Features of air flow and heat transfer control behind a backward-facing step using a vortex generator pair

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Abstract. The paper presents experimental results on the study of flow dynamics and heat transfer in the separation region behind the backward-facing step with longitudinal vortex generators (VG) installed at an angle to the flow of 30º at Re = 4000. The VG installation reduces the recirculation region and the induced longitudinal vortices and rearranges the flow structure in the separation region. The influence of a VG on the local and average thermal characteristics behind the backward-facing step is investigated and their thermohydraulic efficiency is estimated.

1. Introduction
The study of heat transfer enhancement in separation regions is a complex and multifaceted task. For a more complete understanding of this process, the dynamic and thermal characteristics of the flow are investigated. Three-dimensional intensifiers that form a system of longitudinal vortices are the easiest to manufacture. One of the types of such intensifiers is a pair of plates of different shapes, installed at an angle to the flow. The flow control behind the step by means of tabs installed at the step edge was investigated in [1, 2]. The mounted tabs reduced the recirculation area behind the backward-facing step due to the induced longitudinal vortices. The influence of the shape, installation angle and pitch between the tabs on heat transfer behind the step was investigated in [3]. The influence of this kind of vortex generators on heat transfer enhancement on the plate was considered in [4, 5]. The aim of this work is to comprehensively study the effect of a pair of VGs mounted in the vicinity of the step edge on the flow dynamics and heat transfer.

2. Experimental setup and procedure
The experiments were carried out in a 1 m long channel with a rectangular cross-section of 20×150 mm² (Fig. 1). The channel was made of a 10 mm thick textolite sheet. At a distance of 600 mm from the channel inlet, there was a backward-facing step with a constant height \( H = 10 \text{ mm} \). On the lower wall of the channel, behind the backward-facing step, a thermal section with a length of 400 mm was located. The entire surface of the lower channel wall behind the step was heated using an electric heater made of 50 μm thick titanium foil. A 135×300 mm² window was cut in the channel wall to perform thermal imaging measurements. The heater at the view point of the thermal imaging camera was blackened. When processing the data, the radiation losses, which in the considered case were less than 8%, were taken into account as well as the losses from free convection from the outer side of the wall.
Vortex generators (VG) with height $h = 6$ mm, length of 15 mm and thickness $e = 1$ mm, attached in the vicinity of the step edge, were made of plastic. The distance between the vortex generator plates was $S = 6$ mm. The angle $\beta$ (the angle between the flow direction and the tab plane) was $30^\circ$. Rectangular tabs were investigated. The Reynolds number calculated from the step height $H$, the average flow rate $U_m$ and the kinematic viscosity of air $\nu$ was $Re = U_mH/\nu = 4000$. The average air temperature in the channel was $21 \pm 1^\circ$C. At a distance of 25 calibers from the inlet, the flow was stabilized, and the velocity profile was close to the power law with an exponent $n \approx 1/7$.

![Figure 1](image_url)  
**Figure 1.** Scheme of setup. All dimensions are in millimetres.

The velocity fields were measured with the digital tracer visualization method (PIV). The measuring area was behind the step $3H \times 8H$. Velocity fields were measured in 4 planes in the transverse direction: in channel plane symmetry ($Z/H = 0$), along the inner edge of the VG plate ($Z/H = 0.3$), along the outer edge of the VG ($Z/H = 1.05$) and at a distance $Z/H = 2$ from the model symmetry center. The equipment of the PIV-method included a pulsed laser with a double flash, synchronized with a digital camera for measuring the two-dimensional velocity field. The time interval between laser flashes in a pair of frames was 20 $\mu$s, and their duration was 5 ns. The measuring area of the PIV complex had dimensions of $30 \times 40$ mm$^2$. For each region 4000 pairs of frames were obtained. The size of the camera matrices was 1360x1025 pixels, and in the calculation of velocity fields the image was divided into computational domains with dimensions of $32 \times 32$ pixels. The velocity fields were calculated using an iterative cross-correlation algorithm with 50% overlap of the computational domains. Then the velocity vectors were filtered in two ways: by signal to noise ratio and by median filter.

