INVESTIGATION OF HEAT TRANSFER AND AERODYNAMIC RESISTANCE OF TUBULAR HEAT-TRANSFER SURFACES IN A WIDE RANGE OF VALUES OF THE REYNOLDS NUMBER

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The greater portion of investigations of heat transfer and aerodynamic resistance of tube bundles in a transverse gas stream is limited to a comparatively narrow range of values of the Reynolds number (Re < 2 \cdot 10^4). In this article the results are presented of an investigation in the area Re = 7.5 \cdot 10^3 - 3 \cdot 10^5.

A diagram of the experimental setup and the position of the measuring devices is shown in Fig. 1. An aerodynamic tube of the open type was used for the investigation. The bundle had eight rows of tubes in the direction of the air stream and six at right angles to the stream. The diameter of the tubes d was 25 mm, the transverse pitch S_1 = 1.32 d, the longitudinal S_2 = 1.5 d. Identical heat-transfer conditions in all the tubes of the bundle were achieved by means of regulation of the water flow rate in each longitudinal row. The middle row of tubes was calorimetric. For avoidance of heat losses to the ambient medium the bundle was insulated with pseudo pipe boards made of Textolite and with asbestos.

Fig. 1. Line diagram of the experimental setup: 1) water pump; 2) measuring tank; 3) measuring section; 4) Venturi nozzle; 5) Prandtl tube; 6) laboratory thermometer; 7) coordinator; 8) hyperthermocouple; 9) calorimeter; 10) water distributing collector; 11) operating section; 12) electric heater; 13) autotransformer; 14) regulating gate valve; 15) air compressor; 16) heating tank.
Fig. 2. Plot of the heat transfer $Nu$ of a smooth tube, eight row staggered bundle versus the $Re$ number: 1) $Nu = 0.296 Re^{0.6}$ ($Re = 10^4 - 10^5$), $Nu = 0.0108 Re^{0.886}$ ($Re = 10^5 - 3 \cdot 10^6$); 2) $Nu = 0.270 Re^{0.6}$ ($Re = 10^4 - 2 \cdot 10^5$), $Nu = 0.008 Re^{0.886}$ ($Re = 2 \cdot 10^5 - 3.2 \cdot 10^6$); 3) $Nu = 0.248 Re^{0.6}$ ($Re = 10^4 - 10^5$), $Nu = 0.0048 Re^{0.94}$ ($Re = 10^5 - 5 \cdot 10^5$) [1]; 4) $Nu = 0.26 Re^{0.6}$ ($Re = 6 \cdot 10^3 - 6 \cdot 10^4$) [2]; O) air heated; ●) air cooled.

Fig. 3. Plot of the aerodynamic resistance $Eu$ of a smooth tube, eight row staggered bundle versus the $Re$ number: O) air heated; ●) air cooled.

Owing to the presence in the circuit of a five-section electric heater and the LATR-1 autotransformer the water temperature was controlled in the range 5-100°C, which permitted studying the effect of the direction of the heat flux on the heat transfer and aerodynamic resistance.

The dynamic and static heads were measured with Prandtl tubes. Measurement of the static pressures was duplicated by annular chambers. Stabilization of the stream flowing into the bundle was carried out in the measuring section. The velocity fields of the incoming stream was measured at 16 points of the measuring section. The field coefficient $\phi$ was practically independent of $Re$ and was equal to 0.96. The static pressure drops in the tube bundle were measured with Prandtl tubes. The Prandtl tube was placed beyond the tube bundle at some distance from the operating section for the purpose of stabilizing the air stream coming out of the bundle.

The air temperature was measured with 15-junction copper-constantan hyperthermocouples (diameter 0.16 mm) and duplicated by laboratory thermometers having 0.1°C scale divisions and placed in the calorimeters. The water flow rate was determined by a water meter with a vane rotor and a measuring tank calibrated by a gravimetric method. The wall temperatures of the tubes were measured with copper-constantan thermocouples (diameter 0.16 mm). The thermocouples, slipped into a tube (diameter 0.8 mm) are rolled into a milled groove, and the thermocouple junctions are soldered to the wall.

The experimental data were interpreted with the aid of the dimensionless equations

$$Nu = f_1(Re); \quad Eu = f_2(Re).$$

The physical constants were referred to the mean air temperature and to the wall temperature.

The heat-transfer coefficient of the air $\alpha_g$ was determined from the formula

$$\alpha_g = \frac{Q_w}{F \Delta t_1},$$

where $Q_w$ is the quantity of heat either given up or received by the water in the bundle, in watts; $F$ is the whole external surface of the bundle in m²; $\Delta t_1$ is the mean logarithmic temperature head in °C.

The rates entering into the $Re$ and $Eu$ numbers were taken in a constricted cross section of the bundle.

The authenticity of the experimental heat-transfer data was checked by comparison with published data [1] and [2]. The results of experiments on heat transfer and the data of [1] and [2] are presented in Fig. 2. The data of [2] were obtained as a result of the recalculation of the generalized equation $Nu = 0.27c_2 Re^{0.6}$ (here $c_2$ is a correction for the number of rows) applicable to the bundle studied. The data presented in Fig. 2 were obtained by referring the physical constants to the mean temperature of the stream.

The scatter of the experimental points in Fig. 2 shows that the use of the mean stream temperature $t_1$ as a determinant does not permit description of the processes of heating and cooling of the air by the general dimensionless equation. In the region $Re \leq 10^5$ the heat transfer during heating exceeds the heat transfer during cooling by 10%, and with the onset of an auto-simulation regime this difference increases, reaching 35%.