Experimental investigation of the influence of large-scale vortex structures on heat transfer and drag on a smooth wall

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Abstract. The results of an experimental investigation on the effect of large-scale vortex structures on the heat transfer and drag coefficients on smooth wall are presented. Cylinder of diameter Ø=8 mm was installed in a channel of height \( H = 30 \text{ mm} \). The models were installed at distance of 40 mm from cylinder. The gap \( Y_0 \) between the wall and the edge of the cylinder varied from 1 to 21 mm. The value of the averaged relative drag coefficient \( c_x/c_{x,0} \) as well as the relative heat transfer coefficient \( St/St_0 \) were determined behind the cylinder at different flow velocity and position of cylinder in the channel. The maximum value of \( c_x/c_{x,0} = 1.48-2.04 \) (depending on the Reynolds number \( Re_x \)) is obtained for the case of \( Y_0 = 11 \) mm. At \( Y_0 = 21 \) mm the shear stress on the wall decreases to \( c_x/c_{x,0} = 0.00-0.17 \). At \( Y_0 = 1-2 \) mm, negative values of \( c_x/c_{x,0} \) are observed. The minimum value \( c_x/c_{x,0} = -1.01 - -0.71 \) corresponds to smallest gap \( Y_0 = 1 \) mm. The local value \( St/St_0 \) for \( Y_0 = 1-2 \) mm has maximum at distance of 80 mm from cylinder’s axis. With further increase in \( Y_0 \) to 9 mm, the value of \( St/St_0 \) decreases along the plate. At \( Y_0 = 11 \) mm local maximum is observed at distance of 90 mm from cylinder’s axis. At \( Y_0 = 21 \) mm, the values of \( St/St_0 = 1 \). The local values of \( St/St_0 \) varied in range \( St/St_0 = 1.0 \) - 2.5.

1. Introduction

Turbulent flows, heat and mass transfer are present in most power plants and energy transport systems. In such systems, to increase efficiency and reduce the material cost, it is necessary to intensify the processes of heat transfer and, at the same time, a decrease in the power for pumping the coolant. To achieve this goal, many researchers use different intensifiers (ribs [1], dimples [2,3], etc. [4,5]). Intensification of heat transfer is essential for the effective use of thermal energy. One of the classical problems is the intensification of turbulent heat transfer on a flat plate or in a slot channel by breaking the boundary layer. In this work, the boundary layer is destroyed by a cylinder located transversely to the flow direction and parallel to the walls of the slot channel.

2. Experimental setup and measurement equipment

The experimental investigation was performed in a subsonic wind tunnel (Figure 1a) [6–8]. An air flow passing through a high-pressure centrifugal fan (1) and a soft connection sleeve (2) arrives in the plenum chamber (3), where vortical structures break up and the air flow is equalized by means of a honeycomb and a straightener. Then the flow arrives in the slot test channel (6) through the contoured nozzle (4) with a flapper (5). The test channel of the setup for determining the heat-hydraulic parameters of the surfaces under study is 1080 mm in length, \( H = 30 \text{ mm} \) in height, and 300 mm in width. The upper and lower walls are made sectional: they consist of four sections of different length. On the lower wall of the channel, the tested model is mounted instead of one of the sections (Figure 1b).
To investigate the thermal characteristics of the surfaces one of the sections of the upper wall had a window made of Zn-Se glass, transparent in the IR radiation range, while above the test section there was mounted the ThermaCam SC3000 IR - imager in a dark box (7). To eliminate the fan vibration effect the test channel was mounted on a separate supporting frame (8) and connected with the fan through a smooth connection sleeve. Smooth variation of the flow velocity in the channel was ensured by a frequency drive. The experimental data were gathered and processed using a modern measurement equipment connected up to a high-performance personal computer (9). In the course of an experiment all the data channels, except from the IR imager, were polled at a frequency of 1 kHz, the signals from pressure transducers and elastic elements arrive at the NI PCI-6071 multichannel analog-digital transformer, and the thermocouples were connected with the NI USB-9213 digital transformer.

The tested model (Figure 1b) consists of two floating elements (10) suspended on a one-component strain-gage balance (elastic elements, 11). They record the force determined by the sum of the drag of the floating element and the force produced by the pressure difference along the plate length. To take account for this component of the total force in the forward and backward gaps of the floating elements there are mounted pressure taps (12). In between the floating and elastic elements there are electric heaters (13) with thermal insulation (14).

