Heat transfer and skin-friction in channel with flow behind a cylinder

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Abstract. The paper presents the results of an experimental study of the parameters of the boundary layer, distribution of static pressure, heat transfer and friction coefficients of smooth surface located in the wake behind the cylinder in the channel. Cylinders of various diameters were placed in a slotted channel with a height of 30 mm on its axis. In all experiments, the flow velocity at the inlet was 50 m/s. The cylinder was made unheated. The friction coefficients of the smooth model were determined both from the velocity profile in the boundary layer and by direct weighing of the model on a one-component strain-gage balance. The local values of the heat transfer coefficients were determined by transient heat-transfer method using a thermal imager. The values of the heat transfer and friction coefficients in the wake behind the cylinder, referred to the values on the smooth wall in the undisturbed flow, varied in the range 1.15–1.65 and 1.3–1.75, respectively. The value of the Reynolds analogy factor for all cylinder diameters turned out to be less than unity.

1. Introduction

Heat transfer enhancement in channels of power equipment is one of the most pressing problems of modern thermal physics [1]. In most cases, heat transfer enhancement process is accompanied by an increase in total pressure loss. The most common way to achieve the desired balance between heat transfer and drag, which determines the level of pressure loss, is by using heat transfer intensifiers. At the same time, for most types of intensifiers, a moderate heat transfer enhancement accompanied by a significant increase in drag.

It is obvious that the searching for methods that cause an outstripping increase in heat transfer parameters is of significant interest from both practical and scientific points of view. According to a number of paper [2,3], the installation of various manipulators (belts, aerodynamic profiles, cylinders) in the boundary layer on a smooth plate leads to a significant violation of the Reynolds analogy: a decrease in friction in the boundary layer on the wall (up to 30% [2]) behind the manipulator accompanied by an heat transfer enhancement (up to 1.9 times [3]).

However, flows in pipes and channels are of greater practical interest. According to the results of some published paper (see, for example, [4]), the installation of a bluff cylinder into the channel also leads to heat transfer enhancement on the wall in the wake behind the cylinder.

In present work, an experimental study of the parameters of the boundary layer, the heat transfer and friction coefficients on a smooth surface located in the slotted channel in the wake behind the cylinder is carried out.
2. Description of the experimental setup

Experimental studies were carried out on a subsonic wind tunnel with a working part in the form of a slotted rectangular channel [5–7] (width \( B = 300 \) mm, height \( H = 30 \) mm, length \( L = 1080 \) mm, see figure 1). A circular cylinder was installed on the axis of the working channel across the flow (parallel to the bottom wall). In the course of experiment, the cylinder diameter varied in the range 2.75-8 mm (cylinders with diameters \( d = 2.75, 3.2, 4.2, 5.66, 7.5, \) and 8 mm were considered). The trailing edge of the cylinder was located at a distance of 700 mm from the beginning of the channel. The inlet velocity for all channel configurations remained constant (50 m/s) and was controlled using a Pitot-Prandtl tube installed at the inlet. In the experiments, the profiles of the averaged velocity component on the wall in the wake behind the cylinder, the friction and heat transfer coefficients, and the distribution of static pressure on the wall both before the cylinder and in its wake were investigated.

Velocity profiles were measured using a DISA 55M01 CTA hot-wire anemometer with a 55M17 measuring bridge, equipped with a small-sized single-wire sensor 55P81 with thermal compensation manufactured by Dantec Dynamics (wire length 1.25 mm, wire thickness 5 μm). The sensor was installed on a coordinate device, which made it possible to move it to any point of the channel at a distance of up to 120 mm from the trailing edge of the cylinder with an accuracy of 0.02 mm and a minimum step of 0.05 mm in the direction normal to the wall. When measuring the velocity profiles, the sensor was moved from the flow core and approached the wall to a distance of 0.2-0.3 mm, until the velocity value at this point dropped to \( V_{\text{min}} / V_{\text{max}} < 0.5 \), where \( V_{\text{max}} \) is the maximum velocity in this section. The smallest distance from the wall \( y^+ \approx 30 \) - the experimental points fell above the transition region of the boundary layer - either on the logarithmic part of the profile, or in the wake. The measurements were carried out at distances: 1-10 mm with a step of 1 mm, then up to 100 mm with a step of 10 mm and at a distance of 120 mm.

![Figure 1. Schematics of the setup (a) and working channel (b)](image_url)

The friction coefficient at the channel wall in the wake of the cylinder was determined in two ways - by the measured velocity profile on the logarithmic section of the boundary layer, and by weighing the model on a floating element. In the first case, the local coefficient of friction is determined, in the
second - averaged over the length of the floating element (0-125 mm from the trailing edge of the cylinder). After averaging the local friction coefficients, they can be compared with the value obtained on the floating element. The result will indicate the acceptability of using a universal velocity profile to determine the characteristics of the boundary layer.

