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Heat transfer in turbulized gradient flows

I A Davletshin, R R Shakirov, A A Paereliy

Institute of Power Engineering and Advanced Technologies, FRC Kazan Scientific Center, Russian Academy of Sciences, 2/31 Lobachevskogo str., Kazan, 420111, Russia

davlet60@mail.ru

Abstract. Convective heat transfer in a plane channel with superimposed pressure gradient (converging and diverging channel) has been studied experimentally. The air flow was turbulized by different arrangements at the channel inlet. Distributions of heat transfer coefficients within the gradient part of the channel have been obtained in different regimes. Patterns of the effect of turbulence intensity on heat transfer have been described.

1. Introduction

Gradient flows are frequently encountered in engineering [1-5]. Experimental studies [1] and numerical simulations [2] demonstrate that favorable pressure gradient (in converging channels) furthers flow laminarization and delays laminar-turbulent transition. This can reduce hydraulic losses and heat transfer: their respective coefficients fall below the turbulent values but remain higher than in zero-gradient flows. Furthermore, the effect of pressure gradient on hydrodynamics and heat transfer increases with its value. According to the literature, there are other factors acting along with the pressure gradient: turbulence promoters, zero-gradient entrance, etc.

Adverse pressure gradient flows (in diverging channels) are yet another class of flows which seems to be more complex due to possible flow separation. Such flows cause increased hydraulic losses, unstable regimes, unfavorable conditions for heat transfer, etc., and therefore they are undesirable [3, 4].

Recent progress in the development of optical methods has significantly expanded the capabilities of experimental research of complex flows. In particular, PIV measurements are employed in different classes of problems for investigation of the kinematic flow structure (fields of velocity, turbulence, vorticity). Such a study of kinematic structure of gradient flows has been reported in [5].

Currently, there are papers that demonstrate a possibility to laminarize turbulent flows by turbulizing impacts on the flow [6], therefore we expect a combined effect of different factors on hydrodynamics and heat transfer in gradient flows with turbulence promotion.

2. Experimental setup and procedure

Heat transfer in gradient flows was investigated on a specialized experimental setup. The air with environmental parameters was supplied to the channel inlet. The flow rate through the test section was provided by a required number of critical flow nozzles with the uncertainty in the flow rate not exceeding 0.25%.

The test section of experimental setup was a plane channel with the length of 1.2 m (figure1). Converging/diverging section *3* was placed in the middle of the channel. The channel width along its whole length was 150 mm. A smoothly shaped (in one plane) inlet *1* was attached to the channel, and the flow was turbulized at the junction point. Flow turbulization was arranged at the channel inlet (position *5* in figure1).

