HEAT TRANSFER IN AERODYNAMIC-TUBE NOZZLES WITH HIGH-TEMPERATURE GAS FLOW

N. I. Khvostov, V. E. Chekalin,
A. D. Sukhobokov, and K. N. Skirda

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A method of analysis of the heat transfer in nozzles with a turbulent flow of air at temperatures up to 3800°K is given, together with experimental data. The results are generalized in terms of the Nusselt and Stanton numbers. It is shown that the heat-transfer coefficients are lower for flow in a nozzle than for flow at a plate.

In [1] it was shown that in determining the heat fluxes in nozzles, the results depend significantly on how the hypothesis of turbulent friction is introduced into the calculation. The calculated heat fluxes may differ by more than a factor of 3. This complicates the design of cooling systems and the calculation of nozzle strength at high gas temperature and pressure in the prechamber. In [2, 3], results on turbulent friction were generalized for flow along a plate without a pressure gradient, which does not correspond to the flow conditions in a nozzle. Heat transfer in aerodynamic-tube nozzles was investigated experimentally in [4], which gives results obtained at temperatures much lower than those required in the production of hypersonic aerodynamic tubes. To allow results to be obtained at high temperatures, a method of determining heat-transfer coefficients from the measured nonsteady temperature field at the nozzle wall was developed in an earlier work [5]; the present work gives results obtained using this method.

The copper experimental nozzle (with a critical cross section of diameter $d_* = 3.12$ mm) had an inlet-section radius of curvature of 30 mm, and a cylindrical section of length 10 mm in the region of the critical cross section. The supersonic part of the nozzle was conical in form, with a vertex angle of 8°; the nozzle outlet diameter was 45 mm. By means of deep cuts of width $\Delta h = 3$ mm extending inward from the outer contour, the nozzle was divided into bands. The first 10 bands, counting from the inlet, were of width 8 mm, while for the next seven, $h = 16$ mm; the outer diameter of the bands was 54 mm. The massive copper bands collected the heat transmitted to the nozzle wall by the flow of hot gas. The annular cuts served to eliminate heat flow along the nozzle contour. In addition, in the region of the thin bridge pieces between the bands, the wall temperature was higher than in the region of the bands, which also prevented heat flow from one band to another, while retaining a smooth nozzle surface in contact with the gas. Thus, each band was regarded as a heat-insulated calorimeter, in which the inner surface of the annular band formed the heat-transfer surface.

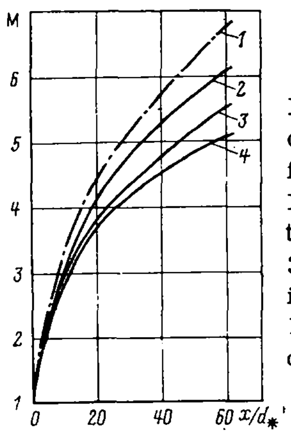

Fig. 1. Mach-number distribution over the experimental nozzle: 1) perfect gas with a specific-heat ratio $\varkappa = 1.4$; 2) $T_0 = 1500$°K, $P_0 = 3.6 \cdot 10^5$ Pa, taking account of viscosity; 3) $T_0 = 3800$°K, $P_0 = 20 \cdot 10^5$ Pa, without taking account of viscosity; 4) $T_0 = 3800$°K, $P_0 = 20 \cdot 10^5$ Pa, taking account of viscosity.

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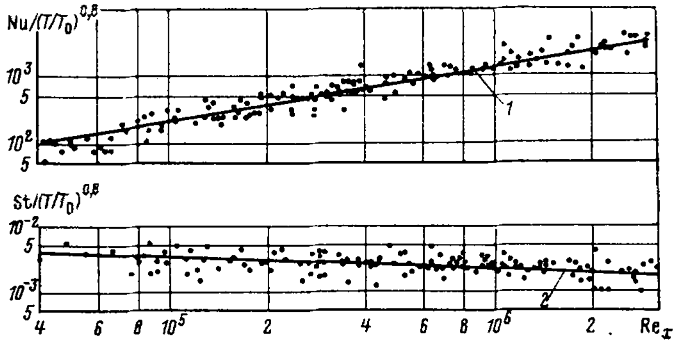

Fig. 2. Dependence of dimensionless heat-transfer coefficients on Reynolds number: 1) $Nu/(T/T_0)^{0.8}$; 2) $St/(T/T_0)^{0.8}$.

