EXPERIMENTAL HEAT TRANSFER INVESTIGATION IN HEAT SHIELD MATERIAL TEST FACILITIES

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An experimental investigation has been made of the thermal interaction of the gas in a facility for testing heat shield materials. The results are presented in terms of the Nusselt number at the stagnation point of a flat plate, accounting for turbulent fluctuations of the oncoming stream. The data obtained are compared with the theory of reference [4].

A possible method of investigating heat shield materials is to test them on the flat rear cover plate of a small model solid or liquid fuel gas generator. To increase the velocity of the combustion products (and thereby increase the heat flux) one can use an intermediate channel of diameter less than that of the gas generator chamber, coupling the gas generator to a short pre-nozzle section.

Analysis of a qualitative picture of the flow in the short section has shown that a subsonic turbulent jet forms there, which intereacts with the flat plate and discharges through the supersonic nozzles. In these circumstances, therefore, the interaction within the short section, in the vicinity of the stagnation point on the plate, reduces to the problem of flow of a subsonic turbulent jet over an infinite two-dimensional obstacle normal to the flow axis.

The heat flux from a subsonic jet to a two-dimensional infinite obstacle, within the entrace section, has been investigated in [1—4], where it was shown that the measured heat flux exceeds that predicted by known correlations (e.g., [5]), because the jet turbulence affects the heat transfer.

References [3, 4] used an analysis of experimental heat flux data and measured turbulence characteristics to derive a correlation for the heat transfer coefficient, allowing for the turbulence of the incident stream. Since one must know the turbulence intensity in order to apply the theoretical relations of [3, 4] to the above problem of the short pre-nozzle section, we attempted to determine the heat transfer from the gas to the wall, and also to measure the fluctuation characteristics along the axis of the jet discharging into a short section as shown in Fig. 1.

The investigation of the gasdynamic and thermal parameters of the flow in a model facility of this kind (Fig. 1) was carried out with cold air. Air from a high-pressure chamber 1 passed through channel 2 into a short section 3 of inside diameter 140 mm. To eliminate the effect of external conditions on the gas flow in the short section, the gas discharged to the surrounding space via the supersonic nozzles 4, which had an equivalent sonic throat diameter of 36 mm.

The surface for the heat transfer investigation was a flat plate, to whose inside surface was fastened a textolite obstacle carrying a strip heating element 5; the heater and the technique for determining the heat transfer coefficient are described in [4]. Two variants for the location of the heating element are provided for: in line with two nozzles, and in between nozzles, to allow us to obtain the distribution of local heat transfer coefficient in the most representative directions of the gas flow over the plate.

The velocity fluctuations of the flow were measured using a constant temperature hot wire anemometer.

The investigation was carried out in the following range of the basic parameters at the end of the

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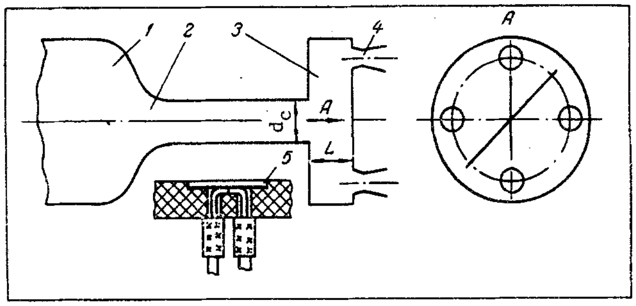

Fig. 1. Schematic diagram of the experimental facility.

nozzle: λ_c = 0.17 to 0.58, stagnation pressure P_0 = (3 to 21)·10^5 N/m^2, Reynolds number $Re_c = \lambda_c a_s D_c \rho_0 / \mu$ = (0.8 to 10)·10^6, channel diameter D_c = 37 to 70 mm, distance from end of nozzle to plate $\bar{L} = L/D_c = 1$.

