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EXPERIMENTAL INVESTIGATION OF MIXED AIR CONVECTION NEAR A HORIZONTAL CYLINDER

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Heat exchange with mixed convection near a horizontal cylinder plays an important role in a number of technological processes. In addition, a cylinder is a convenient model for a fundamental investigation of the process. Several reports on this problem have now been published. The boundary-layer equations, written in the Boussinesq approximation, have been solved numerically for the region of a cylinder where the use of boundary-layer theory is possible [1-3]. The numerical investigation was carried out most fully in [1], where, along with results on heat exchange and friction, data were obtained on the influence of gravitational forces on the separation of the boundary layer. In all the reports the velocity distribution at the outer limit of the boundary layer is taken either from experimental data for purely forced motion or as for streamline flow of an ideal fluid, since there are no data for mixed convection. The average heat transfer at a constant wall temperature or at a constant heat flux is mainly considered in the experimental reports [4-7]. Only in [7] is the local heat exchange of a horizontal cylinder with a constant heat flux investigated for transverse flow over it. Data are absent on the hydrodynamic environment over the entire perimeter of a cylinder under conditions of mixed convection.

In the present report an experimental investigation is made of the flow of a vertical air stream over a horizontal isothermal cylinder when the directions of forced motion and the gravitational forces do and do not coincide. The influence of natural convection on the position of the separation point of the boundary layer is investigated. The velocity and temperature distributions are measured. The local and average heat fluxes are determined. The measurements are made at $Gr \approx 10^5$, $Re = 40\text{-}4000$, and $Gr/Re^2 = 0.01\text{-}20$.

The investigations were conducted in the working chamber of a vertical low-velocity wind tunnel which could operate in closed and open schemes. The stream velocity was varied in the range of 0-1 m/sec and the stream temperature was varied from 20 to 50°C. The degree of stream turbulence in the working chamber did not exceed 0.3%. A cylinder made of copper pipe 60 mm in diameter and 200 mm long was used as the working body. The degree of blockage of the stream by the cylinder was 0.12. As is known, such a level of turbulence and stream blockage does not affect the heat exchange for laminar flow over a cylinder. The cylinder was cooled or heated, depending on the required direction of natural convection.

The separation point of the boundary layer was determined through visualization of the flow by the method of a laser light "knife." A thin streamer of tobacoo smoke, which moved along a streamline in the boundary layer, was supplied in the plane of the "knife" in the vicinities of the upper or lower critical points of the cylinder. The separation point was accurately determined visually and from photographs from the sharp change in the direction of motion of the smoke streamer. The accuracy of determination of the angular coordinate of the separation point was no worse than 2° The velocity was measured with a laser anemometer of type 55L from

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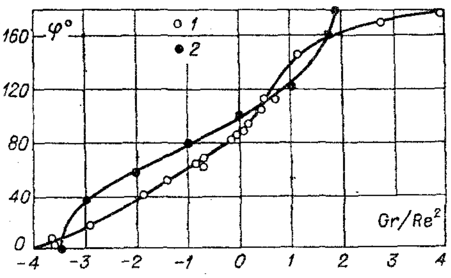

Fig. 1

the DISA Company. The measurement was made in the differential Doppler mode with direct reflection. The measurement volume had a diameter of 0.235 mm. As the light-scattering particles we used wood smoke with particle sizes of 0.4–1 μm. The position of the measurement volume was determined with an error of ~0.05 mm using a special coordinating device. The change in the position of the center of the measurement volume due to the temperature gradient in the boundary layer when max $\Delta t = 50°C$ was ±3% of the value of its diameter.

The local coefficient of heat transfer was determined from the temperature profile, measured with a microthermocouple 0.05 mm in diameter. The coordinate of the temperature sensor was determined with an error of ~0.05 mm in the radial direction and of ~0.5° along the perimeter. The average coefficient of heat transfer was determined by integration of the values of the local coefficient of heat transfer over the perimeter and by the heat-balance method. A detailed description of the experimental installation and of the procedure for conducting the investigation is given in [8, 9].