In studies, two plates with smooth surfaces are mounted on the floating elements. Any section of the lower wall can be heated. At the initial moment of time the temperature field nonuniformity is not greater the 2 K within the limits of any section and between them and 1 K within the limits of the models themselves for the smooth plate (except from the plate edges). The difference between the
initial temperature of the wall surface and the flow core temperature is about 70 K at the beginning of the experiment, whereas the end of the experiment it amounts to 15-25 K depending on the local heat transfer coefficients.

Before the main experiments we determined the boundary layer parameters in different channel sections. The boundary layer velocity profiles were measured using the Pitot—Prandtl probe, 0.5 mm in diameter. The velocity profiles were measured with the step of 0.125 mm, a minimum distance from the wall being 0.25 mm. The deviation of the measured velocity profiles from the logarithmic velocity profile in the turbulent flat-plate boundary layer \((\varphi=5.85\cdot\lg\eta+5.56\ [9], \text{ where } \varphi=V_c, \ \eta=z/V_c/\nu)\) is not greater than ±2.5%. The dynamic velocity \(V_c=\sqrt{\tau_0/\rho}\) was determined using the experimentally obtained value of the tangent stress on the wall \(\tau_0\), (in Pa), averaged over the floating element area.

The Reynolds number was based on the undisturbed boundary layer length, \(Re_s=(V\cdot x_w/\mu)\), and could range from \(Re_s=0.2\cdot10^6\) to \(Re_s=7\cdot10^6\). The flow core velocity \(V\) could vary from 40 to 125 m/s with the step of 5 m/s. The flow temperature varied from 291 K to 308 K. The undisturbed boundary layer length \(x_w\) was measured from the beginning of the test channel (Figure 1c). The density \(\rho\) (in kg/m\(^3\)) and the dynamic viscosity \(\mu\) (in Pa·s) were determined from the air pressure and temperature in the channel. The static pressure taps were mounted on the upper wall, while the thermocouples were at the channel axis, near the leading and trailing edges of the plates under study.

3. Technique of experimental investigations

3.1. Determining the drag coefficient

The drag coefficient was determined by directly weighing the models on a one-component strain-gage balance with account for the static pressure decrease along the plate length. Using the experimental data the drag coefficient was calculated from the formula

\[
c_s = \frac{F_s}{\rho V^2 / 2} = \frac{(F_s - F_{spl})/S_{side}}{\rho V^2 / 2} = \frac{(F_s - \Delta p_{pl} S')/S_{side}}{\rho V^2 / 2},
\]

where \(F_s\) is drag force (in N), \(F_{spl}\) is the overall force recorded by the elastic element (in N), \(F_{spl} = \Delta p_{pl} S'\) is the force induced by the static pressure decrease along the plate length (in N), \(\Delta p_{pl}\) is the pressure difference in the gaps of the floating element (in Pa), \(S_{side}\) is the area of the lateral surface of the channel (in m\(^2\)), and \(S'\) is the floating element endface area (in m\(^2\)).

Within the limits of ±5% the experimental values of the smooth-plate drag coefficient are in agreement with the Prandtl formula for the turbulent boundary layer [9] (Figure 2a) \(c_s=0.472\cdot (\lg(Re_s))^3.36\), where the Reynolds number \(Re_s\) is based on the laminar boundary layer length.

3.2. Determining the heat transfer coefficient

The heat transfer coefficient was determined using the unsteady heat transfer technique. At the initial moment the temperature fields of the model surfaces is recorded by the IR imager. Then the process of model cooling is recorded at a frequency of 1 Hz during 40 s. The flow core temperatures are measured by two thermocouples located at the channel axis near the leading and trailing edges of the plates under study. Then the heat transfer coefficient distribution over the wall is calculated using the time-dependent three-dimensional heat conduction equation for a solid (wall)

\[
\lambda \cdot \left( \frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} + \frac{\partial^2 T_s}{\partial z^2} \right) = \rho \cdot c \cdot \partial T_s / \partial t
\]

subject to the boundary condition of the third kind

\[
\lambda \cdot \nabla T = \alpha \cdot (T - T_0)
\]

where \(T_s\) is the solid temperature (in K), \(c\) is the specific heat of the solid (in J/(kg·K)), \(T\) is the temperature on the solid surface in a flow (on the surface of the relief under study), \(T_0\) is flow core temperature (in K), \(\lambda\) is the thermal conductivity coefficient of the solid (in W/(m·K)), \(t\) is time (in s), and \(\alpha\) is the unknown heat transfer coefficient.
To reduce heat fluxes along the plate surfaces the models under study were made of Plexiglas with small thermal conductivity $\lambda=0.19$ W/(m·K). The initial conditions given from the experiment and the variation in the surface state for the time interval under consideration being known, the heat transfer coefficient field over the surface averaged over the experiment time was determined with account for the flow core temperature. An example of determining the two-dimensional field of the heat transfer coefficients using the solution of the above-mentioned equation, the comparison of the solution obtained with that for an one-dimensional semi-infinite body, and an analysis of the results and the occurring uncertainties are made in detail in [6].

