EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER IN TURBULENT FLOW OF AIR IN A CIRCULAR TUBE WITH LARGE TEMPERATURE DROP AND COOLING OF THE AIR

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An experimental investigation has been made of the influence of the temperature factor on the heat transfer coefficient in cooling of air in a circular tube. It has been shown that in the developed-turbulence section the heat transfer coefficient does not depend on the value of the temperature factor, when the latter is decreased to 0.12.

The problem of heat transfer in flow of a gas in a tube with large temperature differences, in the cooling case, is very topical due to the development of various thermal engineering facilities operating in the presence of large heat fluxes. These problems have not yet been completely solved, neither on the theoretical side nor experimentally.

The difficulties of a theoretical examination of flow with large temperature difference arises from the need to take into account the dependence of the physical properties of the gas on temperature, which leads to considerable complication in the calculations and to a series of crude approximations. At present there are several theoretical investigations [1-3], indicating different degrees of influence of the temperature factor on the heat transfer coefficient in the cooling case. Experimental investigations of heat transfer in gas flow with large temperature changes in the cooling case are also few in number [4-6] and have been carried out in a comparatively narrow range of variation of the temperature factor. All the experimental investigations indicate that the temperature factor has no effect on heat transfer, which contradicts the results of theoretical investigations.

In this paper we present an experimental investigation of heat transfer in turbulent gas flow in a circular tube far from the entrance, with greater temperature differences than in [4-6].

1. The experiments were carried out on apparatus schematically similar to that given in [7]. The main elements of the equipment (Fig. 1) are an electric-arc air-heater 10, a forward chamber 11, a measuring tube 12, and an aft chamber 13. Air, previously heated by compressor 1, is supplied from a bank of gas-bottles 2, at a pressure of up to 7 atm, through filter 3 to a pressure regulator 4, which maintains an accurately constant pressure of 4 atm in the air supply to the equipment. In addition, the air circuit also has a receiver 5 of volume 3 m^3, a resistance thermometer 6, and a control valve 8. The air flow rate is measured in two ways: from the pressure drop in a standard measuring orifice 7, and by measurement of the total pressure in front of a critical mass-flow nozzle 9. The data from the two measurements were in close agreement. Heating of the air was carried out in an electric-arc air-heater with vortex arc stabilization and with water-cooled steel electrodes [8]; the electrical supply for the heater came from a constant current generator, and was triggered by a high-voltage oscillator through an intermediate electrode 17. The electric-arc circuit contained a series-connected ballast tubular rheostat with water cooling. With an air flow of 30 g/sec and an electric arc power of about 240 kW, the heater achieved an airstream temperature at the entrance to the aft chamber of 3400-4500°K. At this condition the flow contamination by electrode breakdown products was very small, not exceeding 0.1% of the air flow rate. The forward chamber, mounted after the heater and having cooled walls, promotes more uniform distribution of the air stream

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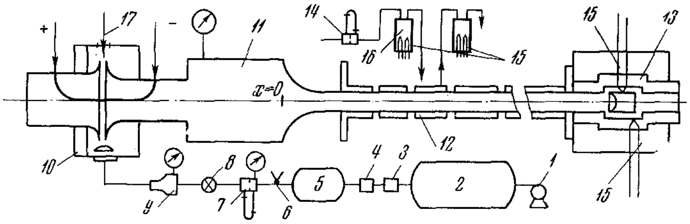

Fig. 1

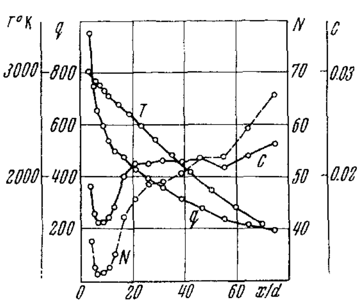

Fig. 2

Fig. 3

parameters before the stream enters the measuring tube. The air pressure in the forward chamber does not exceed 0.3 atm. The pressure oscillations, associated with instability of the heater electric arc, as has been demonstrated experimentally [8], have such a high frequency that they do not affect the measured parameters, because of the substantial inertia of the chosen method of heat transfer measurement.

The heat transfer to the tube wall was determined by a calorimetric method, based on measurement of the rate of increase of the heat content of water washing a specific section of the measuring tube.* A measuring tube of 1Cr18Ni9 with inside diameter of 32 mm, length of 2450 mm, and wall thickness of 2 mm, was divided by annular cuts into 15 unequal sections. Each section was covered on the outside by an individual cylindrical sleeve through which the cooling water flowed. Thus, the measuring tube has 15 separate sections, calorimeters, located in sequence along its axis. The length of each section was chosen to achieve equality of the total amount of heat received by the water in each section with equal flow rates and temperature drops in the water, and also taking into account the heat flux density along the tube length, as obtained on a similar facility [7].

At the locations of the annular cuts between sections, the tube wall thickness was 1 mm, which reduces the heat flow between sections. The length of the first section was 40 mm, and that of the last was 335 mm. The outer surface of the measuring tube was thermally insulated from its surroundings by a layer of asbestos.