Converging/diverging geometries with the angle $\varphi=5°$ were considered. The angle was set by inclination of the top wall of the channel. The channel height upstream of the converging part was $H_0=100$, the one downstream of the converging part was $H_1=63$ mm. In the diverging channel the heights were reversed ($H_0=63$ mm, $H_1=100$ mm). The length of the gradient part was $L=450$ mm. Heat transfer was measured on the lower heated wall *2*. Thermometer *4* measured the air flow temperature at the inlet.
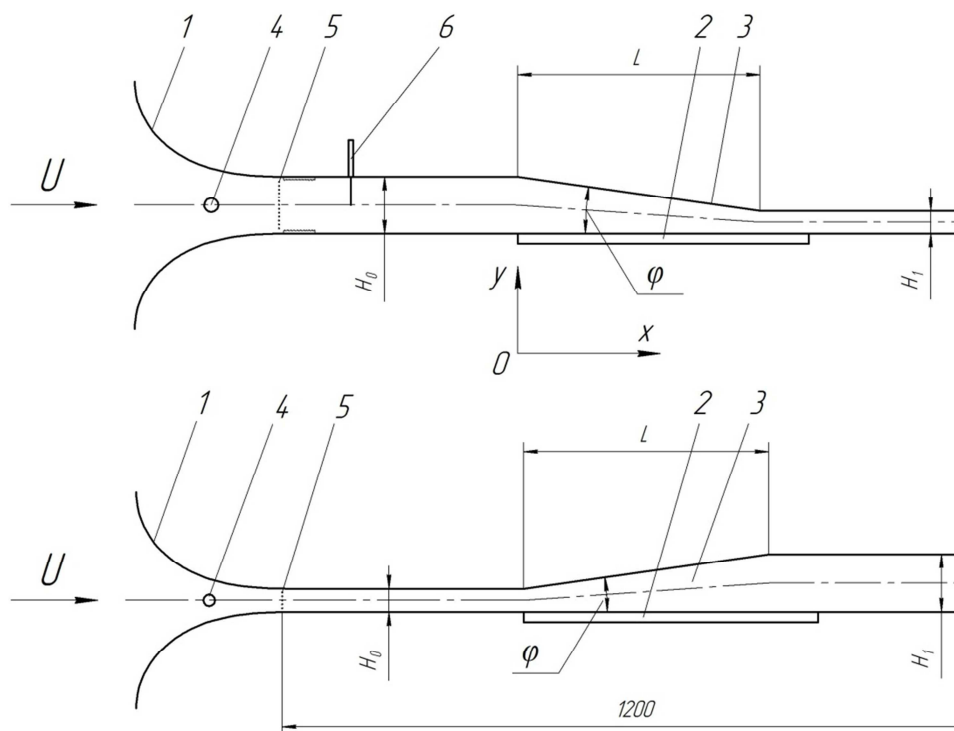

Figure 1. Plane channel with converging (*a*) or diverging (*b*) parts: 1 – inlet; 2 – heated wall; 3 – gradient section; 4 – thermometer; 5 – turbulence promoter (grid, grid with strips, abrasive); 6 – hot-wire anemometer.

Four different arrangements for flow turbulization were employed: 1 – smooth channel; 2 – 70-mm long abrasive with a grain size of ~ 700 µm along the perimeter of the inlet; 3 – abrasive and grid (wire diameter 1.1 mm, grid spacing 6 mm); 4 – abrasive and grid with strips attached to the grid points (strips were made of 20-µm thick polyethene film and had the length of 30 mm and the width of 5 mm).

Different arrangements provided different turbulence intensities at the channel entrance.

A typical flow turbulization arrangement is demonstrated in figure 2. The figure shows the inlet to converging section without the smoothly shaped part. In fact, each subsequent arrangement is obtained by adding one more element to the previous one. Thus, the configuration demonstrated in the photo (figure 2) is the arrangement no.4.
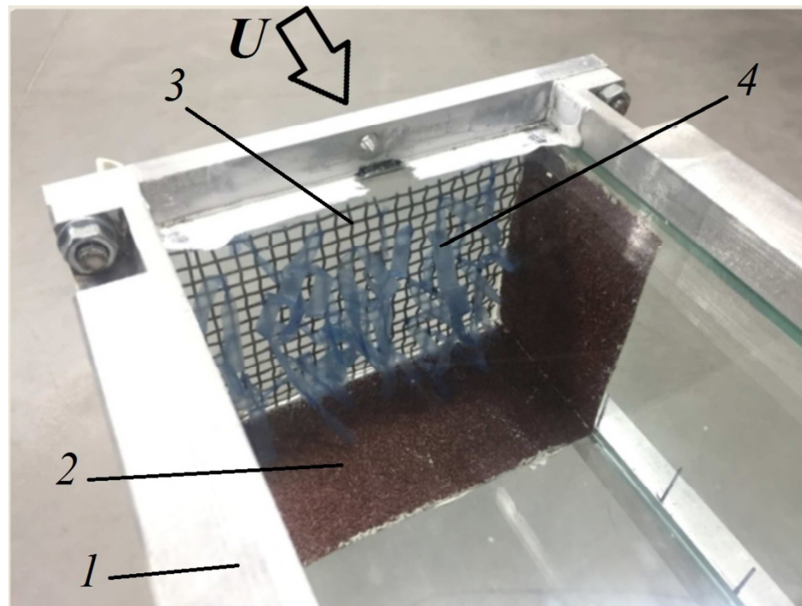
Figure 2. Inlet to the converging section: *1* – channel; *2* – abrasive; *3* – grid; *4* – strips.