The distribution of local heat transfer coefficient α over the plate surface is shown in Fig. 2. It can be seen that the distribution depends on the gas flow direction (between nozzles or in line with nozzles) and on the gasdynamic parameters in the channel entrace section. In the range of the basic parameters investigated a uniform distribution of heat transfer coefficient in the vicinity of the central point of the plate occurs for $\bar{r} = r/D_c < 0.3$. This distribution results from the velocity gradient and the turbulent intensity being constant in this region, and also from the fact that the efflux of gas into the nozzles has little effect on the flow in the vicinity of the plate center. Thus, the region $\bar{r} < 0.3$ can be used to test heat shield materials under uniform heat loading conditions.

For $\bar{r} > 0.3$ the nature of the distribution of local heat transfer coefficient depends appreciably on both the flow direction and on the Reynolds number. The flow in this region is appreciably affected by efflux of gas into the nozzles, which accelerates the flow, thus increasing the velocity gradient, and in turn the heat transfer. The effect of the efflux on the heat transfer increases with increasing Reynolds number.

Figure 2 also shows (broken line) the distribution of α over the surface of an infinite two-dimensional obstacle washed by a subsonic axisymmetric jet [4]. It can be seen that in the central region ($\bar{r} < 0.3$) the distribution of heat transfer coefficient agrees both quantitatively and qualitatively. For $\bar{r} > 0.3$ the divergence in the distribution of α arises, as was noted above, from the effect of efflux of gas into the nozzles, as well as from a number of other known peculiarities of flow of gas in the short section.

Analysis of the experimental data on heat transfer to the central part of the plate shows that the coefficients obtained are considerably larger than the calculated values according to [5], and that the excess increases with increase of Reynolds number. As was pointed out above, the derivation is associated with

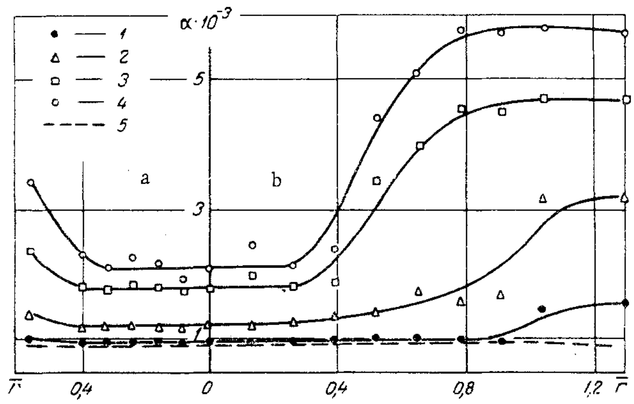

Fig. 2. Distribution of local heat transfer coefficient (a) in line with nozzles, and (b) between nozzles, on the flat plate. D_c = 37 mm, $\bar{L}$ = 1: 1) Re_c = 1.4·10^6; 2) 2.4·10^6; 3) 4.3·10^6; 4) 5.2·10^6; 5) after [4]. $\alpha \cdot 10^{-3}$, W/m^2/°K.

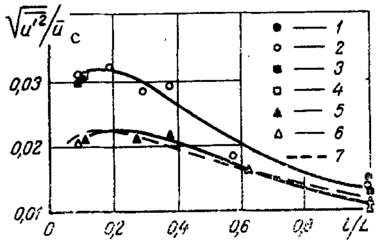

Fig. 3

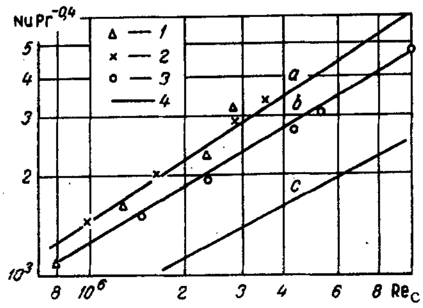

Fig. 4

Fig. 3. Distribution of turbulent intensity along the axis of the short section: 1) D_c = 52 mm, $\bar{L}$ = 1, $Re_c = 10^6$; 2) 52, 1, $1.6 \cdot 10^6$; 3) 70, 1, $0.8 \cdot 10^6$; 4) 70, 1, $1.3 \cdot 10^6$; 5) 37, 1, $1.4 \cdot 10^6$; 6) 37, 1, $2.4 \cdot 10^6$; 7) after [7].