For the smooth plate the experimental values of the heat transfer coefficient agree within ±5% with the dependence for the power-law boundary layer $St=0.0296\cdot Re^{-0.2}\cdot Pr^{-0.6}$ [10] (Figure 3b), where the Reynolds number is based on the thermal boundary layer length.

Thus, for obtaining the drag and heat transfer coefficients the following flow parameters were determined in the course of the experiments: the total pressure in the plenum chamber, the total and static pressures in the flow core ahead of and behind the models, the pressures in the forward and backward gaps of the models, the flow core temperature ahead of and behind the models, the model support temperature, the forces acting on the elastic elements, and the temperature distribution over the model surfaces.

3.3. Determining the uncertainty of experimental data
The uncertainty in determining the heat-hydraulic characteristics was estimated for a 95% confidence level, in accordance with the method presented in [11]. The maximum uncertainties of the measured values are as follows: $(U_{St})_{0.95}=5.4 \%, (U_{C_t})_{0.95}=5.0 \%, (U_{Re})_{0.95}=2.5 \%$.

3.4. Studied geometry
A cylinder with a diameter of $\Theta=8$ mm was placed in the channel at a distance of 0.66 m from the end of the nozzle (Figure 1c). Floating elements with smooth plates were installed at a distance of 40 mm ($5 \cdot \Theta$) from the cylinder and 0.7 m from the entrance to the channel. The cylinder fastening scheme allowed changing the clearance $Y_0$ between the bottom wall and the lower cylinder edge - the gap size $Y_0$ was varied in the range of 1-21 mm. During the experiments, the value of the drag coefficient averaged over the area of the floating element $c_t$ behind the cylinder was determined (and also the value of $c_t/c_{t,0}$, where $c_{t,0}$ was determined from the undisturbed flow parameters) at different flow velocities ($u_0=40$-125 m/s) and cylinder positions in channel (the value of $Y_0$ was 1 7, 9, 11 and 21 mm). The heat transfer coefficient $St$ (and the $St/St_0$) along the length of the plate was determined for a flow with an unperturbed flow rate of 75 m/s for the following cylinder positions: $Y_0=1 \text{ mm}$ and $u_0=108$ m/s in cross section to the
cylinder (at a distance of 0.353 m from the nozzle end) and in sections corresponding to the beginning and end floating element (at a distance of 0.698 and 0.825 mm respectively).

Also after this, the fields of total and static pressure in the Pitot-Prandtl tube receivers were determined in different sections behind the cylinder at a mean velocity of 64 m/s.

4. Result and discussion
The following results were obtained for the considered flows:
- The maximum value of $c_i/c_{i0}=1.48-2.04$ (depending on the Reynolds number $Re_c$) is obtained when the cylinder is positioned on the channel axis ($Y_0=11$ mm). As the cylinder approaches the opposite wall ($Y_0=21$ mm), the tangential stress on the wall decreases to $c_i/c_{i0}=0.00-0.17$. With a decrease in the gap size to $Y_0=1-2$ mm, negative values of the relative coefficient of friction apparently related to the presence of an extended region of backward flow in the wake of the cylinder. The minimum value $c_i/c_{i0}=(-1.01)-(-0.71)$ corresponds to the smallest gap $Y_0=1$ mm (see Figure 3a);
- the value of $c_i/c_{i0}$, determined by means of the pressure profile at $Y_0=1$ mm and $u_0=108$ m/s, is $c_i/c_{i0}=-0.75$. This value is in satisfactory agreement with the value $c_i/c_{i0}=0.71$, obtained by means of a floating element, but both of these quantities have considerable uncertainty (see Figure 3b);
- the value of the drag coefficient $c_i$ of the section of the channel with the cylinder considerably exceeds the friction resistance of the flat channel. At $Y_0=1$ mm and $u_0=108$ m/s, the ratio $c_i/c_{i0}$, determined using the profiles of the total and static pressures in the cross sections $x=0.353$ and $x=0.825$, is $c_i/c_{i0}=12.2$.

