INVESTIGATION OF THE TRANSFER OF HEAT AND OF HYDRODYNAMIC RESISTANCE IN THE TURBULENT FLOW OF A GAS IN THE FIELD OF A LONGITUDINAL PRESSURE GRADIENT OF VARIABLE SIGN. II

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We present results from an investigation into the transfer of heat and into the hydrodynamic resistance in the case of the turbulent flow of a gas through a channel formed by an alternating sequence of plane nonsymmetric diverging and converging sections of diffusers with convergence angles of 12° in the range Re = $(10\text{-}80) \cdot 10^3$.

This paper is a natural development of [1] and is devoted to a study of the transfer of heat and the resistance in the case of constant negative pressure gradients on the basis of the effective duration and absolute magnitude.

The gas flowed through a channel formed by an alternating sequence of flat nonsymmetric diverging and converging diffuser sections with $\varphi = 12°$, formed by two copper plates (flat and shaped) with h = 300 mm and l = 960 mm. The edges of the shaped plate were rounded off.

The variation in the positive pressure gradients with respect to operational duration and absolute magnitude was achieved by altering the expansion angles of the diffusers. We studied the diffusers. We studied the transfer of heat and the resistance for the flow of a gas through channels formed by diverging–converging sections in a ratio of 1 : 1 with $\gamma = 12°$ (b = c = 40 mm), a diverging–converging section in a ratio of 2 : 1 with $\gamma = 6°$ (b = 80 mm, c = 40 mm), and a diverging–converging section in a ratio of 3 : 1 with $\gamma = 4°$ (b = 120 mm, c = 40 mm).

With an increase in the ratio of the diverging–converging sections (from 1 : 1 to 3 : 1) the negative pressure gradients do not change, while the positive gradients gradually increase in terms of operative duration, simultaneously diminishing in terms of absolute magnitude (i.e., in terms of effective intensity). The latter circumstance clouds the picture and, of course, makes difficult the evaluation of the relative effect of pressure gradients of different signs.

To determine the effect on heat transfer and resistance as a consequence of a simultaneous change in the positive and negative pressure gradients in terms of absolute magnitude, we varied the distance a, with respective values of 47.7 mm, 33.3 mm, and 16.8 mm. For the channel formed by a diverging–converging section with a ratio of 1 : 1 and $\gamma = 12°$, the reduction in the distance a indicated not only an increase in the pressure gradients, but also a change in the flow regime [1].

The experimental installation, the measurement techniques, and the method for processing the experimental data are described in [1].

The results of the investigation into the transfer of heat and resistance are shown in Fig. 1.

Examination of the heat-transfer data for the channel as a whole shows that for all values of a the increase in the ratio for the diverging–converging section of the channel leads to a reduction in heat-transfer intensity. This effect can be explained if we assume that for a given diffuser expansion ratio the more

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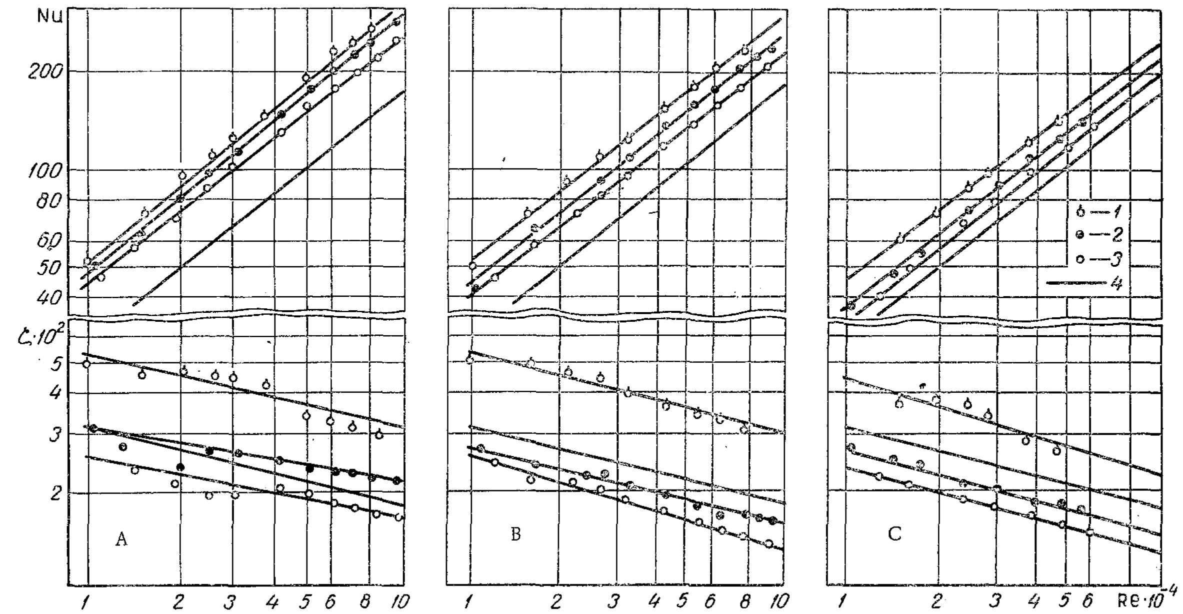

