutateladze and Leont'eva [9] proposed to calculate friction and heat transfer in turbulent gas flow in the initial section of a cylindrical pipe. The advantage of this method is that it sufficiently simply makes it possible to determine the friction and heat-transfer coefficients within a broad range of Reynolds, Prandtl, and Mach numbers. Here, the problem was solved in a boundary-layer approximation for a plane-parallel channel, with the hydraulic diameter having been used as the characteristic dimension. We examined a steady turbulent flow of a gas in a channel of height h. The velocity at the channel inlet (x = 0) was assumed to be subsonic. The velocity and temperature profiles at the channel inlet were taken as uniform over the channel height. It was also assumed that the dynamic and thermal boundary layers increased simultaneously from the inlet section of the channel. With allowance for these assumptions, we write the integral relations for the momenta and energy in the form:

$$\frac{d\,\mathrm{Re}_{\delta_2}}{dx}+\mathrm{Re}_d(1+H)\,F_{\delta_2}=\mathrm{Re}_d\,\frac{c_f}{2}, \tag{6}$$

$$\frac{d\,\mathrm{Re}_{\delta_{2T}}}{dx}=\mathrm{Re}_{\delta_{2T}}F_{\delta_{2T}}=\mathrm{Re}_d\mathrm{St},$$

$$\mathrm{Re}_{\delta_2}=\frac{\rho_\infty u_\infty \delta_2}{\mu_{\infty_0}},\ F_{\delta_2}=\frac{\delta_2}{u_\infty}\frac{du_\infty}{dx},\ \mathrm{Re}_{\delta_{2T}}=\frac{\rho_\infty u_\infty \delta_{2T}}{\mu_{\infty_0}},$$

$$F_{\delta_{2T}}=\frac{1}{\Delta T}\frac{d\Delta T}{dx},\ \mathrm{Re}_d=\frac{\rho_\infty u_\infty d}{\mu_{\infty_0}},\ H=\delta_1/\delta_2, \tag{7}$$

$$c_f=2\tau_w/\rho_\infty u_\infty^2,\ \mathrm{St}=q_w/c_p\rho_\infty u_\infty\Delta T,\ \Delta T=T_w^*-T_w,$$

T_w^* is the equilibrium temperature of the wall, $\bar{x}$ = x/d, and d = 2h in the hydraulic diameter of the channel.

Following [9], we introduce the following laws of friction and heat transfer into our examination:

$$\Psi=\left(\frac{c_f}{c_{f_S}}\right)_{\mathrm{Re}_{\delta_2}},\ \Psi_T=\left(\frac{\mathrm{St}}{\mathrm{St}_S}\right)_{\mathrm{Re}_{\delta_{2T}}}. \tag{8}$$

Presented below are results of an analytical solution of system (6-7) with allowance for (8) obtained, in accordance with [9], for the boundary condition T_w = const and used in the present study to determine c_f and St.

The change in velocity in the core of the flow is determined by the expression

$$(2.25+1.625\psi)\left[4(\bar{u}-1)^{0.25}-\frac{1}{\sqrt{2}}\ln\frac{(\bar{u}-1)^{0.5}+\sqrt{2}(\bar{u}-1)^{0.25}+1}{(\bar{u}-1)^{0.5}-\sqrt{2}(\bar{u}-1)^{0.25}+1}-\right.$$

$$\left.-\sqrt{2}\,\mathrm{arc\,tg}\frac{\sqrt{2}(\bar{u}-1)^{0.25}}{1-(\bar{u}-1)^{0.5}}\right]-(1+1.3\psi)\frac{(\bar{u}-1)}{\bar{u}}=\frac{0.4\psi^{0.25}x}{(\sqrt{\psi}+1)^2\mathrm{Re}_{1w}^{0.25}}, \tag{9}$$

where $\bar{u}=u_\infty/u_{\infty 1}$; $u_{\infty 1}$ are the velocities on the axis at x = 0; $\psi=T_w/T_{\infty 0}$ is the temperature factor; $\mathrm{Re}_{1w}=\rho_{\infty 1}u_{\infty 1}2h/\mu_w$ is the Reynolds number, determined from the parameters at the channel inlet and the wall temperature.

The Reynolds number with respect to the momentum thickness

$$\mathrm{Re}_{\delta_2}=\frac{\mathrm{Re}_{1w}(\bar{u}-1)}{5.2\psi}. \tag{10}$$

The local friction coefficient

$$c_f=\left(\frac{\mu_w}{\mu_\infty}\right)^{0.25}\left(\frac{2}{\sqrt{\psi}+1}\right)^2\frac{0.0256}{\mathrm{Re}_{\delta_2}^{0.25}}. \tag{11}$$

The Reynolds number with respect to the energy thickness

$$\mathrm{Re}_{\delta_{2T}}=\frac{\mathrm{Re}_{1w}(\bar{u}-1)}{5.2\psi}\left\{(2+1{,}3\psi)-\frac{1.25+1.625\psi}{(\bar{u}-1)^{1,25}}\left[4(\bar{u}-1)^{0.25}-\right.\right. \tag{12}$$

$$\left.\left.-\sqrt{2}\,\mathrm{arc\,tg}\frac{\sqrt{2}(\bar{u}-1)^{0.25}}{1-(\bar{u}-1)^{0.5}}-\frac{1}{\sqrt{2}}\ln\frac{(\bar{u}-1)^{0.5}+\sqrt{2}(\bar{u}-1)^{0.25}+1}{(\bar{u}-1)^{0.5}-\sqrt{2}(u-1)^{0.25}+1}\right]\right\}^{0.8}.$$