![](image)

Figure 3. The dependence of averaged over the length of the plate the relative drag coefficient, $c_i/c_{i0}$, on the Reynolds number $Re$, and the gap between the wall and the cylinder $Y_0$ (a) and the velocity profiles in sections up to ($x=0.353$ m) and after ($x=0.698$ and 0.825 m) with a gap between the wall and cylinder $Y_0=1$ mm (b)

- the heat transfer intensification after the cylinder was considered in the section from 710 to 823 mm. The local value $St/ST_0$ for $Y_0=1.2$ mm has a maximum at a distance of 80 mm from the axis of the cylinder. With a further increase in $Y_0$ to 9 mm, only a decrease in the value of $St/ST_0$ along the longitudinal coordinate is observed. At $Y_0=11$ mm, the local maximum is observed at a distance of 90 mm from the axis of the cylinder. At $Y_0=21$ mm, the values of $St/ST_0$ are 1. The local values of $St/ST_0$ (except for the case of $Y_0=21$ mm) varied in the range $St/ST_0=1.35$ (see Figure 4);
- when considering the fields of the total and static pressures in the Pitot-Prandtl tube receivers at $Y_0=2$ mm gap (Figure 5), the presence of an extended flow behind the cylinder is seen, which is satisfactorily correlated with the values of the relative coefficients of resistance (the presence of a region of backward flow reduces the value of resistance on the wall in the direction of flow, however, the presence of such zones substantially increases the loss of the total pressure) and heat transfer (intensification of heat transfer in the region of flow attachment);
- the data obtained are in satisfactory agreement with the data of [12,13].
sections with the cylinder considerably exceeds the friction resistance of the flat channel corresponds to smallest gap \( Y_0 = 1 \) mm. At \( Y_0 = 11 \) mm. The value of the averaged relative drag coefficient \( c_f/c_{f0} \) as well as the relative heat transfer coefficient \( St/St_{0} \) were determined behind the cylinder at different flow velocity and position of cylinder in the channel. The flow core velocity \( V \) varied from 40 to 125 m/s.

The maximum value of \( c_f/c_{f0} = 1.48 - 2.04 \) (depending on the Reynolds number \( Re_c \)) is obtained for the case of \( Y_0 = 11 \) mm. At \( Y_0 = 21 \) mm the shear stress on the wall decreases to \( c_f/c_{f0} = 0.00 - 0.17 \). At \( Y_0 = 1 - 2 \) mm, negative values of the \( c_f/c_{f0} \) are observed. The minimum value \( c_f/c_{f0} = -1.01 - -0.71 \) corresponds to smallest gap \( Y_0 = 1 \) mm. The value of the drag coefficient \( c_f \) of the section of the channel with the cylinder considerably exceeds the friction resistance of the flat channel, i.e. \( c_f/c_{f0} = 12.2 \) in the sections between \( x = 0.353 \) and \( x = 0.825 \).

5. Summary and conclusion

The results of an experimental investigation on the effect of large-scale vortex structures on the heat transfer and drag coefficients on smooth wall are presented. Cylinder of diameter \( \Phi = 8 \) mm was installed in a channel of height \( H = 30 \) mm. The drag coefficient was determined by direct weighing of models using floating elements. Heat transfer coefficients were determined by means of the unsteady heat transfer method. The models were installed at distance of 40 mm from cylinder. The gap \( Y_0 \) between the wall and the edge of the cylinder varied from 1 to 21 mm. The value of the averaged relative drag coefficient \( c_f/c_{f0} \) as well as the relative heat transfer coefficient \( St/St_{0} \) were determined behind the cylinder at different flow velocity and position of cylinder in the channel. The flow core velocity \( V \) varies from 40 to 125 m/s.

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The local value \( \text{St}/\text{St}_0 \) for \( Y_0=1-2 \text{ mm} \) has maximum at distance of 80 mm from cylinder’s axis. With further increase in \( Y_0 \) to 9 mm, the value of \( \text{St}/\text{St}_0 \) decreases along the plate. At \( Y_0=11 \text{ mm} \) local maximum is observed at distance of 90 mm from cylinder’s axis. At \( Y_0=21 \text{ mm} \), the values of \( \text{St}/\text{St}_0 = 1 \).

The local values of \( \text{St}/\text{St}_0 \) varied in range \( \text{St}/\text{St}_0 = 1.0\ 2.5 \).

Further studies should be aimed at identifying the dependencies between the parameters of large-scale vortex structures and the processes of heat and momentum transfer, and also on the study of the effect of large-scale vortex structures on the processes of heat transfer and drag on relief surfaces.

6. References

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