Fig. 1. The values of Nu and ζ as functions of Re for a = 47.7 mm (A), 33.3 mm (B), and 16.8 mm (C): 1, 2, 3, 4) respectively, the channels of the diverging–converging sections of the diffuser, with the ratios 1 : 1, 2 : 1, and 3 : 1, and a channel with a constant cross section over the length.

intensive turbulization corresponds to a greater expansion angle, i.e., to a greater pressure gradient.

Each of the channels exhibits the highest heat-transfer intensity at a = 47.7 mm. As the distance a is reduced, the negative gradients increase in absolute magnitude, thus intensifying the degeneration of the turbulence and the effect of the flow in the converging section on the flow in the diverging section. In all cases, the heat-transfer intensity is substantially greater (by a factor of approximately 1.86-1.12) than in the case of flow through a rectilinear channel with a cross section that is constant over the length ($Nu = 0.018\ Re^{0.8}$).

The nonsymmetrical nature of the flow has a significant effect on heat-transfer intensity. The average values of Nu for the shaped and flat plate in channels formed by a diverging–converging section with a ratio of 1 : 1 with γ = 12°, 2 : 1 with γ = 6°, 3 : 1 with γ = 4°, respectively, yield ratios of 1.4, 1.25, and 1.15.

The experimental heat-transfer data are well described by the relationship

$$Nu = A\,Re^{n}.$$

The values of the coefficients A and n are shown in Table 1.

Examination of the resistance data (see Fig. 1) shows that with an increase in the ratio of the diverging-to-converging section there is a reduction in ζ. This result is quite natural, since the generation of turbulence in the diverging section diminishes, whereas the role of the flow in the converging section increases.

For each of the channels, with a reduction in a, we have a reduction in ζ, which is a consequence of the increase in the negative pressure gradients and, consequently, a consequence of the increased effect of the flow in the converging section on the flow in the diverging section.

For channels formed by diverging–converging sections in a ratio of 2 : 1 with γ = 6° (with the exception of the flow in the case of a = 47.7 mm) and with a section in the ratio of 3 : 1 with γ = 4° the values of ζ are smaller than in the case of flow through a rectilinear channel with a cross-section constant over the length ($\zeta = 0.3164\ Re^{-0.25}$).

In these tests we noted the substantial effect of heat transfer on resistance. Thus, the value of ζ in the heating of a gas for channels formed by diverging–converging sections with ratios of 1 : 1 and γ = 12°,

TABLE 1. Values of the Coefficients

a, mm	47,7			33,3			16,8		
Diverging – converging section	1:1	2:1	3:1	1:1	2:1	3:1	1:1	2:1	3:1
Coefficients A and n									
A	0,049	0,028	0,038	0,041	0,032	0,042	0,040	0,026	0,026
n	0,760	0,805	0,765	0,770	0,780	0,745	0,760	0,790	0,780
Coefficients B and m									
B	0,460	0,170	0,150	0,620	0,310	0,400	0,680	0,290	0,280
m	0,235	0,180	0,195	0,270	0,260	0,300	0,300	0,260	0,270

TABLE 2. Values of the Coefficients k

a, mm	47,7			33,3			16,8		
Diverging – converging section	1 : 1	2 : 1	3 : 1	1 : 1	2 : 1	3 : 1	1 : 1	2 : 1	3 : 1
$Re=10\cdot10^3$	7,00	4,65	4,05	7,70	4,40	4,55	7,00	5,10	4,95
$Re=80\cdot10^3$	6,86	4,85	4,45	6,90	4,10	4,15	6,35	4,55	4,45

