EXPERIMENTAL INVESTIGATION OF THE HEAT EXCHANGE OF A SUBSONIC JET FROM A NORMALLY DISPOSED FLAT OBSTACLE

I. A. Belov, G. F. Gorshkov,
V. S. Komarov, and V. S. Terpigor'ev

UDC 536.244:532.522

Results of investigations are processed in the form of dependences for the Nusselt number at the stagnation point taking account of free-stream pulsations. The experiments are compared with a computation by an approximate method [8].

Results of the experimental investigations of heat fluxes from a jet to normally disposed flat obstacles outside the dependence on the jet parameters in a nozzle exit section [1-4] differ significantly from the computed magnitudes of the heat fluxes computed by means of known dependences for a uniform stream in the neighborhood of the stagnation point [5]. Large discrepancies between theory and experiment cannot be explained by inaccuracies in the computation but are associated with the known singularity of jet flows, namely, with the pulsations (turbulence) of the gas dynamic parameters of the jet stream.

The influence of the turbulence of a subsonic axisymmetric cold jet on the heat exchange with a hot flat obstacle disposed along the normal to the jet axis is investigated herein. Because of the compexity of the problem posed and the lack of a sufficient quantity of data permitting association of the pulsation characteristics of a jet stream with parameters averaged with respect to time, we decided to gather experimental results on jet turbulence characteristics near obstacles and to find an empirical relation between these latter and the magnitudes of the heat fluxes from the jet to the obstacle in direct proximity to the stagnation point.

The investigation was conducted on an experimental apparatus consisting of a wind tunnel, a flat obstacle with a mounted heater, a coordinate apparatus and measuring unit. The obstacle was a 300 × 200 × 10 mm Textolite slab on which 0.1 mm thick tape heating elements were glued. Chromel–Copel thermocouples from 0.1 mm diameter wire were fastened to the inner surface of the element. Prior to fabrication

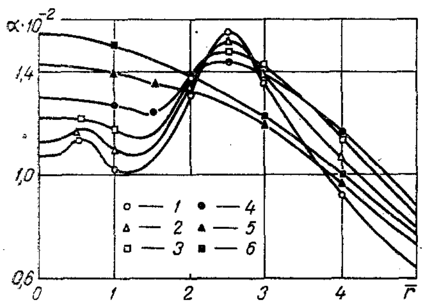

Fig. 1

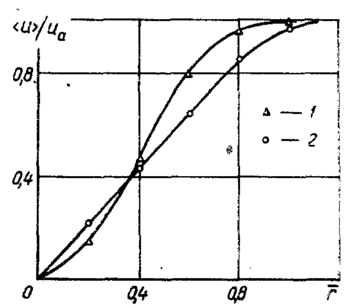

Fig. 2

Fig. 1. Distribution of the heat-transfer coefficient over the obstacle (u_a = 21 m/sec, d_a = 100 mm, α in W/m^2·deg): 1) $\bar{X}$ = 0.5; 2) 1.0; 3) 2.0; 4) 3.0; 5) 4.0; 6) 5.0.

Fig. 2. Distribution of the time-averaged velocities over the obstacle (u_a = 8 m/sec, d_a = 100 mm): 1) $\bar{X}$ = 0.5; 2) $\bar{X}$ = 2.0.

Leningrad Mechanical Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 20, No. 5, pp. 893-897, May, 1971. Original article submitted June 10, 1970.

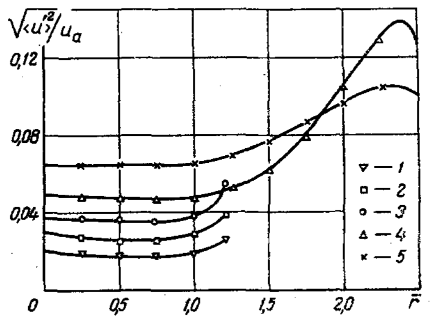

Fig.3

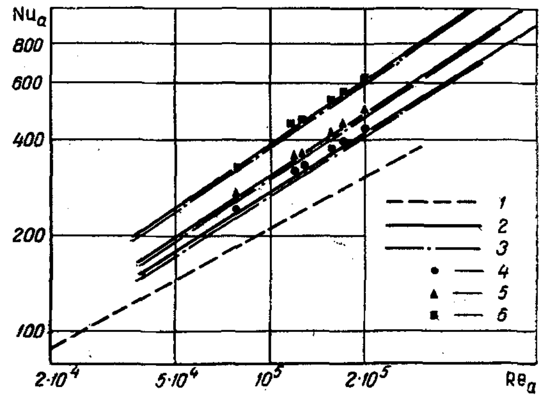

Fig.4

Fig.3. Distribution of the root-mean-square velocity pulsations over the obstacle (u_a = 8 m/sec, d_a = 100 mm): 1) $\bar{X}$ = 1.0; 2) 2.0; 3) 3.0; 4) 4.0; 5) 5.0.

