Journal of Engineering Physics and Thermophysics, Vol. 77, No. 3, 2004

THERMAL STATE OF THE ELEMENTS OF ACTIVE AND COMBINED THERMAL PROTECTION IN GASDYNAMIC TESTS

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A complex experimental-computational procedure of investigation of heat exchange on the exterior surface of a porous element of active thermal protection and along the surface of an element of combined thermal protection has been given; the procedure is based on the processing of results of thermogasdynamic tests by the methods of inverse heat-conduction problems. Results of investigation of the influence of the parameters of porous cooling on the efficiency of thermal protection of the heated surface have been presented. Results of investigation of the intensity of heat exchange on the surface of a glass-reinforced-plastic plate in the curtain zone for different parameters of injection have been given, and conclusions on the efficiency of porous cooling as a function of the intensity of injection of a coolant have been drawn.

Efficient means of protecting structures against the thermal and erosion action of high-enthalpy single- and multiphase flows include the methods of active and combined thermal protection in injection of a coolant gas through a permeable surface. Problems on investigation of physicochemical processes in active injection of a coolant have been the subject of numerous works, for example, [1–3]. In developing systems of active and combined thermal protection, particular significance is attached to the data obtained in bench tests of different cooling systems under conditions that model the actual conditions of thermal and gasdynamic loading. The basic parameters characterizing the above means of thermal protection are the nonstationary temperature of the surface $T_w(t)$, the density of the heat flux to the surface $q_w(t)$, and the mass rate of thermochemical destruction $(\rho v)_w$ in the presence of the removal of mass from the surface in the curtain zone. However, the conditions of bench tests preclude, as rule, direct measurement of nonstationary thermal loads on the working surface of porous elements. In this case, the most efficient method is a complex computational-experimental approach that is based on the employment of special temperature sensors and heat flux transducers with methodological support [4, 5]; this approach enables one to restore with a sufficient degree of accuracy nonstationary thermal loads on the exterior surface through the employment of indirect data on a change in the temperature at one or several points in the depth of the element under study in the process of gasdynamic tests.

In this work, we have presented a procedure of investigation of heat exchange by the methods of inverse heat-conduction problems and have analyzed the results of investigation of heat exchange on the exterior surface of the elements of a system of active and combined thermal protection according to the data of laboratory gasdynamic tests.

Heat exchange on the surface of porous elements of active thermal protection has been investigated under the following assumptions:

(1) the porosity of the material is determined by the presence of just transport pores whose number remains constant in the process of the tests;

(2) the pressure difference along the pore length is constant: ΔP = const;

(3) radiation heat exchange is absent on both the exterior surface and the pore wall;

(4) the injected coolant gas is inert to the material of a thermal-protection element throughout the range of temperatures realized in the process of the tests;

(5) heat transfer by conduction in the gas phase is negligible as compared to the convective heat transfer inside the pores.

In the general case, the heat exchange is calculated based on a mathematical model allowing for the process of transfer of heat in both the solid matrix and the gas phase. However, as has been substantiated in [1, 6], when the heat exchange is intense, the local temperature equilibrium

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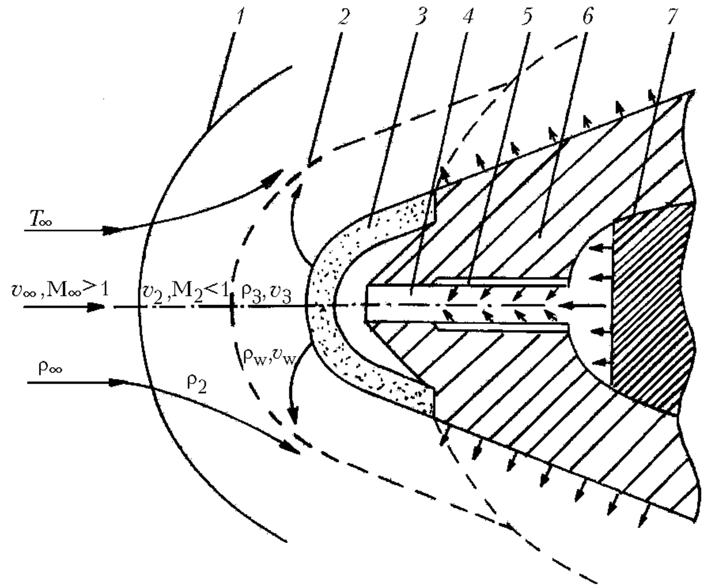


Fig. 1. Scheme of the method of active thermal protection: 1) shock wave; 2) dividing streamline; 3) permeable head; 4) channel for feeding the coolant gas; 5) solid coolant; 6) heatproof material; 7) gas generator.

$$T_s(x) \approx T_g(x), \quad x \in [0; l] \tag{1}$$

is established between the solid skeleton and the filtered gas.