TABLE 1. Parameters of the Flow

Number of regime	Re_{1w}	Pr	ψ	$\frac{\mu_w}{\mu_\infty}$
1	$5{,}35\cdot 10^3$	0,724	0,480	0,646
2	$8{,}5\cdot 10^3$	0,719	0,452	0,612
4	$2{,}45\cdot 10^4$	0,715	0,410	0,470
6	$1{,}53\cdot 10^4$	0,719	0,485	0,755

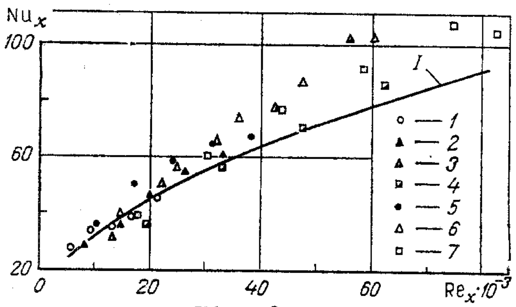

Fig. 3

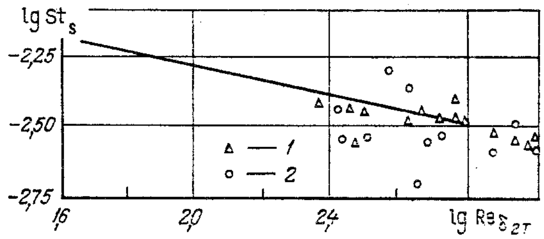

Fig. 4

Fig. 3. Dependence of the number Nu_x on the Reynolds number Re_x: points denote empirical data; I denotes calculation by the relation in [3]; 1-7 denote operating regimes of the unit.

Fig. 4. Dependence of log St_s on log $Re\delta_{2T}$: the straight line denotes calculation by the method of [9]; 1 denotes the tests in [10]; 2 denotes the tests in [11].

The local heat-transfer coefficient

$$St = \left(\frac{\mu_w}{\mu_\infty}\right)^{0.25}\left(\frac{2}{\sqrt{\psi}+1}\right)^2 \frac{0.0128}{Re_{\delta_{2T}}^{0.25} Pr^{0.75}}. \tag{13}$$

Using the algorithm (9)-(13), we solved the problem numerically on a computer for the values of the flow parameters found from the tests (see Table 1). Some of the results of the computations are examined below.

For the values of the temperature factor obtained in the tests, the theoretical ratio δ_{2T}/δ_2 does not exceed 1.06 along the entire channel. Thus, we can approximately take $Re_{\delta_{2T}} \approx Re_{\delta_2}$. On the other hand, the fact that the ratio $\delta_{2T}/\delta_2 > 1$ is explained by the fact that $Pr < 1$.

The change in the friction and heat-transfer coefficients c_f and St along the channel obtained in the calculations showed that the principal relation of the Reynolds analogy deviates negligibly from these results. If we introduce the proportionality factor $A = Pr^{-2/3}$, then the Reynolds analogy is nearly satisfied:

$$St = \frac{1}{2} c_f A. \tag{14}$$

Figure 4 shows the dependence of log St_s on log $Re\delta_{2T}$, where all of the data is referred to the thermal insulation conditions in accordance with Eq. (8). The values of St_s for all four flow regimes (see Table 1) lie on one straight line. Also, this line (more precisely, its right side) passes roughly through the middle of the empirical scatter [10, 11]. Such a result is evidence of the generality of the friction and heat-transfer laws for flows in the initial sections of channels and flow about a flat plate.

Results of calculation of heat transfer in [9] were compared with calculations in [12]. The number St was determined from the following formula in accordance with [12].

$$St = \frac{c_f}{2} Pr^{-0.5}\left\{\frac{1+\left[1-\frac{4K}{(c_f/2)^2}\right]}{2}\right\},$$

where $K = (\nu_\infty/u_\infty^2)(du_\infty/dx)$ is the acceleration parameter. The small (on the order of 8%) difference in the numbers St calculated by the different methods is due to the fact that small acceleration parameters K (pressure gradients) do not have a significant effect on the heat-transfer law for the flow under consideration.

The studies completed here permit us to conclude that the proposed unit makes it possible to reproduce flow conditions sufficiently close to the conditions of external flow about a flat plate.

NOTATION

c_p, heat capacity at constant pressure; c_f, friction coefficient; d, hydraulic diameter of channels; $H = \delta_1/\delta_2$, form parameter; h, channel height; F, force due to friction of flow against the wall and the static pressure drop; F_{δ_2}, $F_{\delta_{2T}}$, form parameters; K, acceleration parameter; P, pressure; Pr, Prandtl number; q, heat flux on the wall; Re, Reynolds number; S, cross-sectional area of specimen; S_{end}, area of end of specimen; St, Stanton number; Nu, Nusselt number; T, temperature; u, longitudinal component of velocity; x, y, longitudinal and transverse coordinates; δ, thickness of boundary layer; δ_1, displacement thickness; δ_2, momentum thickness; δ_{2T}, energy thickness; Δ, height of transducer; λ, thermal conductivity; μ, absolute viscosity; ν, kinematic viscosity; ρ, density; τ_w, shear stress on the wall; Ψ, Ψ_T, relative laws of friction and heat transfer; ψ, temperature factor. Indices: 1, inlet (x = 0); w, channel wall; ∞, channel axis; s, "standard conditions;" 0, stagnation.

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