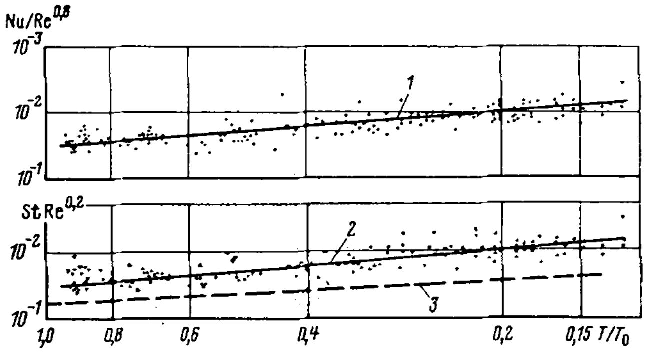

Fig. 3. Dependence of dimensionless heat-transfer coefficients on T/T_0: 1) $Nu/Re^{0.8}$; 2) $St \cdot Re^{0.2}$; 3) calculation by the method of [2].

The change in temperature of each band was determined from the readings of Chromel—Copel thermocouples pressed into the massive bands using copper sleeves, positioned 3-6 mm from the internal nozzle surface. The depth of material below the thermocouple was determined with an accuracy of 0.01 mm.

The gas was heated by an electrical-arc heater with a prechamber diameter of 70 mm. The time for the heater to reach steady pressure and temperature conditions was 3-5 sec. In the course of the experiment, the pressure P_0 in the prechamber and the nozzle wall temperature were measured as a function of the time, using oscillographs. The air temperature in the prechamber T_0 was determined from the ratio P_0/P_{0x} on the basis of the data in [6]. In the experiments the air temperature varied from 1400 to 3840°K, and the pressure from $5 \cdot 10^5$ to $72 \cdot 10^5$ Pa; the duration of the experiments was in the range 4-60 sec.

To find the heat flux and the heat-transfer coefficients, the one-dimensional heat-conduction equation $\partial t/\partial \tau = a[\partial^2 t/\partial r^2 + (1/r)\partial t/\partial r]$ was solved, with the following conditions: when $r = R_1$, $\alpha(T_{ad} - T_w) = -\lambda \partial t/\partial r$; when $r = R_2$, $\partial t/\partial r = 0$; and when $\tau = 0$, $t = \text{const}$. The equation was solved by the elementary-balance method on a BÉSM-6 computer.

For given α and T_{ad}, the calorimeter temperature at the thermocouple site was calculated for $\tau = 1$ sec. When the calculated and measured temperatures agreed, the specific heat flux $q_1 = \alpha(T_{ad} - T_w)$ was taken to be equal to the mean specific heat flux at time $\tau = 1$ sec, and the resulting temperature distribution over the calorimeter was adopted as the initial distribution for the determination of q_1 at $\tau = 2$ sec, and so on.

If the temperatures did not agree, a correction was introduced in the heat-transfer coefficient, and the calculation repeated. Agreement of the temperatures with an accuracy of 0.05° required calculations of no more than four approximations, which involved about 40 sec of machine time.