Heat transfer measurements were performed using the technique that allows heating the surface with simultaneous estimation of its local temperatures from measurements of the respective electrical resistances of the heating elements. Detailed description of the method was provided in [7].

The wall was heated by electric current within the gradient section of the channel starting from its inlet according to the boundary conditions of the second kind, q=const.

3. Results and discussion

Initially, the hot-wire measurements (position *6* in figure 1) estimated the turbulence intensity of the flow at the distance of 100 mm from the channel inlet for all the considered arrangements of turbulence promotion. Smooth channel (no turbulence promotion) and abrasive yielded an estimate of Tu=2–9%. Identical turbulence intensities were attributed to the fact that wall roughness promotes turbulence first and foremost in the boundary layer. However the measurements were performed in the flow core. Abrasive together with grid increased turbulence up to the level of Tu=9–31%. Addition of strips further promoted turbulence intensity (Tu=13–45%).

The bulk velocity at the inlet of converging channel was U_0=3.7 m/s, and Reynolds number Re = $U_0 H_0/\nu$ = 24800. Subscript "0" refers to the inlet of gradient section of the channel. The respective values for the diverging channel: U_0=3.0 m/s, and Re=12600.

Distribution of heat transfer coefficients was estimated from experimental data along the section length (see figure 3 where dashed line shows the limits of the gradient part). Heat transfer in both the converging (figure 3, *a*) and diverging (figure 3, *b*) channels exhibites overall descending behavior. This is due to the fact that the wall was heated starting from the inlet of the converging channel, and thermal entrance length developed there.

As far as converging channels are concerned, augmentation of turbulence intensity in the smooth channel using different techniques demonstrated negligible effect on convective heat transfer (figure 3, *a*). The increase in turbulence intensity resulted predominantly in heat transfer enhancement. The regime with maximum turbulization of flow using the abrasive, grid and strips was an exception. Heat transfer deterioration was pronounced here if compared to other regimes with promoted turbulence. Possible reasons are given below. First, forced promotion of turbulence at the channel inlet makes the turbulence intensity decline further downstream approaching its equilibrium value. Second, favorable pressure gradient contributes to the flow laminarization trend [1]. Third, large-scale turbulization of flow (by the strips) can make velocity profiles more uniform (fuller) and reduce turbulence intensity,

even leading to flow laminarizarion. This is the effect that was revealed in experiments and described in [6] using, among others, DNS. Moreover, the laminarization of turbulent water flow in a circular pipe was achieved in [6] using a variety of turbulence promoters: honeycomb, near-wall blowing, mechanical mixing. In our study, we probably observed a combined effect of two factors, i.e. strong mixing (turbulization) of flow and favorable pressure gradient. On the whole, turbulence promotion led to heat transfer enhancement of up to 15% if compared to the flows without turbulizing effects.