The pressure coefficient was calculated by formula $C_p = 2(p_i-p_0)/U_m^2$, where $p_i$ is the static pressure, $p_0$ is the reference measured pressure at a distance of 40 mm from the step, and $U_m$ is the average velocity in the same cross-section.

A thermal model of the same size for measuring the temperature fields in the longitudinal direction on the wall behind the backward-facing step was made of a heat-insulating material. A thin, 37 $\mu$m thick, conductive film was glued onto its surface; therefore, the boundary condition $q = \text{const}$ was satisfied. The wall temperature was measured with a NEC Thermo Tracer TH7102 IR Imager (Japan) with a spectral range of 8-14 $\mu$m. The obtained temperature field was digitized by at least two thermocouples, and the thermograms were plotted using special computer programs.

The local heat transfer coefficient was calculated by the formula $\alpha = q / (T_w - T_0)$, where $q$ is the heat flux measured on the heated wall of the model behind the step, $T_w$ is the temperature of the heated wall, and $T_0$ is the temperature of the flow in the channel before the step. The Nusselt number was calculated using the following formula:

$$Nu = \frac{\alpha H}{\lambda}$$  \hspace{1cm} (1)

where $\alpha$ is the heat transfer coefficient, and $\lambda$ is heat conductivity of air, determined by the air flow temperature.
3. Results and discussion

Distribution of the average longitudinal velocity at different distances from the step in the center of symmetry of the channel \((Z/H=0)\) is shown in Fig. 2. The scale of the longitudinal mean velocity is shown in the upper right part of the figure; and \(U_{\text{ref}}\) is the velocity on the axis of the undisturbed flow before separation. As in [6], in a flow without disturbances, the negative value of longitudinal velocity component takes place at \(X/H=2\div5\). Installation of a VG introduces disturbances into the flow, whose values depend on the region of disturbance input; in particular, in the presence of VG, the region of the return flow does not differ much from the classical case. It was noted in [6] that the maximum return velocity behind the step does not exceed \(0.2U_{\text{ref}}\), and in cases of VG installation, the latter did not exceed \(0.2U_{\text{ref}}\) either.

![Figure 2. Average longitudinal velocity profiles.](image)

![Figure 3. Transverse average velocity profiles.](image)

If a VG is mounted in the center of symmetry of the channel, on the first caliber immediately behind the step, the transverse component of velocity behaves similar to the case of the flow around a smooth step, and a downward flow is observed in the middle of the channel. As in the case of a smooth step,
with a distance from the separation point downstream, the transverse component increases and at the step height a minimum, being 16% of the velocity in the flow core, is reached.

At a distance of one caliber from the step bottom, directly at the VG height, the transverse velocity is negative (Fig. 3) and its value increases downstream. At a distance of 7 calibers, the minimum $V$ is 10% of the velocity in the flow core.

For the case of a flow around a step without a VG, the maximum of pulsations on the first caliber is at the level of the step height (Fig. 4); i.e., in the area of mixing of the separated boundary layer with the flow in the recirculation area. If a VG is installed, the flow pattern, for both average and rms characteristics, resembles the flows behind a two-dimensional obstacle mounted at the step edge. The maximum flow pulsations on the first caliber are just below the obstacle height and are observed along 7 calibers. Diffusion of turbulent vortices occurs downstream.

In the central cross-section, the recirculation zone for the diffuser flow ends before ~ 5 calibers, in contrast to the smooth step where $X_r = 6.8H$. For the case with a VG, one half of the channel was investigated in the longitudinal plane between the tabs $Z/H = 0$, on the outer edge of the VG plate ($Z/H = 0.3$), on the inner edge ($Z/H = 1.05$), and at a distance of 1 caliber from VG ($Z/H = 2.05$). The distance $X_r$ in plane $Z/H = 0.3$ was 5.16H. For $Z/H = 1.05$, $X_r = 5.22$ and for $Z/H = 2$, $X_r = 2.89H$. All this indicates non-uniformity of the flow in the transverse direction.