Since the calorimetric element had a cut of depth $R_2 - R_3$, a correction was introduced into the final result of the calculation, and the value of q was determined from the formula $q = q_1 k$, where $k = h/H[1 + (\Delta h/h) \times (R_3^2 - R_1^2)/(R_2^2 - R_1^2)]$, $H = h + \Delta h$. The meaning of the correction is that the heat flux enters the calorimeter over a surface proportional to H, while the increase in temperature is proportional to its mass, which is

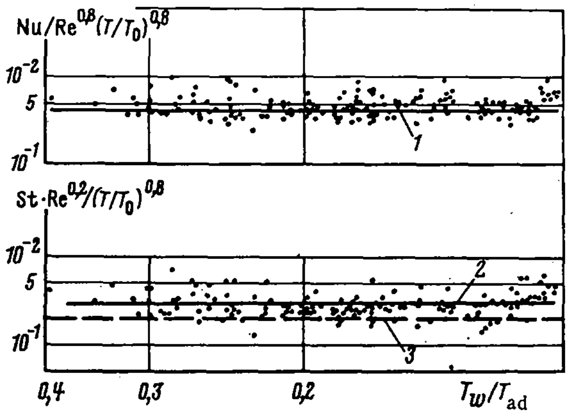

Fig. 4. Dependence of dimensionless heat-transfer coefficients on the temperature factor: 1) $\mathrm{Nu}/(\mathrm{Re}T/T_0)^{0.8}$; 2) $\mathrm{St}\cdot\mathrm{Re}^{0.2}/(T/T_0)^{0.8}$; 3) calculation by the method of [3].

determined mainly by the height h of the band. For the colorimeter dimensions chosen, the second term in brackets is of magnitude 0.03-0.06.

For Mach numbers above 4, the flow in the nozzle is significantly influenced by the boundary layer, the displacement thickness of which depends on the Reynolds number and wall temperature. The Mach number in the nozzle may differ considerably from the Mach number determined from the geometric relations of the nozzle. The displacement thickness was determined from the formula of [7], reduced using the data of [8] for a turbulent boundary layer to the form

$$\delta^*/x = (0.011 + 0.0065T_w/T_{ad})\,\mathrm{M}^{1.5}/\mathrm{Re}^{0.2},$$

where δ^* is the displacement thickness of the boundary layer at a wall temperature T_w; x is the distance from the critical cross section. In the case of a laminar boundary layer the analogous relation takes the form

$$\delta^*/x = (0.179 + 0.292T_w/T_{ad})\,\mathrm{M}^{1.8}\mathrm{Re}^{0.5}.$$

When $M > 4$, the method of successive approximation was used to determine the Mach number in the cross section including the thermocouple site on the basis of the above relations and the known nozzle shape.

The effect of the real properties of air on the gasdynamic flow parameters in the nozzle was taken into account according to the data of [6]. The results are shown in Fig. 1 as a function of x/d_*. The temperature in the prechamber and the viscosity evidently have a significant effect on the Mach-number distribution over the nozzle.

Experimental data on heat transfer are analyzed in the form of curves of the Nusselt number $\mathrm{Nu} = qx/(T_{ad} - T_w)\lambda$ and the Stanton number $\mathrm{St} = q/(I_{ad} - I_w)\rho u$ on the Reynolds number, the temperature factor T_w/T_{ad}, and T/T_0. For a perfect gas, the parameter T/T_0 is expressed analytically in terms of the Mach number in the form $T/T_0 = [1 + (\varkappa - 1)/2M^2]^{-1}$, where $\varkappa$ is the specific-heat ratio. For real gases the dependence of T/T_0 on the Mach number was tabulated in [6].

At a temperature $T_0 > 2500$°K, air begins to dissociate, and its thermophysical properties depend in a complex manner on the temperature and pressure. In the experiments, the maximum degree of dissociation in the prechamber was 0.166. With expansion of gas in the nozzle, the degree of dissociation decreased. In the boundary layer at large Mach numbers, the static temperature is much less than the adiabatic stagnation temperature. Therefore, despite the decrease in pressure, the degree of dissociation cannot be larger in the boundary layer than in the prechamber. According to [2], the maximum expected effect of dissociation on the heat transfer should be no more than 4%.