Condition (1) holds [1] for values of the coefficients of internal heat exchange of $\alpha_v = (10^3–10^4)$ kW/(m^3·K). In this case, according to the evaluations of [6], the porous heatproof material can be considered as a hypothetical heat insulator with an effective thermal conductivity dependent on the thermophysical properties of the actual material of the solid skeleton and the coolant gas:

$$1)\ \lambda_{ef} = \lambda_s + \lambda_g, \quad \text{if} \quad \lambda_s \approx \lambda_g;$$

$$2)\ \lambda_{ef} = \lambda_s, \quad \text{if} \quad \lambda_s >> \lambda_g.$$

Thus, under the actual conditions of thermal loading, it is required that the value of the coefficient of internal heat exchange be estimated from experimental data in advance so as to identify nonstationary thermal loads on the permeable surface in active injection of the coolant.

Models manufactured from stainless steel have been subjected to tests in a nitrogen-plasma jet under laboratory conditions for complex investigation of heat exchange on the permeable surface of an element of active thermal protection. The scheme of cooling is shown in Fig. 1.

The thermocouples were installed at the entry of the coolant into the pores on the surface opposite to the heated one and at a single point in depth. The time of blowing of the specimens was $t_f = 16$ sec. The models represented porous hemispheres with a diameter of the base of $D = 2 \cdot 10^{-2}$ m, a thickness of the porous wall of $l = 2 \cdot 10^{-3}$ m, a diameter of the pores of $d = 100$ μm, and a porosity of $\Pi = 10\%$. The rate of flow of the gas through a single pore was $G_p = 3.5 \cdot 10^{-2}$ kg/sec. The mass flow rate of the coolant and the pressure at the inlet to the pores were maintained constant: $G = 2.8 \cdot 10^{-4}$ kg/sec and $P = 34$ bar. The temperature of the coolant gas was 300 K, and the specific mass rate of feed was $(\rho v)_g = 4.46$ kg/(m^2·sec). To evaluate the heat-transfer coefficient in the model experiment we employed the following data on the thermophysical properties of the materials of the solid and gas phases: $\rho_s = 7.8 \cdot 10^3$ kg/m^3 [7], $c_s = 0.5 \cdot 10^3$ J/(kg·K) [7], $\lambda_s = 40$ W/(m·K) [8], $\lambda_g = 2.74 \cdot 10^{-2}$ W/(m·K) [9], $c_{pg} = 1.094 \cdot 10^3$ J/(kg·K) [9], and $\mu_g = 1.84 \cdot 10^5$ kg/(m·sec) [9].

Under the conditions of the tests, the local Reynolds number was $\mathrm{Re}_0 \approx 24$ and the coefficient of internal heat exchange was $\alpha_v = 6.64 \cdot 10^3$ kW/(m^3·K). Since the Re_0 value obtained is much lower than 200 and the coefficient of internal heat exchange is $(1000 < \alpha_v < 10{,}000)$ kW/(m^3·K), we can assume, according to [1, 6], that a local temperature equilibrium is established between the solid skeleton of the porous specimen and the coolant gas under the experimental conditions in question [3]. The condition $\lambda_s >> \lambda_g$ is fulfilled for the stainless-steel models. In this case, the process of propagation of heat into the porous specimen is modeled mathematically by the one-dimensional nonlinear heat-conduction equation for the material with effective thermophysical characteristics

$$(\rho c)_s \frac{\partial T(x,t)}{\partial t} = \frac{\partial}{\partial x}\left(\lambda(T)\frac{\partial T(x,t)}{\partial x}\right) - (\rho v)_g\, c_{pg} \frac{\partial T(x,t)}{\partial x}\,,$$
$$0 < x < 1\,, \quad 0 < t \le t_f\,. \tag{2}$$