The character of the air movement in the vicinity of the cylinder was established as a result of the visual observations and the measurement of the velocity distribution. It is known that in purely forced flow over a cylinder at subcritical Reynolds numbers Re the separation of the boundary layer occurs at $\varphi \approx 86°$ with the formation of a region of vortex motion in the rear zone of the cylinder. The heat flux is the least in the separation region. With natural convection at Grashof numbers Gr of 10^3–10^9 the boundary layer is preserved over essentially the entire perimeter and the heat flux decreases monotonically in the direction of motion. The variation of the position of the separation point owing to the interaction of forced and natural convection is presented in Fig. 1 (points 1). A positive value of the parameter Gr/Re^2 corresponds to concurrence of the directions of forced motion and of the gravitational forces while a negative value corresponds to their opposing action. With concurrent mixed convection an increase in the parameter Gr/Re^2 leads to a downstream shift of the separation point, while when $Gr/Re^2 \geq 2$ separation hardly occurs, i.e., the boundary layer is preserved over almost the entire perimeter of the cylinder. The vicinity of the rear point of the cylinder is an exception.

For nonconcurrent convection an increase in the absolute value of Gr/Re^2 shifts the separation point closer to the front point of the forced flow. As shown by the flow visualization, when $Gr/Re^2 > 0.1$, countermotion develops on the lower part of the cylinder, the region of extension of which increases as the separation point shifts closer to the front point. The countermotion, interacting with the external stream and the wall, forms a boundary layer with a directional velocity which varies over the thickness. Since the external flow is formed here in the vortex region of the cylinder after separation of the boundary layer, the velocity and temperature pulsate in the outer part of the layer. The thickness of the layer is considerably greater than for purely natural convection. With an increase in Gr/Re^2 the countermotion developing under the action of free convection becomes more stable, and when $Gr/Re^2 > 4$ it extends over the entire perimeter of the cylinder. The rising plume of heated air mixes with the forced stream, creating additional temperature and dynamic disturbances, which lead to the appearance of velocity and temperature pulsations in the outer region of the boundary layer.

The results of a numerical solution of the boundary-layer equations [1] are also given in Fig. 1 (points 2). A velocity distribution of the external stream like that for an ideal fluid was adopted for the solution of the problem. This was one of the reasons for the disagreement of the calculation and the experiment, although it is not too considerable in the region of concurrent convection.

Examples of the distribution of the tangential velocity component in the boundary layer are given in Fig. 2. It must be noted that these results could be obtained only with a laser Doppler velocity meter, allowing the measurement of the local velocity down to 0.01 m/sec.

Velocity profiles with $Gr/Re^2 = 1.28$ are given in Fig. 2a, for different angles along the perimeter. In this case the influences of the forces of free and forced convection have the same order of magnitude. The

