METHODS OF INTENSIFICATION OF HEAT-EXCHANGE PROCESSES DURING TESTING OF HEATPROOF MATERIALS

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An experimental study is made of the intensification of heat-exchange processes in an instrument for testing heatproof materials without considerable modernization of existing instruments through an increase in the intensity of tubulence of the impinging stream.

The testing of heatproof materials (HPM) is at present a necessary stage in the finishing and study of the heatproof coatings (HPC) of various instruments. The widely known methods of testing HPM in a jet of high-temperature gas directed normal to the surface of the specimen reproduce well enough the aerodynamic heating of HPC at pressures close to atmospheric.

However, in design practice one must often deal with HPC which operate under conditions of increased pressures at relatively low gas flow velocities. An experimental instrument for conducting such tests can consist of a small solid-fuel or liquid gas generator whose combustion products escape from a channel into some restricted space containing a barrier, mounted normal to the stream, on which the test HPM is placed. However, the necessity of obtaining high heat loads leads to a considerable increase in the dimensions and energy consumption of such instruments.

Therefore a study was made of the intensification of heat-exchange processes without considerable modernization of instruments for testing HPM.

As experiments have shown in advance, the heat exchange from the gas to the wall in instruments of this type depends considerably on the intensity of turbulence of the impinging stream. Two methods of increasing the intensity of turbulence, and consequently the heat flux, were studied:

1) an increase in the relative distance from the channel mouth to the cover;

2) the use of turbulizing grids mounted within the channel at different distances from its mouth.

The experiments were conducted both on cold air (the average and pulsation velocities of the stream and the heat exchange between the air and a preheated barrier were measured) and on a hot gas with a stagnation temperature of $T_0 = 1000°K$ (the heat fluxes from the gas to the wall were measured).

The experimental study was conducted on an instrument consisting of a high-pressure chamber joined by a channel to a cylindrical bounded space ending in a flat cover. To eliminate the influence of external conditions on the gas flow in the bounded space the discharge of the latter was accomplished through four supersonic nozzles in the cover with an equivalent diameter of the critical cross section of 36 mm. In

TABLE 1

Grid number	Diameter of openings, mm	Center-to-center distance, mm	Number of openings	Obstruction coefficient
1	5	10	37	0,81
2	4	10	31	0,90

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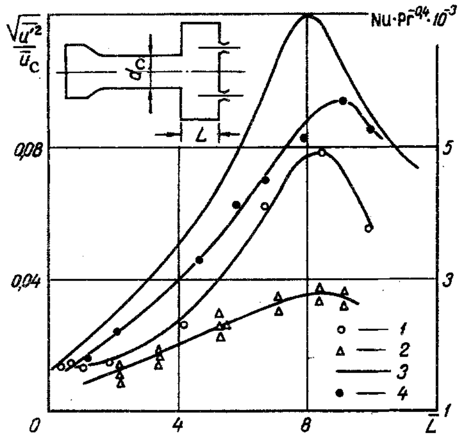

Fig. 1

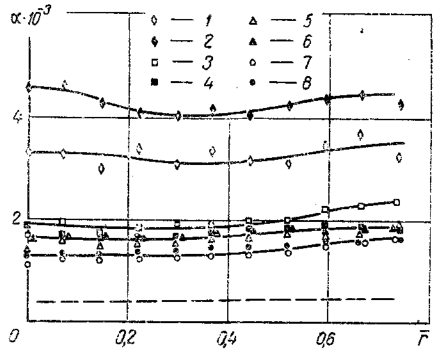

Fig. 2

Fig. 1. Distributions of Nusselt numbers at stagnation point and intensity of turbulence as a function of the distance L: 1) d_c = 37 mm, $Re_c = 2.4 \cdot 10^6$, air; 2) 43 mm, $5.5 \cdot 10^5$, combustion products; 3) subsonic flooded jet; 4) bounded space, d_c = 37 mm, $Re_c = 2.4 \cdot 10^6$.

Fig. 2. Distribution of local heat exchange coefficients "between nozzles" on flat cover (d_c = 70 mm, $Re_c = 1.3 \cdot 10^6$, $\bar{L} = 1$). Turbulizing grids: 1) $\bar{x} = 0$, grid No. 1; 2) 0, No. 2; 3) 0.5, No. 1; 4) 0.5, No. 2; 5) 1, No. 1; 6) 1, No. 2; 7) 2, No. 1; 8) 2, No. 2; dashed line: without grids; α, $W/m^2 \cdot {}^\circ K$.

studying the average and pulsation velocities of the stream and the heat exchange between the air and the preheated barrier compressed air was supplied to the high-pressure chamber by a wind tunnel having a brief duration of action. In studying heat exchange on a hot gas the combustion products of an alcohol—air mixture with a coefficient of oxidant excess of 1.6 from a liquid gas generator were supplied to the same chamber. Intermediate inserts were used to increase the distance from the channel mouth to the flat cover, which allowed this distance to be varied from 0.25 to 10 channel diameters. An assembly of grids, whose characteristics are given in Table 1, was provided in the channel 70 mm in diameter for turbulization of the stream emerging into the bounded space.