Fig. 4. Nusselt number as a function of Reynolds number in the central part of the flat plate. Experiment: 1) D_c = 70 mm, $\bar{L}$ = 1; 2) 52, 1; 3) 37, 1. Calculation: 4) from [4]; a—ε = 0.03; b) 0.02; c) 0.

the effect of turbulence inherent in jet flows.

In order to elucidate this effect we measured the intensity of turbulence along the axis of the short section, and some of the results are shown in Fig. 3. It can be seen that the turbulent intensity at the channel outlet varied from 1 to 1.5%. The low level of fluctuations at the channel exit in comparison with fully developed pipe turbulence (ε = 5%, [6]) can be explained by the small relative length of the channel (less than ten diameters). As we approach the wall ε increases, and reaches 2 to 3% close to the wall. This variation of ε along the axis of the short section corresponds to the distribution of turbulent intensity in the case of flow of a subsonic axisymmetric jet over a two-dimensional infinite obstacle (Fig. 3, broken line, [7]).

Figure 4 shows the experimental results for heat transfer in the central part of the plate, in correlation form, as a function of the turbulent intensity. The different values of heat transfer for different channel diameters can be explained by the effect of the flow turbulence. Since the turbulent intensity varies as D_c varies (see Fig. 3), but remains practically constant over a range of one channel diameter, the experimental values of Nusselt number are grouped into a family of curves where the turbulent intensity ε appears as a parameter, i.e., Nu = f(Re) for ε = const. The similarity noted above as regards the distribution of heat transfer coefficient and the turbulent intensity between the flow in the short section and with interaction of a subsonic axisymmetric jet flowing over a two-dimensional infinite obstacle normal to it justifies our use of the relation [4]:

$$Nu = 0.8\,Pr^{0.4}(\bar{L})^{-0.08}(1 + 0.8\varepsilon^{1.1}Re_c^{0.28})\,Re_c^{0.5},$$

where $Nu = \alpha D_c/\lambda$; $Re = \bar{U}_c D_c/\nu$; $Pr = c_p\mu/\lambda$; $\varepsilon = \sqrt{\overline{u'^2}}/\bar{U}_c$ for calculation of heat transfer in the central part of the plate. Figure 4 shows the results of calculations using this relation (curves a, b, and c) for the values of incident stream turbulent intensity given in Fig. 3. It can be seen that there is satisfactory agreement between the measured and calculated values.

This investigation of the flow and heat transfer in facilities for testing heat shield materials allows the following conclusions to be drawn:

When calculating heat transfer in a short section of channel one must take into account the turbulence of the incident stream, otherwise the heat flux values will be too small (by a factor of about two).

One can calculate the approximate heat transfer in the central part of the facility ($\bar{r}$ < 0.3) from the correlation obtained in [4] for flow in the vicinity of the stagnation point of a subsonic jet impinging normally on an infinite obstacle.

The turbulent intensity ε required for the calculation of the heat transfer should be determined near the flat end plate of the facility in each specific case; for an estimate of ε one can use available data for free jets where the end of the nozzle is far from the obstacle.

NOTATION

l and r	are the cylindrical coordinates;
L	is the distance from the end of the channel to the plate;
$\bar{r} = r/D_c$; $\bar{L}/D_c$;	
D_c	is the channel diameter;
$\overline{U}$	is the velocity;
$\sqrt{\overline{u'^2}}$	is the root mean square velocity fluctuation;
ε	is the intensity of turbulence;
α	is the heat transfer coefficient, $W/m^2/deg$;
Nu, Re, and Pr	are the Nusselt, Reynolds, and Prandtl numbers, respectively;
λ	is the thermal conductivity;
ν	is the kinematic viscosity;
μ	is the dynamic viscosity;
c_p	is the specific heat at constant pressure;

Subscripts

c refers to parameters at the end of the channel.

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