The heat transfer coefficient was determined using the transient heat transfer method [7–9]. The one-dimensional equation of transient cooling of a wall of finite thickness was used with account for the initial temperature nonuniformity over the plate thickness. To reduce heat fluxes along the plate surface the model was made of Plexiglas with small thermal conductivity $\lambda=0.19$ W/(m·K) and was 6 mm in thickness. At the initial moment of experiment the IR camera recorded the temperature field of the model surface. The process of model cooling was recorded during 60 s. Then a heat transfer coefficient most accurately describing the experimentally obtained cooling rate was selected. The value thus obtained was averaged over the entire model surface.

To measure the static pressure profile, the bottom wall was drained (the hole diameter was 0.5 mm) both before (up to the section 45 mm from the trailing edge of the cylinder) and after the cylinder (up to 145 mm downstream from the trailing edge). The pitch of the main holes was 10 mm, and the pitch of the outer holes was 5 mm.

3. Results of experimental studies

As the cylinder diameter increases and, accordingly, the channel becomes blocked, the static pressure drop increases - the minimum static pressure value caused by flow acceleration decreases with increasing cylinder diameter. An increase in pressure losses on the cylinder is caused by an increase in the form drag due to an increase in the cylinder diameter, an increase in wake losses is caused by an increase in the friction coefficient on the wall in the wake behind the cylinder (figure 2).

![Figure 2](image_url)

**Figure 2.** Variation in static pressure along the length of the channel for different diameters of the cylinder installed on the axis of the channel

Directly behind the cylinder (at a distance of up to 3 mm from the trailing edge), the profile becomes much less filled, while the wake parameter of the boundary layer increases. Further, with an increase in distance from the cylinder and an increase in its diameter, the velocity profile becomes more filled (the form parameter of the boundary layer in this case becomes lower than the form parameter of the boundary layer in the undisturbed flow), and for cylinder diameters greater than 4.2 mm in the outer part of the boundary layer, the experimental points turn out to be below the universal logarithmic curve (figure 4). The logarithmic velocity profile in the inner part of the boundary layer is preserved in the considered range of flow parameters and channel geometries (figure 4).

As the cylinder diameter increases, the velocity profile in the inner part of the boundary layer ($y^+<100$) practically does not change compared to the boundary layer in the undisturbed flow, while in the outer part of the velocity profile begins to deform significantly (figure 4).
The friction and heat transfer coefficients on a smooth wall in the wake behind the cylinder are always higher than in a channel without a cylinder and grow with the diameter of the cylinder (figure 5).

The averaged relative friction coefficient of the smooth model (figure 6) in the wake behind the cylinder \( \frac{c_f}{c_{f0}} \) increased from \( \frac{c_f}{c_{f0}} = 1.28 \) (at \( d = 2.75 \text{ mm} \)) to \( \frac{c_f}{c_{f0}} = 1.75 \) with a cylinder diameter of 8 mm. The average \( St/\text{St}_0 \) values increased from \( St/\text{St}_0 = 1.15 \) with a diameter of 2.75 mm to \( St/\text{St}_0 = 1.6 \) with a diameter of 8.0 mm.

![Figure 3. Velocity profiles in external coordinates: a - in an undisturbed flow, b - in a wake behind a cylinder with a diameter of 6.6 mm](image)

![Figure 4. Velocity profiles in universal coordinates: a - in an undisturbed flow, b - in a wake behind a cylinder with a diameter of 6.6 mm](image)
Figure 5. Change in the relative friction (a) and heat transfer (b) coefficients on the smooth wall in the wake behind the cylinder along the length of the channel.

Figure 6. Values of friction coefficient, heat transfer and Reynolds analogy factor averaged over the 0-125 mm section behind the trailing edge of the cylinder.

4. Summary and conclusion
The paper presents the results of an experimental study of the parameters of the boundary layer, distribution of static pressure, heat transfer and friction coefficients of smooth surfaces located in the wake behind the cylinder in the channel. The following results were obtained:

- an universal logarithmic part of velocity profile is preserved in the considered range of flow parameters and channel geometries. The averaged friction coefficients determined by the velocity profile and by means of the floating element coincide;
- the friction and heat transfer coefficients in the wake behind the cylinder (in the section 3-120 mm from the trailing edge) are always higher than in the channel without the cylinder and grow with the increase in the diameter of the cylinder. In this case, the shape parameter of the boundary layer is always lower than the shape parameter of the undisturbed boundary layer;
• the friction coefficients averaged over the 0-120 mm section are always higher than the heat transfer coefficients.

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