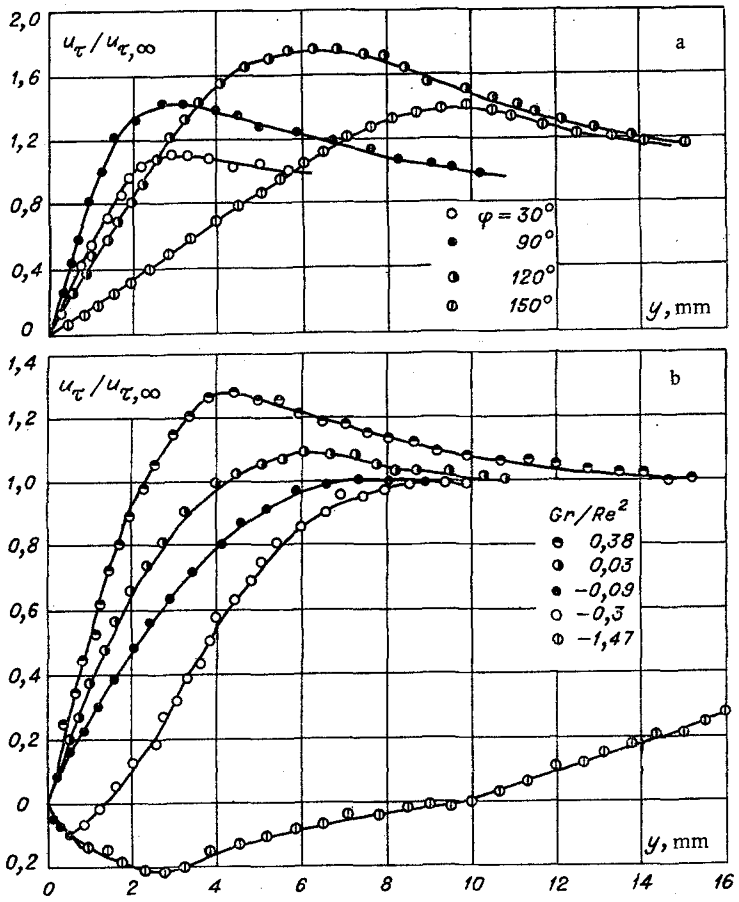

Fig. 2

velocity profiles have the maximum which is characteristic of natural convection. In this case the maximum value of the velocity is reached at $\varphi \approx 120°$. Separation of the boundary layer is not observed up to $\varphi \leq 150°$, which agrees with the data presented in Fig. 1.

Velocity profiles for different values of the parameter Gr/Re^2 and $\varphi = 90°$ are given in Fig. 2b. Both concurrent and nonconcurrent mixed convection are considered here. For concurrent convection the velocity profile becomes fuller with an increase in Gr/Re^2, with the formation of the characteristic maximum. For nonconcurrent convection the velocity profiles are deformed in the opposite direction with an increase in Gr/Re^2, approaching a separation profile in shape. A region containing return flow appears at $Gr/Re^2 \approx 0.3$. The thickness of the boundary layer grows considerably.

The results of a determination of the local coefficient of heat transfer are shown in Fig. 3 (lines 1). The character of the variation of heat transfer over the perimeter fully correspond to the properties of the flow in the boundary region. For concurrent convection (Fig. 3a) with $Gr/Re^2 \leq 0.02$ the distribution of the coefficient of heat transfer corresponds to forced streamline flow with a minimum in the region of $\varphi \approx 90°$. With an increase in Gr/Re^2 there is a redistribution of the coefficient of heat transfer connected with the downstream shift of the separation point. When $Gr/Re^2 \geq 2$ the variation in heat transfer at all points takes place monotonically with an increase in this parameter, approaching the values characteristic of natural convection.

In Fig. 3 we also present the results of numerical solutions of the boundary-layer equations (lines 2-4), made in accordance with [1-3]. The solutions of [2, 3] yield satisfactory agreement with experiment only in the vicinity of the front point with concurrent convection. The results of [1] diverge considerably from the experiment for the most part. Clearly, a solution of the problem in the boundary-layer approximation cannot yield a correct result for the conditions under consideration.

The variation of the local heat transfer has an entirely different form for nonconcurrent convection (Fig. 3b). Whereas for a small value of Gr/Re^2 the distribution of heat transfer over the perimeter corresponds to forced streamline flow, i.e., there is a heat-transfer minimum at $\varphi = 90°$ and a maximum at $\varphi = 0$, when the parameter Gr/Re^2 reaches 20 the distribution of heat transfer over the perimeter is monotonic and