To determine the temperature field inside the specimen we supplement Eq. (2) with initial and boundary conditions:

$$T(x, 0) = T_0 \equiv \mathrm{const}\,, \quad 0 \le x \le l\,; \tag{3}$$

$$T(0, t) = T_w(t)\,, \quad 0 \le t \le t_f\,; \tag{4}$$

$$-\lambda_s \frac{\partial T(l,t)}{\partial x} = \alpha_0 \left(T_1^{\exp}(t) - T_g\right), \quad 0 \le t \le t_f\,. \tag{5}$$

The system of relations (2)–(5) is closed by the additional condition

$$T(l, t) = T_1^{\exp}(t)\,, \quad 0 \le t \le t_f\,. \tag{6}$$

In relations (2)–(6), the temperature of the filtered coolant gas is T_g = 300 K $\equiv$ const and $(\rho v)_g c_{pg}$ = $4.88 \cdot 10^2$ W/(m^2·K) $\equiv$ const.

The coefficient of heat exchange α_0 at the inlet to the pores has been evaluated based on rough calculations [10].

Thus, the formulated problem (2)–(6) is a boundary-value inverse heat-conduction problem by whose solution one determines the nonstationary temperature and density of the heat flux on the exterior surface of the element of active thermal protection. The procedure of investigation by the methods of inverse heat-conduction problems was analogously applied to study heat exchange along the surface of a plate of heatproof material in the curtain zone.

Flow of the wall jet along the plate with the formation of a wall boundary layer has been modeled in experiments on investigation of heat exchange on the surface of a thermal-protection element in the presence of a curtain zone [11]. Plates of heatproof material were installed along the axis of a plasma jet generated by a low-temperature-plasma plasmatron. Local injection of the coolant (gaseous nitrogen) was carried out perpendicularly to the surface of the plate through a number of holes located in front of it.

In the series of typical experiments, the parameters of the flow on the plate's edge were as follows: enthalpy of the incoming gas H_e = 5400 kJ/kg, Reynolds number $\mathrm{Re} = 2 \cdot 10^4$, and Mach number $\mathrm{M} \le 0.3$.

In the process of the tests, we continuously measured the nonstationary temperature on the exterior and interior surfaces of the active-protection element. The change in the geometric characteristics of the plane due to thermochemical destruction was determined experimentally [12].

One basic assumption in investigating heat exchange in injection of the gas through a permeable surface is the assumption of the equality of the temperatures of the porous skeleton and the coolant at any cross section of the porous wall. A possible difference between the temperatures of the porous skeleton and the coolant in a semitransparent porous wall heated by the radiative-convective heat flux has been evaluated in [13]. The temperature levels and gradients for semitransparent bodies are lower than those for opaque bodies by virtue of the volume character of heating.

It has been shown that the temperature profiles over the wall thickness become flatter in surface layers, which results in a reduced difference of the temperatures of the porous skeleton and the injected gas. The possibility of the assumption of the equality of temperatures as applied to calculation of the temperature state of a glass-reinforced-plastic heatproof material under the conditions of thermochemical destruction has also been shown in [14].

The employment of the assumption that the temperatures of the porous skeleton and the coolant gas are equal enables us to consider a semitransparent material in porous injection as a heatproof material with effective thermophysical characteristics.

Taking into account the above propositions, we determine the thermal state of the plate in each cross section by solution of the primal problem of heat conduction in the region with a moving external boundary:

$$\rho c\,(T)\,\frac{\partial T\,(x, t)}{\partial t} = \frac{\partial}{\partial x}\left(\lambda\,(T)\,\frac{\partial T\,(x, t)}{\partial x}\right), \tag{7}$$

$$0 < x < \delta\,(t)\,,\quad 0 < t < t_{\mathrm{f}}\,,$$

$$T\,(x, 0) = T_0 \equiv \mathrm{const}\,,\quad 0 < x < \delta\,(t)\,, \tag{8}$$

$$T\,(0, t) = T_{\mathrm{w}}^{\exp}\,(t)\,,\quad 0 \le t \le t_{\mathrm{f}}\,, \tag{9}$$

$$T\,(\delta\,(t), t) = T_1^{\exp}\,(t)\,,\quad 0 \le t \le t_{\mathrm{f}}\,. \tag{10}$$

Then the nonstationary heat fluxes along the plate are determined from the relation on the exterior surface

$$q_{\mathrm{w}}\,(t) = -\,\lambda\,(T)\,\frac{\partial T_{\mathrm{w}}^{\exp}\,(t)}{\partial x}\,,\quad 0 \le t \le t_{\mathrm{f}}\,. \tag{11}$$