A textolite barrier with a built-in ribbon heating element, for which a description and the method of determining the coefficients of heat exchange are presented in [1], was fastened to the inner surface of the flat cover to measure the heat exchange of the heated plate with cold air.

Heat-flux pickups of the calorimetric type were used to measure the heat exchange of the hot gas with the wall. The heat exchange coefficient α was determined from the data of the heat flux measurement at known wall and gas temperatures T_W and T_0, which were also measured, from the equation

$$\alpha = \frac{q_w}{T_0 - T_w}.$$

The average and pulsation velocities of the stream were measured by a constant-temperature thermoanemometer in which a tungsten filament 6.5 μ in diameter and 1 mm long was used as the sensitive element. At the selected degree of superheating of the wire N = 2 a time constant $M = 4 \cdot 10^{-6}$ sec provided for a frequency range of measurements up to 25 kHz.

The study was conducted with the following range of variation of the principal stream parameters: gas velocity coefficient at the exit from the channel λ_c = 0.17–0.58, stagnation pressure of stream $P_0 = (3\text{–}21) \cdot 10^5$ N/m^2, Reynolds numbers calculated from the parameters at the channel mouth $Re_c = (0.8\text{–}10) \cdot 10^6$, channel diameter d_c = 37–70 mm, distance from channel mouth to cover $\bar{L} = L/d_c$ = 0.25–10, distance from channel mouth to grid $\bar{x} = x/d_c$ = 0–2 (measured inside channel).

The distributions of the Nusselt numbers at the stagnation point as a function of the distance $\bar{L}$ from the channel mouth to the flat cover are presented in Fig. 1 (curves 1 and 2). As follows from the graph, the thermal effect of the gas jet on the barrier increases with greater distance from the channel mouth, reaching its maximum at $\bar{L} \simeq 8.5$, where the heat exchange coefficient is increased threefold (at Re_c = 2.4 10^6) in comparison with small distances from the channel mouth to the flat cover. The increase in heat

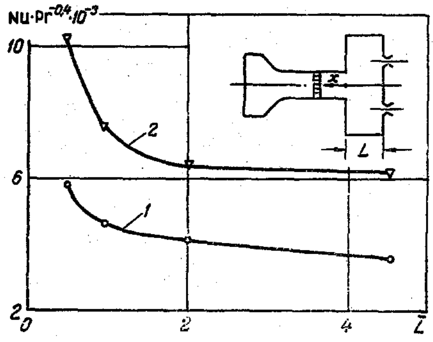

Fig. 3

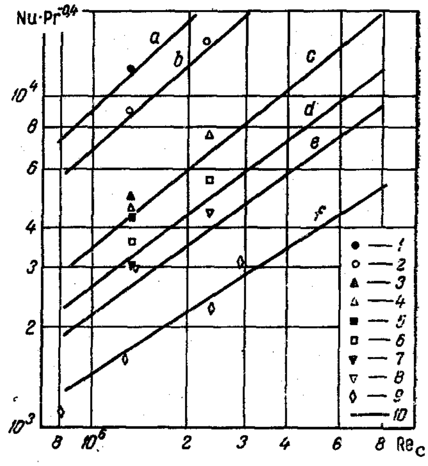

Fig. 4

Fig. 3. Dependence of Nusselt number at stagnation point on distance $\bar{L}$ with installation of turbulizing grid (d_c = 70 mm, $\bar{x}$ = 0.5, grid No. 1): 1) $Re_c = 1.3 \cdot 10^6$; 2) $2.4 \cdot 10^6$.

