EXPERIMENTAL STUDY OF TRANSIENT HEAT TRANSFER IN A MAJOR GAS FLOW IN THE PRESENCE OF A TIME-DEPENDENT TEMPERATURE

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On studying transient heat transfer in a major gas flow in the presence of a time-dependent temperature, it is established experimentally that considerable changes take place in the heat-transfer coefficient when $dT_0^*/dt < 0$, where T_0^* is the stagnation temperature of the gas flow.

Transient heat-transfer experiments subject to time-dependent temperatures were carried out in a gas-dynamic installation of the open type with electric-arc heating of the working substance, as described in our earlier paper [1].

The experimental part consists of a cylindrical channel (assembled from 10 individual sections) 25.5 mm in diameter and 10 diameters long. The individual sections are connected together with heat-resistant cement, which, together with the small thickness of the section walls ($\Delta_W \approx 0.08$ mm), greatly reduces axial heat losses. At distances from the entrance corresponding to $x/D = 1.5$, 3.5, 5.5, 7.5, 9.5 are Chromel–Alumel thermocouples 0.1 mm in diameter for measuring the wall temperature. The working parts of the thermoelectrodes, previously rolled to a thickness of 0.02 mm, are welded to the outer surface in the middle of the sections.

This method of determining the temperature of the surface immersed in the gas flow is completely justified in view of the fact that the Bi number, calculated for the experimental conditions envisaged and characterizing the temperature distribution over the thickness of the wall, is much smaller than unity, being equal to ≈ 0.0014.

The gas temperature at the entrance into the experimental section is measured up to 1200°K with a Chromel–Alumel thermocouple and then up to 1900°K with a platinorhodium–platinum thermocouple. The diameters of the thermocouple junctions are no greater than 0.04 mm in either case. These small dimensions ensure almost inertia-free operation.

Figure 1 shows some characteristic temperature/time curves characterizing the wall and main gas flow. Under conditions involving a sharp change in thermal load, if the change in the temperature of the main gas flow from the "hot" to the "cold" state occupies a total of 2.4 sec, the period during which the condition $T_0^* > T_W$ is satisfied amounts to 0.1 sec. Such a short duration of the process naturally necessitates the introduction of special corrections in order to increase the accuracy of the final result. In view of the possible occurrence of errors due to the dynamic characteristics of the diagnostic apparatus in this region the authors restricted consideration to the case in which $T_W > T_0^*$ at this stage of the investigation.

Figure 2 illustrates the evolution of the enthalpy factor in time. The enthalpy/time curve contains an extremum. In the first stage the enthalpy factor increases and the derivative $d\psi_h/dt$ is greater than zero. This situation should correspond to conditions under which the heat-transfer process takes place more intensively than would be the case under steady-state conditions for the same R_h^{**} number. Beyond the extremum the enthalpy factor diminishes, approaching unity asymptotically. According to [2] the change in the sign of the derivative $d\psi_h/dt$ leads to a state of heat transfer in which $St/St_0 < 1$. An analysis of experimental data provides an illustration of this, as may readily be seen from Fig. 2, which is based on local simulation experiments [1]. A combined analysis of the relationships appearing in this figure shows that at the onset of the heat-release process, when the derivative of the temperature of the main gas flow with respect to time is

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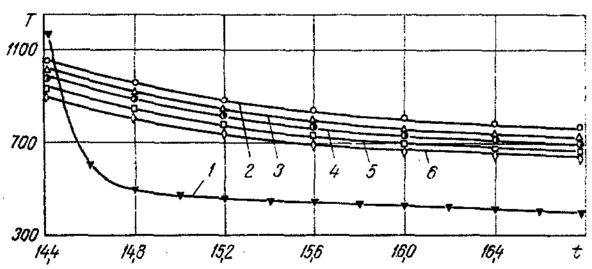

Fig. 1

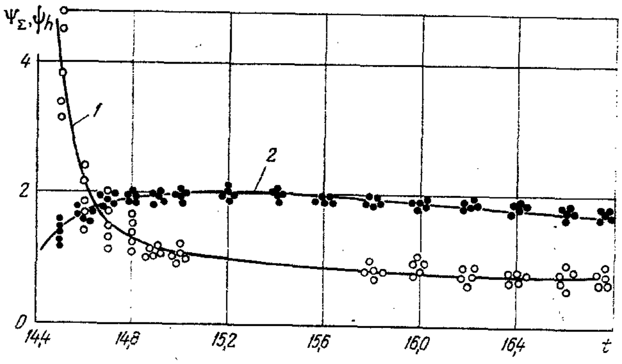

Fig. 2

Fig. 1. Temperatures of the main gas flow (curve 1) and various points on the wall of the experimental section ($\bar{x}$ = 1.5, 3.5, 5.5, 7.5, 9.5 – curves 2, 3, 4, 5, 6, respectively) as functions of time. T, °K; t, sec.

Fig. 2. Time dependence of the enthalpy factor and heat-transfer coefficient for R_h^{**} = idem: 1) $\Psi_\Sigma = \mathrm{St}/\mathrm{St}_0$; 2) $\psi_h = h_w/h_0^*$.

large and positive, the relative heat-transfer coefficient is considerably greater than unity, despite the fact that in the case under consideration the depature from isothermal conditions reduces the heat-transfer coefficient ($\psi_h > 1$; $\Psi_h < 1$). Thus, the prevailing contribution to the Stanton number arises from the time dependence of the temperature of the main flow. At the extremum $d\psi_h/dt = 0$, $\psi_h > 1$, and $\Psi_\Sigma = 1$, i.e., although there is a deviation from isothermal conditions, its effect is compensated by the action of the derivative dT_0^*/dt on the heat transfer. For such a complicated combination of parameters a situation may well occur under which the heat-transfer coefficient will assume a value hardly differing from that corresponding to standard conditions.