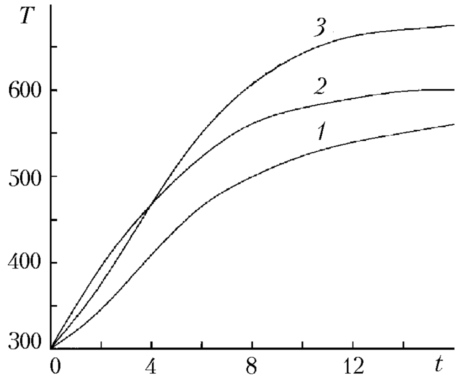
In problem (7)–(11), $\delta(t)$ is the variable thickness of the plate, determined experimentally [12] and $\rho c(T)$ and $\lambda(T)$ are the coefficients of volumetric heat capacity and thermal conductivity respectively.

According to [8], the formulated problem (7)–(11) is pseudoinverse.

Both the inverse problem (2)–(6) and the pseudoinverse problem (7)–(11) belong to the class of ill-posed problems, i.e., small errors in the initial information can lead to large errors in the characteristics sought. Not only do ill-posed problems necessitate highly accurate experimental information on the change in the temperature $T_{\mathrm{w}}^{\exp}(t)$ and $T_1^{\exp}(t)$ in numerical realization, but they also necessitate temperature dependences $\rho c(T)$ and $\lambda(T)$ consistent with the conditions of the tests. As is well known [15–17], the thermophysical properties of glass-reinforced plastics can significantly change not only with the regimes of external thermal loading but also with the technological methods of their manufacture. However, the density of the heatproof material and the coefficient of specific heat are present in problem (7)–(11) only in the form of a complex coefficient of volumetric heat capacity ($\rho c(T)$) whose value varies from the initial one less than twofold under the experimental conditions in question [1]. Since this coefficient appears in (7) as a linear factor, it is easy to allow for the influence of the error on the reliability of solution in assigning it. The values of the effective coefficient of thermal conductivity $\lambda(T)$ nonlinearly appearing in (7) are more sensitive to thermochemical changes in the heatproof material and can differ by an order of magnitude for the same material depending on the regime of external thermal loading, which cannot be disregarded in solving ill-posed problems.

Thus, it becomes necessary to determine the effective thermophysical characteristics of decomposing glass-reinforced plastic which enable us to determine the intensity of heat exchange on the destroyed surface with the required degree of accuracy. One method of determination of the effective coefficient of thermal conductivity based on solution of the coefficient inverse heat-conduction problem has been presented in [18].

Taking account of what has been said above, we divide the solution of problem (7)–(11) into two steps. In the first step, according to the data of temperature measurements at several points in depth, we have solved the coef-



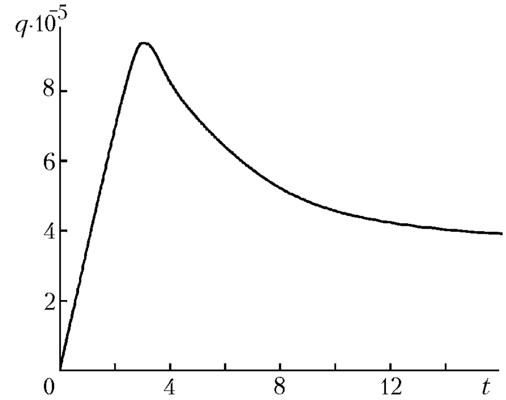


Fig. 2. Thermal state of the porous specimen in the tests: 1) $T_1^{\exp}(t)$, temperature at the inlet to the pores; 2) temperature at a depth of 10^{-3} m; 3) $T_w(t)$, temperature of the exterior surface. T, K; t, sec.

Fig. 3. Density of the heat flux on the exterior surface of the element of active thermal protection in the tests. q, W/m^2; t, sec.

ficient inverse problem of heat conduction in the region with a moving external boundary whose position has also been determined experimentally in [19]. In the second step, we have numerically determined the temperature field and the density of the nonstationary heat flux on the exterior plate surface from relation (11) for the known thermophysical characteristics of the plate material under study and in the case of temperature measurements on both surfaces.

The above procedure of complex computational-experimental investigation of heat exchange on the exterior surface of the elements of thermal-protection systems was employed for studying the influence of the characteristics of porous cooling on its efficiency.

We give results of the computational-experimental investigation of heat exchange on the surface of the elements of active and combined thermal protection.