Dependences of the following form were investigated as generalizations of the experimental results

$$\mathrm{Nu} = A\,\mathrm{Re}^{m_1}(T/T_0)^{m_2}(T_w/T_{ad})^{m_3},$$
$$\mathrm{St} = B\,\mathrm{Re}^{n_1}(T/T_0)^{n_2}(T_w/T_{ad})^{n_3}.$$

The Nusselt and Reynolds numbers may be determined by various means, depending on the conditions under which the heat conduction and viscosity are chosen. The experimental results were generalized using the heat conduction and viscosity determined from the materials of [9] at the temperature and pressure in the prechamber, and in the static pressure and adiabatic stagnation temperature T_{ad} of air. The best generalization was that obtained in determining the heat conduction at T_{ad} from the curve which is the envelope of the minimum values of the heat conduction given in [9]. These results are shown in Figs. 2-4.

In Fig. 2, the dependence of $Nu/(T/T_0)^{0.8}$ and $St/(T/T_0)^{0.8}$ on the Reynolds number is shown. The experimental results indicate that in the range of Reynolds numbers investigated heat transfer occurs during turbulent flow conditions. The lines shown in Fig. 2 correspond to $m_1 = 0.8$ and $n_1 = -0.2$.

The scatter of the experimental data depends mainly on the accuracy of determining and maintaining the constant mean-mass temperature of the gas in the electric-arc heater, the time for the heater to reach steady conditions, and the accuracy of taking and recording readings from the oscillogram.

In Fig. 3, $Nu/Re^{0.8}$ and $St \cdot Re^{0.2}$ are shown as a function of T/T_0. Introducing the parameter T/T_0 allows data obtained at different Mach numbers to be generalized. The lines in Fig. 3 correspond to $m_2 = n_2 = 0.8$. In determining λ from the local values of temperature and pressure, $m_2 = 0.5$. Generalizing the results with respect to I/I_{ad} does not lead to any narrowing of the spread of experimental values of St. Also in Fig. 3, the results obtained using the formulas of [2] derived for the heat transfer to a plate in a real gas are shown. Comparison shows that the calculated curve is in qualitative agreement with the experimental data, while the heat-transfer coefficients in the nozzle are less than for the corresponding conditions at a plate by a factor of approximately 1.6-2.0.

In Fig. 4, the dependence of $Nu/(ReT/T_0)^{0.8}$ and $St \cdot Re^{0.2}/(T/T_0)^{0.8}$ on the temperature factor T_w/T_{ad} is shown, together with the data of [4] obtained at $T_0 = 673$ and 973°K by an independent method. It is evident that the range of the investigations with respect to the temperature factor is considerably broader than in [4]. On the basis of the combined results, it may be asserted that, for an analysis in the given terms, the temperature factor has no effect on the heat transfer in aerodynamic nozzles, or that its effect, in turbulent flow, does not exceed the scatter of the experimental results. This significantly improves the accuracy of heat calculations for aerodynamic-tube nozzles at high gas temperatures.

Using the Reynolds analogy, in the form of the dependence $St = Pr^{-2/3}C_f/2$, where C_f is the friction coefficient and Pr is the Prandtl number, Fig. 4 shows the results of the calculation in the case when the local coefficient of friction is determined by the method of [3] for supersonic flow along a plate. The calculated curve of the Stanton number as a function of the temperature factor is in qualitative agreement with the experimental data and, numerically, is larger by a factor of 1.4-1.8.

The experimental results obtained for the heat transfer in nozzles with turbulent flow, at temperatures $T_0 \le 4000$°K, Reynolds numbers $5 \cdot 10^4 - 10^7$, and $T_w/T_{ad} = 0.1-1.0$, are generalized by the following dependences

$$Nu = 0.022\,Re^{0.8}\,(T/T_0)^{0.8},$$

$$St = 0.035\,Re^{-0.2}\,(T/T_0).$$

In the calculations it should be remembered that the appearance of turbulent flow in aerodynamic-tube nozzles may be deferred by special means [7] to Reynolds numbers of $(1-1.5) \cdot 10^6$.