Fig. 4. Dependence of Nusselt number on Reynolds number in central part of flat cover (d_c = 70 mm, $\bar{L}$ = 1). Experiment: 1) $\bar{x}$ = 0, grid No. 2; 2) 0, No. 1; 3) 0.5, No. 2; 4) 0.5, No. 1; 5) 1, No. 2; 6) 1, No. 1; 7) 2, No. 2; 8) 2, No. 1; 9) without grids; calculation: 10) after [1], a) ε = 0.32; b) ε = 0.23; c) ε = 0.11; d) ε = 0.08; e) ε = 0.06; f) ε = 0.03.

exchange in this case is explained by the increase in turbulence of the impinging stream with larger $\bar{L}$ (see Fig. 1, curve 4), which also leads to high Nusselt numbers at the stagnation point with a maximum corresponding to the transitional section of the jet ($\bar{L} \simeq 8.5$). Since, as the results of preliminary studies showed, the gas temperature in front of the cover in the range of variation of the distance from the channel mouth to the flat cover studied remains approximately constant and equal to the stagnation temperature at the channel mouth, the nature of the variation in heat flux is analogous to the nature of the variation in the heat exchange coefficient.

A certain difference in the positions of the maxima of the Nusselt number Nu and of the intensity of turbulence ε (the maximum of the Nu number is located closer to the channel mouth) can be explained by the fact that along with ε the velocity gradient in the vicinity of the stagnation point affects the value of Nu. The velocity gradient outside the initial section of the jet ($\bar{L}$ = 4-5) does not remain constant but has a tendency to decrease. The joint influence of these two opposing effects leads, as indicated above, to a shift in the Nusselt number maximum.

As follows from Fig. 1, where the distribution of $\sqrt{\overline{u'^2}}$ in the free, axially symmetrical, flooded turbulent jet (curve 3) is presented along with the variation in velocity pulsation $\sqrt{\overline{u'^2}}$ near the flat cover of the instrument studied (curve 4), the distributions of velocity pulsations have a common nature in the two cases. This property confirms the similarity of the interactions in the vicinity of the stagnation point in a bounded space and at an unbounded barrier.

The method of intensification of heat exchange examined above makes it possible with the same gas parameters to increase the coefficient of heat exchange and consequently the heat flux by three times. Obtaining even higher specific fluxes requires the artificial turbulizing of the gas, which can be obtained with the help of turbulizing grids mounted in the supply channel at different distances from its mouth.

As the study conducted showed, the intensity of turbulence ε_c at the channel mouth varies as a function of the position of the grid in the channel with respect to its mouth from 6 to 23% at $\bar{x}$ = 2-0 for grid No. 1 and from 9 to 28% at $\bar{x}$ = 2-0 for grid No. 2. The mounting of grids directly at the channel mouth leads not only to a high initial intensity ε_c but to considerable dynamic irregularity of the stream emerging into the bounded space. However, this property does not have much effect on the distribution of heat exchange coefficients α over the surface of the cover in the investigated range of variation of the principal stream parameters, which is confirmed by the results of the experimental study presented in Fig. 2. Thus, the installation of turbulizing grids leads to an increase in specific heat fluxes while preserving the uniform distribution of α in the central part of the flat cover. This permits the use of turbulizing grids to increase the heat flux in instruments for HPM testing.

The considerable increase in the heat exchange coefficient at a fixed distance from the channel mouth to the cover (Fig. 2) upon the approach of the grid to the channel mouth allows one to regulate the thermal load on the HPM while retaining the original stream parameters.

On the other hand, the heat exchange coefficient also varies considerably at a fixed $\bar{x}$ with variation in the distance $\bar{L}$ from the channel mouth to the flat cover (Fig. 3). The heat transfer is increased upon the approach of the flat cover to the channel mouth. The greatest increase in heat flux is observed when $\bar{L} < 2$. The increase in heat exchange in this region can be explained by a growth in the intensity of turbulence at the stagnation point with a decrease in the distance from the cover to the grid.

A generalization of the results of the present study in criterial form is presented in Fig. 4. The results of a calculation after [1] for the specific turbulence intensities obtained in the present study are plotted here (curves a, b, c, d, e, f). As seen from the figure, the use of turbulizing grids allows one to considerably (by 10 times) increase the heat exchange coefficient and consequently the heat flux from the gas to the wall.

The satisfactory agreement of the calculation with the experiment allows one to estimate the heat flux from the gas to the test HPM from the known parameters of the turbulizing grids.

NOTATION

r	is the radial distance from stagnation point along cover;
L	is the distance from channel mouth to cover;
$\bar{r} = r/d_c$; $\bar{L} = L/d_c$;	
d_c	is the channel diameter;
$\bar{u}$	is the velocity;
$\sqrt{\overline{u'^2}}$	is the root-mean-square value of velocity pulsation;
ε	is the intensity of turbulence;
α	is the heat exchange coefficient, $W/m^2 \cdot {}^\circ K$;
Nu, Re, Pr	are the Nusselt, Reynolds, and Prandtl numbers;
c	are the parameters at channel mouth.

LITERATURE CITED

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