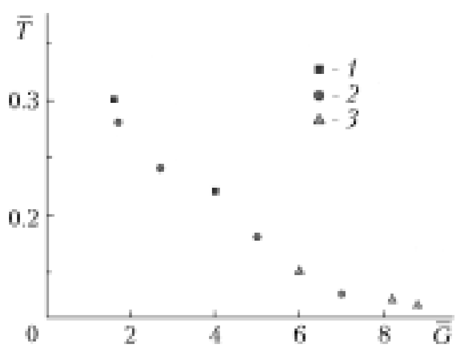
The results of thermocouple measurements of the temperature in one typical test of porous specimens in active injection of a coolant are presented in Fig. 2. Curve 3 shows the data on a change in the temperature on the exterior surface of a model porous specimen that have been calculated based on solution of the boundary-value inverse heat-conduction problem (2)–(6) from the temperature measured at the inlet to the pores (curve 1). Curve 2 shows the readings of the control thermocouple.

Figure 3 shows the nonstationary heat flux on the exterior surface of a model porous specimen; it has been determined by solution of the inverse heat-conduction problem (2)–(6).

The reliability of the results obtained numerically has been monitored by the "thin wall" method according to the procedure of [20]. The legitimacy of the employment of this method of monitoring at small times ($0.1 < t \le 0.5$ sec) is based on evaluations by the Bi and Fo numbers [21]. According to [20], the error of this method amounts to less than 4% for the experimental conditions in the time interval in question. The stationary value of the heat-flux density in the presence of injection is determined from the relation [21]

$$\overline{q}_{\text{st}} = (\rho c)_{\text{s}}\, l \left.\frac{\partial T_{\text{s}}}{\partial t}\right|_{t=t^*} + (\rho c)_{\text{g}}\, c_{p\text{g}}\, (T_{\text{s}} - T_{\text{g}})\,\Big|_{t=t^*} .$$

A comparison of the values of the stationary heat flux $\overline{q}_{\text{st}}$ obtained by the "thin wall" method and the nonstationary heat flux $q_{\text{w}}(t)$ obtained by solution of the inverse heat-conduction problem according to the data of thermophysical experiments has shown that the maximum difference is attained at t = 0.5 sec and amounts to $\pm$7%, which is comparable to the error of the experimental measurements of temperature. The comparison made points to the reliability of the values of the nonstationary characteristics of heat exchange that have been obtained with the use of sensors with methodological support.



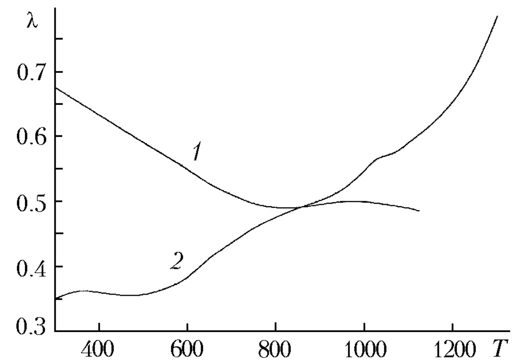


Fig. 4. Temperature of the surface of porous models vs. injection parameter: 1) hemisphere, Π = 0.16; 2) plane end, Π = 0.4; 3) hemisphere, Π = 0.3.

Fig. 5. Coefficient of thermal conductivity of the heatproof material: 1) H_e = 3.2; 2) 6.2 MJ/kg. T, K; λ, W/(m·K).

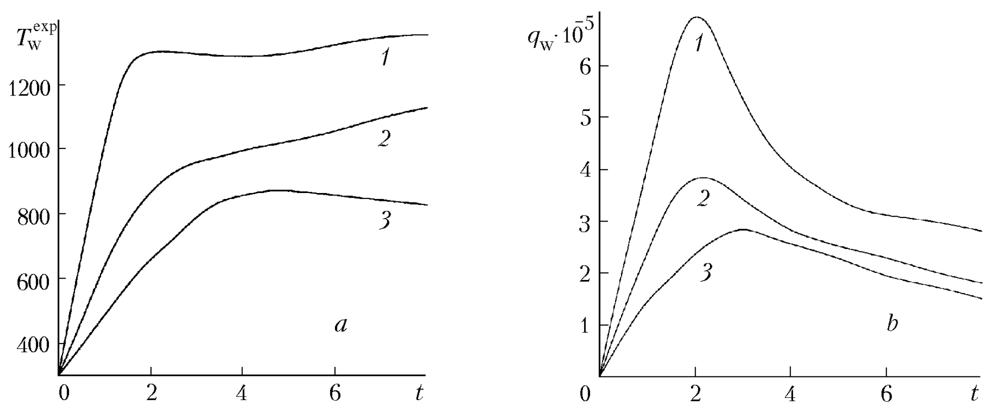


Fig. 6. Temperature along the surface of the plate (a) and the heat-flux density along the surface of the plate (b) in the curtain zone: 1) $\overline{y}$ = 0.15; 2) 0.3; 3) 0.6. q_w, W/m^2.