TABLE 3. Results from Comparison of the Degree of Efficiency for Channels Formed by Diverging–Converging Sections

a, mm	47,7			33,3			16,8		
Diverging – converging section	1 : 1	2 : 1	3 : 1	1 : 1	2 : 1	3 : 1	1 : 1	2 : 1	3 : 1
K_N									
$Re=10\cdot10^3$	0,19	0,16	0,19	0,26	0,22	0,25	0,30	0,36	0,46
$Re=80\cdot10^3$	0,26	0,18	0,37	0,30	0,27	0,33	0,37	0,36	0,56
K_F									
$Re=10\cdot10^3$	0,49	0,48	0,51	0,57	0,54	0,58	0,60	0,67	0,71
$Re=80\cdot10^3$	0,58	0,55	0,62	0,62	0,60	0,79	0,69	0,67	0,77
K_Q									
$Re=10\cdot10^3$	1,60	1,68	1,64	1,49	1,56	1,48	1,44	1,36	1,26
$Re-80\cdot10^3$	1,52	1,65	1,48	1,44	1,52	1,39	1,34	1,36	1,26

2 : 1 with γ = 6°, 3 : 1 with γ = 4° increased by factors of 1.2, 1.35, and 1.45, respectively, as compared to isothermal flow.

The data on resistance are approximated by the relationship

$$\zeta = B\,\mathrm{Re}^{-m}.$$

The values of the coefficients B and m are given in Table 1.

For these flows we found pronounced disruption of the Reynolds analogy in favor of heat transfer. Table 2 shows the values of the proportionality factor k in the equation:

$$\mathrm{St} = \frac{\zeta}{k}.$$

Let us recall that in the gradient-free flow of an ideal gas (Pr = 1) k = 8. As we can see, the greatest disruption of the analogy is found for channels formed by diverging–converging sections with a ratio of 2 : 1 with γ = 6° and 3 : 1 with γ = 4°.

The pronounced disruption of the Reynolds analogy in favor of heat transfer is apparently explained by the effect of the negative pressure gradients. The turbulent vortices which are generated in the diverging sections, degenerating in the converging sections, behave as free-turbulence vortices [1]. The effect of the diverging sections in weakened because of penetration into these sections of rarefaction waves from the converging sections. In addition, it is possible that the flow subject to the effect of rather strong negative pressure gradients may become laminar in some portion of the diverging section.

Comparison of the experimental data on heat transfer and resistance for channels formed by diverging–converging sections with the theoretical data for a rectilinear channel with a cross section that is constant over the length – a comparison which was performed in accordance with the method of comparative evaluation of convection heating surfaces [2] – shows the high degree of efficiency for these channels (Table 3).

In conclusion, we stress that all of the results cited in the article (the formulas, the conclusions as to the disruption of the Reynolds analogy, and the comparison) were derived by making the equivalent diameter of the inlet section of the diverging portion of the diffuser the decisive dimension. Thus, in this comparison we selected the straight channel with a cross section constant over the length as the basic channel, and we assumed the cross section to be equal to the inlet cross section of the diverging portion of the diffuser. This choice for the basic channel is, of course, conditional in nature. With a comparative study of flows in channels with variable cross sections over the length, the characteristic dimension may be the equivalent diameter of the cross section, in which the velocity of motion is equal to the average velocity of real flow. This makes it possible to compare flows by referring the pressure differences to the average kinetic energy of the flow.

NOTATION

φ	is the convergence angle of the converging portion of the diffuser;
h	is the height of the plate;
l	is the length of the plate;
γ	is the expansion angle for the diverging portion of the diffuser;
b	is the diffuser length;
c	is the length of the converging portion of the diffuser;
a	is the distance between the flat and the shaped plate;
Nu	is the Nusselt number;
Re	is the Reynolds number;
ζ	is the coefficient of hydrodynamic resistance;
St	is the Stanton number;
K_N, K_F, K_Q	are, respectively, the ratios of power consumption (at fixed heat flow and surface), of the surfaces (for fixed heat flow and power consumption), and of heat flows (at fixed power consumptions and surface) for the channel formed by a diverging–converging section and a channel with a cross section that is constant over the length.

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