Fig.4. Dependence of the number Nu at the stagnation point on the number Re: 1) computation from [5]; 2) from [3]; 3) from [8]; 4-6) experiment; 4) $\bar{X}$ = 0.5, ε = 2.6%; 5) respectively 2.0, 3.5; 6) 5.0, 6.5.

of the junction, the measuring ends of the thermocouple were flattened to thicknesses on the order of 20 μ. To eliminate any electrical contact between the thermocouple and the heating element, mica spacers about 10 μ thick were used. The thermocouple was calibrated against a standard mercury thermometer prior to each series of experiments. The time-averaged and pulsation characteristics (velocities) were measured by using a standard dc hot-wire anemometer.

Under stationary heat exchange conditions, by neglecting radiation because of the smallness of the wall temperature, the thermal interaction between the jet and the hot plate is determined from the heat balance condition

$$Q_{\text{elec}} = Q_w + Q_{\text{loss}}, \tag{1}$$

where Q_{elec} = IU is the heat liberated in the plate because of the passage of the electric current; Q_w is the heat flux removed by the gas; Q_{loss} is the heat losses in the Textolite. The following expression for the heat transfer coefficient

$$\alpha = \frac{\frac{IU}{S} - \lambda_{\text{т}} \frac{T_w - T_{\text{am}}}{h}}{T_w - T_e}, \tag{2}$$

follows from (1), where S is the surface of the heated plate; h is the thickness of the Textolite obstacle; T_{am} is the temperature of the ambient medium; T_e is the recovery temperature (determined experimentally under the action of a jet on the obstacle with disconnected heating element, I = 0).

An investigation of the flow field near the obstacle and the thermal effect was carried out on the initial portion of the jet in the range of variation between the nozzle exit and the obstacle $\bar{X} = X/d_a$ = 0.5-5.0. The diameter (d_a) of the nozzle exit section varied between 10-100 mm, the stream velocity (u_a) at the nozzle exit varied between 10-300 m/sec, and the Reynolds number calculated by means of the nozzle exit parameters were in the range Re = 10^4-$2\cdot10^5$.

Typical distribution profiles of the local heat transfer coefficients α over the obstacle surface are presented in Fig.1 for different ranges $\bar{X}$ between the nozzle exit and the obstacle, from which it follows that the distribution profile of the heat transfer coefficient typical for the interaction between a completely developed turbulent jet and an obstacle for the $\bar{X}$ range considered (see [6], for example) does not change only for large values of $\bar{X}$. For $\bar{X}$ = 5, the heat transfer coefficient is practically constant in direct proximity to the stagnation point ($\bar{r}$ = 0) and then diminishes uniformly as $\bar{r}$ increases. For short ranges between the nozzle exit and the obstacle, the α distribution curve has a minimum at the stagnation point. A number of papers investigating the heat exchange between axisymmetric [1] or plane jets [2] and normally disposed obstacles report on the existence of a minimum value of α at the stagnation point for small ranges between the nozzle exit and the obstacle. According to [2], this minimum exists at $\bar{X} < 0.5$ for plane jets. As the results of the investigations conducted show, in the case of an axisymmetric jet the range $\bar{X}$ of existence of minimum α at the stagnation point is broadened and reaches to $\bar{X} = 3$.