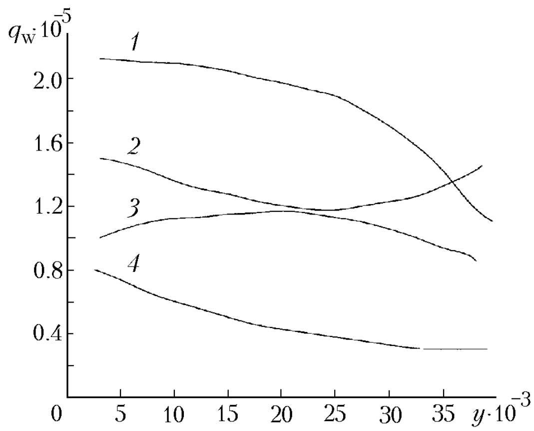
Figure 4 gives the dependences of the dimensionless temperature $\overline{T}$ on the injection parameter $\overline{G}$ ($\overline{T} = T_w/T_e$ and $\overline{G} = (\rho u)/(\rho v)_g$), obtained in testing specimens with different porosities for the same diameter of the pores.

The analysis of the results obtained (Fig. 4) shows that the value of $\overline{T}$ is virtually independent of the shape of the specimen and the porosity in the investigated range of variation of the injection parameters. The dependence $\overline{T} = \overline{T}(\overline{G})$ that is experimental in essence can be described well by a single curve in whose vicinity (its dimensions do not exceed the confidence intervals of measurement) there are all the experimental points of Fig. 4.

The experimental-calculated results obtained are in satisfactory agreement, on the whole, with the basic propositions of the theory of active thermal protection [1, 20] and the ideas of the leading role of the injection parameter in regulation of this process.

In the series of tests of the heatproof material (glass-reinforced plastic based on phenol-formaldehyde binder), the temperature of the exterior plate surface was measured at different distances from the edge, which enabled us to obtain information on the change in the intensity of heat exchange in the curtain zone along the surface. In accordance with the procedure developed, we initially determined the temperature dependences of the thermal-conductivity coefficient of the decomposing glass-reinforced plastic under different conditions of external thermal loading. Figure 5 gives the obtained dependences λ(T). We took λ(T) = 0.35 W/(m·K) ≡ const as the initial approximation [22]. The coefficient of volumetric heat capacity was $\rho c = 1.21 \cdot 10^6$ J/(kg·K) ≡ const.

Figure 6a gives the values of the nonstationary temperature of the surface $T_w^{exp}(t)$ at different distances from the plate's edge $\overline{y}$ = ($\overline{y} = y/L$), whereas Fig. 6b gives the values of the density of the heat flux $q_w(t)$ on the exterior surface along the plate, determined based on solution of problem (7)–(11). It has been noted that at different distances



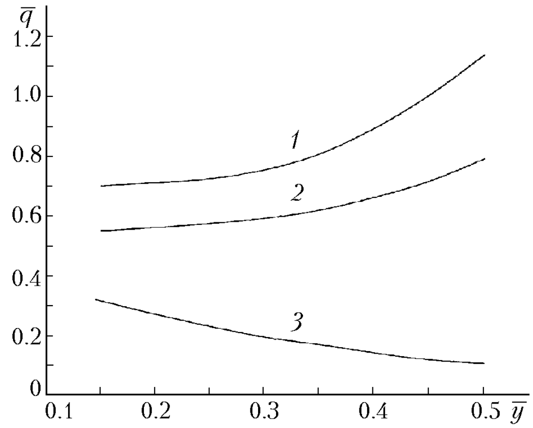


Fig. 7. Heat flux on the surface of the heatproof material for different injection parameters: 1) $G = 0$; 2) $0.17 \cdot 10^{-3}$; 3) $0.5 \cdot 10^{-3}$; 4) $0.8 \cdot 10^{-3}$ kg/sec. q_w, W/m^2; y, 10^{-3} m.