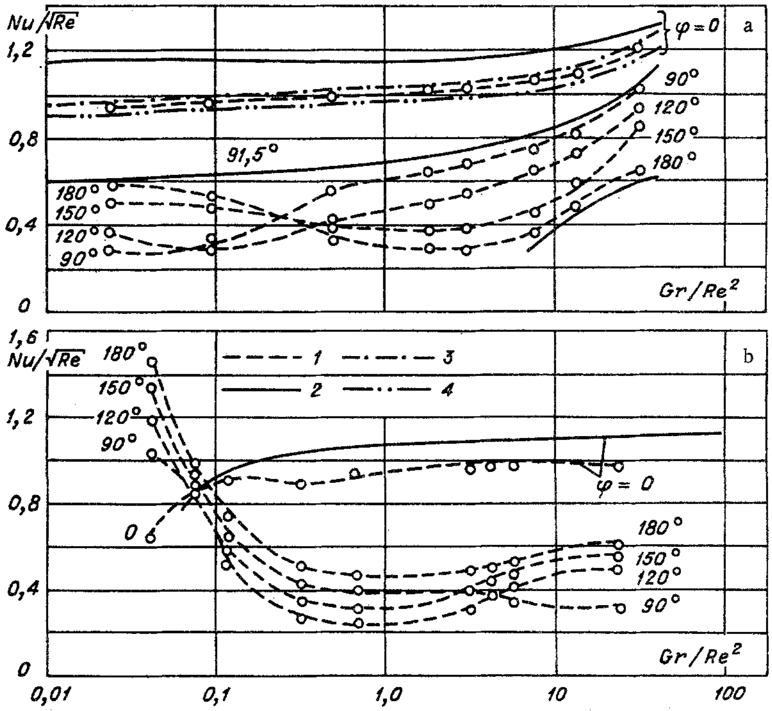

Fig. 3

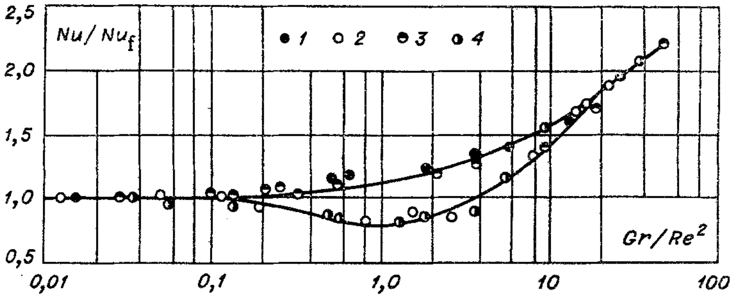

Fig. 4

corresponds to natural convection, and as a result of the variation in the direction of motion in the boundary region the heat transfer becomes greatest at $\varphi = 180°$ and least at $\varphi = 0$. With concurrent convection the influence of forced flow on the heat exchange is little noticed when $Gr/Re^2 > 10$, while with nonconcurrent convection this influence is important up to $Gr/Re^2 = 15$. This is explained by the properties of the interaction of the boundary flow and the external stream which were described earlier. It is interesting to note that under these conditions there is a mode of $Gr/Re^2 = 12$-15 in which the heat-transfer intensity over the perimeter remains constant.

In Fig. 4 points 1 and 2 are data on the average heat transfer for concurrent and nonconcurrent convection, respectively, obtained by the heat-balance method; 3 and 4 are data for concurrent and nonconcurrent convection, respectively, obtained by averaging the data on local heat transfer. The Nusselt number Nu_f corresponds to heat transfer during purely forced convection. It was determined from the equation [10] $Nu_f = 0.3737 + 0.37Re^{0.5} + 0.057Re^{0.67}$, which was tested by an investigation of heat transfer in the corresponding mode on the author's experimental installation.

On the basis of the investigation which was conducted we can note the following.

1. In transverse air flow over a cylinder the region of strictly mixed convection, determined by heat-transfer measurements, lies in the range of variation of the parameter Gr/Re^2 of $0.01 \le Gr/Re^2 \le 15.0$.

2. With concurrent action of free and forced motion, mixed convection leads to intensification of heat exchange. In the region of variation of $0.01 \le Gr/Re^2 \le 10$ the heat exchange is higher than for purely forced or purely free convection.

3. With free and forced motions of opposite directions the heat-transfer intensity decreases by 40% in comparison with concurrent flow.

4. Criterial functions were obtained for calculating the average heat exchange in the region of $0.01 \leq Gr/Re^2 \leq 15$:

a) for concurrent mixed convection

$$\frac{Nu}{Nu_f} = \left[0.74 + 0.07\left(1 - \lg\frac{Gr}{Re^2}\right)\right]\left[1 + \left(\frac{Gr}{Re^2}\right)^{0.5}\right]^{0.5}; \tag{1}$$

b) for opposing mixed convection

$$\frac{Nu}{Nu_f} = \left[0.74 + 0.07\left(\lg\frac{Gr}{Re^2} - 0.3\right)^2\right]\left[1 + \left(\frac{Gr}{Re^2} - 2\right)^2\right]^{0.0625}. \tag{2}$$

5. Separation of the boundary layer is practically absent for concurrent action of mixed convection with $Gr/Re^2 \geq 2$.

6. For opposing mixed convection the coefficient of heat transfer retains a constant value over the perimeter of the cylinder for $12 \leq Gr/Re^2 \leq 15$.

The diameter of the cylinder and the temperature of the oncoming stream are taken as the characteristic parameters in Eqs. (1) and (2).

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