Somewhat different pattern of heat transfer was observed in the diverging channel (figure 3, *b*). Distribution of heat transfer coefficient in the smooth channel was non-monotonic. This pattern is typical of alternating laminar (at the beginning) and turbulent (further downstream) boundary layer on the wall. Forced flow turbulization led to heat transfer enhancement at the channel entrance (laminar part) by the factor of up to 1.5 in terms of local values. Maximum turbulization resulted in the highest heat transfer within this section. The patterns of heat transfer coefficient became monotonic. Heat transfer coefficient in the second part of the diverging channel (originally turbulent) did not change much. In general, distributions of heat transfer coefficients in diverging channels with a small angle ($\varphi=5°$) featuring no flow separation were in qualitative agreement with the ones in zero-gradient flows. The difference here was probably in earlier laminar-turbulent transition.
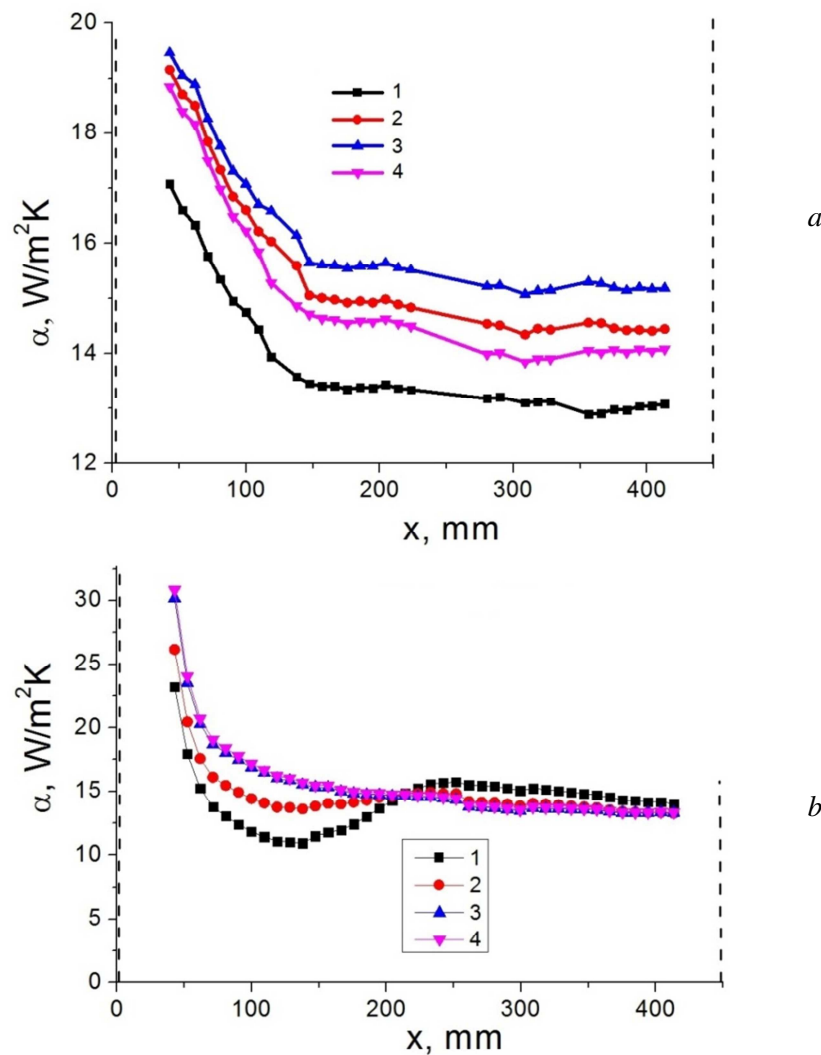
Figure 3. Heat transfer coefficient on the wall of converging (*a*) and diverging (*b*) channels: *1* – smooth channel; *2* – abrasive; *3* – abrasive+grid; *4* – abrasive+grid+strips.

4. Conclusions

It has been shown experimentally that promoted turbulence intensity generally leads to heat transfer enhancement in gradient flows. Heat transfer patterns in channels with favorable and adverse pressure gradients exhibit their distinct features. Heat transfer in the converging channel with large-scale turbulization by the abrasive, grid and strips appears to be somewhat lower than in the case of moderate turbulization. This is consistent with flow laminarization due to a number of factors (decrease of turbulence intensity down to its equilibrium value further downstream, favorable pressure gradient, and more uniform velocity profiles due to large-scale mixing).

Thus, the experiments have yielded non-monotonic correlation between heat transfer in the channel with favorable pressure gradient and turbulence intensity at the channel inlet. In the channel with adverse pressure gradient, heat transfer grows with turbulence intensity up to a certain level.

Acknowledgments

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