Fig. 8. Reduced heat flux on the surface of a glass-reinforced-plastic plate in the curtain zone: 1) $\bar{G} = 3$; 2) 5; 3) 12.

from the plate's edge, the time dependence of the heat-flux density is characterized by the presence of a maximum at t = 1–3 sec and tends to the quasistationary value with time. A comparison of the results obtained and the calculations carried out in [14] based on the model of a porous reacting body has shown their coincidence with an accuracy of 12%.

The results of the investigation of heat exchange on the surface of a heatproof material with a gas curtain for different values of the mass flow rate of the coolant gas G are given in Fig. 7. As could be expected, the injection of the coolant leads to a reduction in the intensity of heat exchange on portions close to the injection zone; however, the results of the calculations have shown that, when the mass flow rates of the coolant are low (for example, when $G_w = 0.17 \cdot 10^{-3}$ kg/sec), the value of the heat flux $q_w(t)$ on the surface of the element of the heatproof material begins to increase with distance from the injection zone in the presence of injection. At a sufficient distance from the plastic's edge, the injection of the coolant influences only slightly the intensity of heat exchange on the exterior surface of the glass-reinforced plastic (Fig. 7, curves 1 and 2).

Figure 8 gives the values of the reduced heat flux $\bar{q}_w = q_w/q_{m,w}$ ($q_{m,w}$ is the heat flux on the surface in the absence of injection) along the exterior surface of the porous plate for different dimensionless parameters of injection $\bar{G}$. The analysis of the results obtained has shown that for values of the injection parameters $\bar{G} \le 5$ we observe substantial intensification of heat exchange along the plate surface. This effect is attributed mainly to the small length of the curtain zone: the coolant gas injected enhances the intense mixing of the gaseous products of thermochemical decomposition of the binder of the heatproof material and the oxidant-containing incoming flow. This results in an increase in the total rate of physicochemical transformations near the surface. Thus, with distance from the injection zone, the intensity of heat exchange on the destroyed surface is determined primarily by the occurring thermochemical reactions.

When the values of the injection parameter are very high ($\bar{G}$ = 12), we have a total rejection of the high-temperature flowing gas. The contribution of the thermochemical processes to the heat exchange on the exterior surface is not stimulated by the intense inflow of the oxidant because of which the heat flux on the surface of the thermal-protection element is substantially reduced.

The investigations of the thermal state of thermal-protection elements carried out make it possible to draw the following conclusions.

The employment of the methods of inverse heat-conduction problems enables us to determine nonstationary thermal loads on the exterior surface of the elements of active and combined thermal protection in the process of gas-dynamic tests with an accuracy comparable to the error of thermocouple temperature measurements.

The intensity of injection is the governing factor influencing the efficiency of cooling in both active cooling and in the presence of a curtain zone.

NOTATION

x, *y*, space coordinates in the transverse and longitudinal directions respectively, m; *t*, time, sec; *l*, thickness of a porous element, m; *L*, length of a glass-reinforced-plastic plate, m; *D*, diameter of the base of a hemispherical specimen, m; *d*, pore diameter, μm; *T*, temperature, K; λ, thermal-conductivity coefficient, W/(m·K); *c*, specific-heat coefficient, J/(kg·K); ρ, density, kg/m^3; α, heat-exchange coefficient, W/(m^2·K); α_v, coefficient of internal heat exchange, W/(m^3·K); *q*, heat-flux density, W/m^2; μ, coefficient of dynamic viscosity of the coolant gas, kg/(m·sec); *u*, velocity of the incoming flow, m/sec; *v*, linear rate of filtration of the coolant gas, m/sec; t^*, characteristic time of regular heating of the thin-walled sensor, sec; *G*, mass flow rate of the coolant gas, kg/sec; Π, porosity, %; Re, Reynolds number; Bi, Biot number; Fo, Fourier number; *P*, pressure, bar; H_e, enthalpy of the incoming gas, J/kg; M, Mach number. Subscripts and superscripts: s, solid; g, gas; exp, experiment; w, wall; 1, rear wall; 0, initial value; f, final value; st, stationary; v, volume; ef, effective; *p*, at constant pressure; e, external; ∞, 2, and 3, parameters of the gas at infinity, ahead of the shock wave, and in the mixing zone respectively; m, maximum.

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