The results of thermographic visualization of the temperature field on the wall behind the backward-facing step are presented in Fig. 5. The data for a smooth step (Fig. 5a) and data for a step with a VG, installed at its edge (Fig. 5b), are shown there. In the first case, behind the step edge, a stagnant area is formed in the central zone, clearly visible with the help of soot-oil visualization [7]. In the thermogram (Fig. 5a), this is the hottest zone. The colder zone is in the area of attachment. Vortex generators (Fig. 5b) affect the flow near the step, for example, in the attachment area, three cooled regions are observed: one region is in the midsection (it is elongated along the flow and has the shape of a feather) and two regions of a similar shape are behind each VG plate.

The region formed between VGs is colder, perhaps due to the forming longitudinal vortices. Unlike tabs, the cold area does not shift to the step, since in the presence of tabs, the distance to the attachment point decreases [1].

Figure 6a shows the longitudinally averaged profiles of the Nu number. When installing a VG, the coordinate of the maximum heat transfer $X_{max}$ approaches the step. So, for a smooth step, $X_{max}$ is 5.5 calibers, and in the presence of vortex generators, the $Nu_{max}$ position is shifted to 4.9 calibers. The $Nu_{max}$ value for the case with mounted VG is 11.5% higher than heat transfer for a smooth step. The $Nu_L$
number is averaged over a distance of 20 calibers; this is the distance where the pressure recovers. Thus, with the VG, the average NuL number exceeds heat transfer for a smooth step by 6.7%.

Figure 5. Thermograms behind the backward-facing step: (a) Step; (b) with VG.

Figure 6. Profiles behind the step: (a) – transversely averaged Nu number; (b) – Cp along the centre line of the model.

The distribution of static pressure (Fig. 6b) on the lower wall of the channel after flow separation in the presence and absence of a VG is of a similar nature. However, for the case with a VG, $C_{p_{\text{min}}} = -0.223$, the pressure recovery occurs at $C_{p_{\text{max}}} = 0.203$, and the difference between them is $\Delta C_p = 0.426$. For a smooth step, this value is noticeably lower due to weak rarefaction in the separation zone.

According to the distribution of pressure coefficients, the local hydraulic resistance was calculated at a distance of 0.2 m from the sudden expansion, since at this distance, all transverse disturbances caused by a VG almost cease affecting pressure distribution. Local hydraulic resistance was calculated by the formula $f = 5/9 - (U_{\text{m}}/U_{\text{m0}})^2 C_p$, where $U_{\text{m0}}$ is the average flow rate. Resistance with the introduction of a VG increases by 50%. To take into account the thermohydraulic characteristics of heat transfer intensifiers, the Reynolds analogy factor, determined by the ratio of the normalized Nu number to the
normalized coefficient of hydraulic losses, is widely used. Normalization is performed according to the characteristics of a channel with a smooth step. The Reynolds analogy factor is 0.76. To assess the thermohydraulic efficiency, a complex is used, which, in contrast to the Reynolds analogy factor, allows considering the power for coolant pumping $\frac{Nu}{Nu_0}/(f/f_0)^{1/3}$. The thermohydraulic efficiency for a step with a VG deteriorates by 7%.

Conclusions
The paper has presented the results of an experimental study of the flow structure and heat transfer in the separation region behind the backward-facing step with longitudinal vortex generators installed at an angle of 30° to the flow at $Re = 4000$. The VG installation reduces the recirculation region, and induced longitudinal vortices rearrange the flow structure in the separation region. The installation of a VG leads to a change in the flow structure and, as a consequence, the area of heat transfer enhancement is redistributed into three zones (two lateral and one central zone). The character of the averaged heat transfer does not change; the maximum of heat transfer somewhat shifts towards the step and becomes 11% higher than when flowing around a smooth step.

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