The distribution of the heat transfer coefficient α over the obstacle surface for $\bar{X} \leq 1.0$ is characterized by the presence of two maximums far from the stagnation point: centrally at a spacing $\bar{r} = 0.8$ from the stagnation point and peripherally at $\bar{r} = 2.5$. As the results of investigations of the time-averaged velocity distributions over the obstacle surface show (Fig. 2), for $\bar{X} \leq 1.0$ the central heat flux maximum corresponds approximately to the location of the maximum in the gradient of the mean stream velocity with respect to time, i.e., the appearance of a central maximum can be explained by the nonuniformity of the velocity distribution over the obstacle surface. In the domain of jet rotation on the obstacle, for $\bar{r} > 1.0$ the jet mixing zone with the ambient medium emerges on the obstacle surface, which significantly increases the intensity of jet stream turbulence in the near-wall domain, and hence results in the appearance of the peripheral heat flux maximum. As the results of measuring the stream velocity pulsations in the near-wall domain (Fig. 3) show, the root-mean-square value of the velocity pulsations $\sqrt{\langle u'^2 \rangle}$ approximately equals the velocity pulsation in corresponding sections of a free jet [7].

An increase in the heat flux at the stagnation point and the disappearance of the central and peripheral maximums in α as $\bar{X}$ increases are explained by the increase in turbulence intensity on the jet axis for $\bar{X} > 4.0$. This results in measured high values of the heat flux at the stagnation point with a maximum corresponding to the transition section of the jet $\bar{X} \approx 8$.

Results of an experimental investigation of the heat flux at the stagnation point are presented in Fig. 4 in criterial form. Taking account of the experimental results on the magnitude of the turbulent stream intensity in the neighborhood of the stagnation point which were obtained for the same stream parameters at the nozzle exit, the criterial dependence of the heat exchange at the stagnation point is described with sufficient accuracy by the following empirical formula:

$$\mathrm{Nu}_a = 0.8\,\mathrm{Pr}^{0.4}\,\mathrm{Re}_a^{0.5}\,\bar{X}^{-0.08}\,(1 + 0.8\varepsilon^{1.1}\,\mathrm{Re}_a^{0.28}), \tag{3}$$

where $\mathrm{Nu}_a = \alpha d_a/\lambda$, $\mathrm{Re}_a = u_a d_a/\nu$; $\varepsilon = \sqrt{\langle u'^2 \rangle}/\langle u_a \rangle$.

The results of a computation using (3) agree well with the results of the approximate solution elucidated in [8]. For $\varepsilon = 0$ formula (3) yields results which agree with the computation of Nu_a at the stagnation point for a uniform stream impinging on an obstacle [5].

NOTATION

r	is the distance along the obstacle to the stagnation point;
X	is the distance from the nozzle exit to the obstacle;
$\bar{r} = r/d_a$;	
$\bar{X} = X/d_a$;	
d_a	is the diameter of the nozzle exit section;
h	is the obstacle thickness;
u	is the velocity;
T	is the temperature;
$\mathrm{Nu} = \alpha d/\lambda$, $\mathrm{Re} = ud/\nu$, $\mathrm{Pr} = \mu c_p/\lambda$	are the Nusselt, Reynolds, and Prandtl numbers;
α	is the heat transfer coefficient;
λ	is the heat conduction coefficient;
ν	is the kinematic viscosity;
ε	is the turbulence intensity;
U	is the voltage drop on the plate;
I	is the current intensity flowing over the plate;
S	is the plate surface.

Subscripts

a	is the nozzle section;
w	is the wall;
T	is Textolite;

am is the ambient medium;
e is the recovery parameters.

LITERATURE CITED

1. R. Gardon and F. Akfirat, Int. Journ. of Heat and Mass Transfer, 8, 1261 (1961).
2. R. Gardon and F. Akfirat, Trans. ASME, Journ. of Heat Transfer, 88, 101 (1966).
3. I. P. Ginzburg, I. A. Belov, V. A. Zazimko, and V. S. Terpigor'ev, Trans. Third All-Union Conf. on Heat and Mass Exchange [in Russian], Énergiya, Moscow (1968).
4. I. A. Belov, I. P. Ginzburg, V. A. Zazimko, and V. S. Terpigor'ev, Trans. Third All-Union Conf. on Heat and Mass Exchange [in Russian], ITMO (1969).
5. M. Sibulkin, Mekhanika, 3, 19 (1953).
6. K. Perry, The Inst. of Mechanical Engineers, London, 168, 775 (1954).
7. P. O. A. L. Davies, M. J. Barrat, and M. J. Fisher, J. Fluid Mech., 15, 337 (1963).
8. I. A. Belov and V. S. Terpigor'ev, Inzh. Fiz. Zh., 17, No. 6 (1969).