To measure the air temperature at the exit from the measuring tube an aft chamber was used in which there was a multichannel ceramic mixer. In the mixer the air stream was distributed among four channels whose axes were displaced parallel to the axis of the measuring tube; then they were combined again in a discharge tube, which led to a system of exhaust fans. In the mixer there were two platinum/platinum-rhodium thermocouples, the sensor of one of which is in the air stream and the other in the ceramic wall of the mixer. After the facility has been brought to steady conditions, the readings of the two thermocouples were in agreement, which indicates that the flow temperature distribution is uniform over the transverse section of the mixer channels. Thus, the temperature measured by the thermocouple mounted in the aft chamber corresponds to the mean-mass temperature of the air at the exit of the measuring tube.

Cooling of the air-heater, the forward chamber, and the measuring tube was done with water, directly from the mains. The water flow rate through each tube section was measured with a differential U-tube manometer, from the pressure drop at a measuring orifice-plate 14. All the differential manometers were located on a vertical panel and their readings were photographed. The temperature difference of the water between the entrance and exit of each tube section was measured by four-junction differential chromel/kopel thermocouples 15, located in the water mixers 16, and recorded by electronic self-balancing

*In general, this method allows determination of the average heat transfer along the tube. However, at high heat fluxes, when the extent of the section can be made fairly small, at constant tube wall temperature, and for a symmetrical distribution of the parameters relative to the tube longitudinal axis, the calorimetric method can be used to determine local heat transfer.

chart potentiometers. Measurements of all the parameters were taken after the equipment had come to steady conditions.

2. The processing of the experimental data was performed under the assumption of one-dimensional flow conditions (i.e., the flow parameters vary only along the tube axis). At each section the mean heat flux density to the inner tube surface was determined from the formula

$$q = cG\Delta t/\pi dl$$

Here c, G, and Δt are the specific heat capacity, the flow rate, and the temperature difference between the entrance and exit of the section; d is the tube inner diameter; and l is the length of the section in the axial direction.

Because of the rather high thermal conductivity of 1Cr18Ni9 steel and the small tube wall thickness, the temperature of the inner tube surface (T_W) was very little different from that of the cooling water. Therefore, the wall temperature was assumed constant and equal to 300°K.

The distribution of the mean-mass stream enthalpy along the tube axis was determined by the heat balance of the corresponding sections of the tube and the temperature of the air measured in the aft chamber. At the tube cross section corresponding to the beginning of each section, the mean-mass enthalpy was determined from the formula

$$i = i_{-} + Q/G_{+}$$

Here i_{-} is the enthalpy of the air at atmospheric pressure, corresponding to the temperature in the aft chamber; Q is the heat flux to the tube wall in the section between the cross section considered and the aft chamber; G_{+} is the mass flow rate of air. From the mean-mass enthalpy distribution, the mean-mass temperature distribution (T) was determined. Values of the thermophysical constants of air at the various temperatures were determined from the data of [9, 10]. The mean Nusselt (N) and Reynolds (R) numbers for each tube section were calculated from the formulas

$$N = \frac{qd}{(T - T_w)\lambda}, \qquad R = \frac{4G_{+}}{\pi d\mu}$$

Here T is the mean-mass flow temperature at the middle of the section; T_W is the wall temperature; and λ, μ are the thermal conductivity and dynamic viscosity of air at temperature G.

The Prandtl number (P) was assumed constant and equal to 0.74. The distribution of the flow parameters along the tube axis for one experiment is shown in Fig. 2. The origin of the longitudinal coordinate x is shown in Fig. 1.

The smooth nature of the heat flux density distribution along the tube indicates a high accuracy in measurement of heat flux. At the entrance to the measuring tube the mean-mass temperature reached 3000°K. The calculations performed showed that the mean flow velocity over the cross section at the beginning of the tube was 340 m/sec, and the Mach number was 0.34. At the end of the tube the velocity decreased to 170 m/sec and the Mach number to 0.22. Thus, the flow in the tube was at low subsonic speed and compressibility effects can be neglected.

The nature of the Nusselt-number distribution indicates that at the initial section of the tube the boundary layer is laminar and then undergoes transition to turbulence (for $x/d > 20$). The experiments performed cover a narrow range of Reynolds number (from $1.5 \cdot 10^4$ to $2.4 \cdot 10^4$), with an appreciable change of the temperature factor. A correlation of the experimental data was performed using the well-known parametric equation for developed turbulent flow

$$N = CR^{0.8}P^{0.4}$$

The distribution of the coefficient C with tube length is also shown in Fig. 2.

Figure 3 shows experimental values of C as a function of the inverse of the temperature factor for a section of fully-developed turbulent flow in the tube ($x/d > 20$). For a variation of the ratio T/T_W from 4.7 to 8.5 the average value of C remains constant and equal to 0.022. The scatter in the experimental points does not exceed ±7% of the reading. Thus, the experimental investigation made indicates that for fully developed turbulent gas flow cooled in a tube (R = 17000–24000), the heat transfer coefficient does not depend on the temperature factor when the latter is varied up to a value of 0.12, and can be determined by the empirical equation $N = 0.022 R^{0.8} P^{0.4}$. The results obtained agree with those of [6] for smaller temperature differences.

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