NOTATION

P_0 and T_0, gas pressure and temperature in prechamber; t, calorimeter temperature; τ, time; r, radius; a, α, λ, μ, thermal diffusivity, heat-transfer coefficient, heat conduction, and viscosity, respectively; q, specific heat flux; T_{ad} and I_{ad}, temperature and enthalpy of adiabatically stagnant gas; I, I_w, T, T_w, enthalpy and temperature of gas in flow core and at wall temperature; M, Nu, St, Re, Mach, Nusselt, Stanton, and Reynolds numbers, respectively.

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FLOW OF VISCOUS INCOMPRESSIBLE LIQUID IN A PLANE CHANNEL WITH ABRUPT ONE-SIDED BROADENING

V. I. Korobko, É. M. Malaya, and V. K. Shashmin

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A numerical solution is obtained for the Navier—Stokes equations in the problem of laminar flow of a viscous incompressible liquid in a plane channel with abrupt one-sided broadening. The solution is compared with experiment.

The flow of a viscous incompressible liquid in a channel with abrupt broadening is of great practical interest. Such flows are investigated on the basis of the complete Navier—Stokes equations, since viscosity effects play a large role. Abrupt broadening of the channel is associated with breakaway of the flow from the wall and the formation of a region of return flow.

The laminar flow of viscous incompressible liquid in channels with abrupt broadening has been studied both numerically and experimentally. In [1], numerical calculations were made of the flow in an abruptly broadening and abruptly narrowing channel, with displacement of one of its walls, at Reynolds numbers $Re \le 1000$, and the dependence of the breakaway-region length on Re was obtained for a given broadening. In [2] the periodic flow in a channel with abrupt broadening was investigated numerically for $Re \le 200$. The difficulty in experimental investigations of such flows is to measure the velocities in the return-flow region, which are very small. Electrothermoanemometers and laser anemometers are used for this purpose. In [3-5], the results of laminar-flow-velocity measurements in channels were given, and in [6] the velocity field in the recirculation region was measured for abrupt broadening of the channel. In [7] the development of an unbounded flow along a wall with a step was investigated; the effect of the boundary-layer thickness at the step on the length of the recirculation region was found to be small [7]. In [8] the velocity field was measured for a laminar field of viscous incompressible liquid in a plane channel with one-sided broadening ($H/h_1 = 1.5$; $Re = u_0h_1/\nu = 146, 250, 382, 458$). The numerical results of [9] show that the recirculation region is longer than in [8], which is a consequence of the form of the initial velocity profile; this effect was not investigated experimentally. Analogous conclusions were reached in [10, 11] in an experimental investigation of the development of a flow of viscous incompressible liquid in channels with "ribs." In [12], the motion of an unbounded flow at a step was investigated numerically at $Re \ge 10^4$. Numerical calculation of the flow in a channel with a step [13] ($H/h_1 = 1.1$) showed that the length of the return-flow region is independent of Re ($25 \le Re \le 100$). In [14], the results of a numerical calculation of the flow in a channel with abrupt broadening ($H/h_1 = 8$) are given and, in particular, the dependence of the length of the return-flow region on Re ($Re \le 120$) was obtained. In [15], an analogous investigation was made for $H/h_1 = 1.5$ and $Re \le 40$. Numerical methods of solving Navier—Stokes equations for use with the given type of flow are reviewed in [16-18]. The present work gives the results of a numerical calculation of a laminar flow of viscous incompressible liquid in a channel with abrupt one-sided broadening ($H/h_1 = 3.2, 5.2, 6.2, 7.4, 8.2$). The calculation is made on the basis of the complete Navier—Stokes equations for $Re = 180$.

The system of Navier—Stokes equations describing the flow of viscous incompressible liquid in a plane channel is of the form [16]

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