Figure 3 shows some experimental data expressed in the form

$$\frac{\mathrm{St}\,\mathrm{Pr}^{0.75}}{\Psi_h} = f\,(R_h^{**}), \tag{1}$$

such that the influence of deviations from isothermal conditions are eliminated. The experimental points on the graph occupy higher positions, the greater the value of the derivative dT_0^*/dt to which they correspond. This change in the heat-transfer coefficient may be explained as being due to two main factors, namely, the sharp dependence of the kinematic and thermal parameters of the system on the gradient of the enthalpy factor and the effect of this factor on the level of turbulent pulsations in the temperature of the main gas flow. In order to reveal the quantitative influence of the enthalpy factor on the relative heat-transfer coefficient, the experimental points of Fig. 3 were analyzed on the basis of the equation

$$\Psi_{Z_h} = \frac{\mathrm{St}}{\mathrm{St}_0\,\Psi_h} = f\left(\frac{dT_0^*}{dt}\right). \tag{2}$$

We found that, independently of position along the length of the initial section and independently of the various degrees of deviation from isothermal conditions, all the experimental points formed a well-defined sequence capable of being approximated by a power polynomial in the form

$$\Psi_{Z_h} = 1 + \left(\frac{dT_0^*}{dt}\right)^{0.13}. \tag{3}$$

Thus, the foregoing investigation shows that thermal transience cannot be characterized simply by the time derivative of the wall temperature. The relative heat-transfer coefficient also depends on the time derivative of the temperature of the main gas flow.

An analysis of the quantitative aspect of the problem indicates that the effect of transience due to the changes taking place in the temperature of the main gas flow is far stronger than the effect due to the changes in wall temperature. Under conditions in which the surface immersed in the gas flow is heated and $dT_0^*/dt = 0$ the transience effect is less than 10% [1]; however, when the surface is cooled and $dT_0^*/dt \neq 0$ the relative heat-transfer coefficient may change by more than a factor of three times.

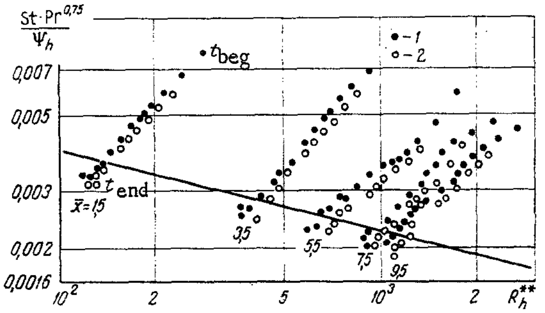

Fig. 3

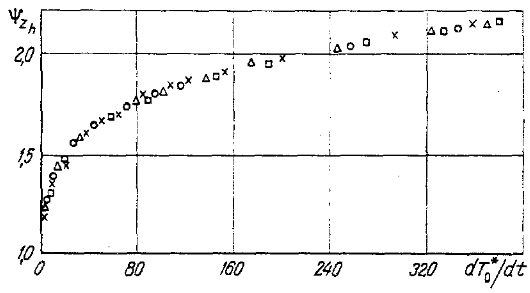

Fig. 4

Fig. 3. Dimensionless heat-transfer coefficient as a function of R_h^{**}. Points, experiment: 1) $\rho_{01}w_{01}$ = 26.9 g/cm^2·sec; 2) $\rho_{01}w_{01}$ = 30.25 g/cm^2·sec; lines, calculation based on the equation $St = 0.0128/R_h^{**0.25}Pr^{0.75}$.

Fig. 4. Influence of dT_0^*/dt (deg/sec) on the relative heat-transfer coefficient.

NOTATION

Bi, Biot number; D, diameter; Pr, Prandtl number; R_h^{**}, characteristic Reynolds number of the thermal boundary layer; St, Stanton number; T_0^*, stagnation temperature of the main gas flow; T_w, wall temperature; t, time; x, longitudinal coordinate; Δ_w, thickness of wall material; ψ_h, enthalpy factor; Ψ_h, parameter representing deviation from isothermal conditions; Ψ_{Z_h}, relative heat-transfer coefficient, allowing for the effect of thermal transience due to the changes in the temperature of the main gas flow; Ψ_Σ, relative change in the Stanton number for R_h^{**} = idem.

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MECHANICS OF JET FLOWS IN GRANULAR LAYERS. EVOLUTION OF SINGLE JETS AND THE NUCLEATION MECHANISM

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A physical analysis of the dynamics of jet development and the formation of a gas bubble in a high fluidized bed is presented.

Fluidized or fixed granular layers with jet delivery of excess liquefier are so widespread that their applied value needs no recommendations. Let us note just two main types of jet flows which differ in principle. First, the whole gas can be supplied as a jet in an initially fixed layer with the possible build-up of a fluidized state, starting from some level above the gas-distributor grating. Secondly, jets can be introduced into an already fluidized or almost fluidized layer to improve the quality of the fluidization or to intensify the exchange processes. In both cases, both the characteristics of the individual jets and their interaction, which mainly govern the layer structure observed, are of primary value.

The physical picture of jet development in a fluidized bed was described phenomenologically in [1-3], for instance. In particular, stationary and self-oscillating escape modes and an intermediate mode of local spouting

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