FRICTION IN THE PRESEPARATION ZONE AND HEAT TRANSFER IN THE SEPARATION ZONE OF A TURBULENT BOUNDARY LAYER IN THE FLOW OF HEATED AIR IN A DIFFUSER WITH COOLED WALLS

P. N. Romanenko, A. N. Oblivin, and L. G. Shelegova

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The friction in the preseparation zone and the heat transfer in the separation zone of turbulent boundary layer have been experimentally investigated for flows of heated air in diffusers with cooled walls. An experimental method of determining the separation point is proposed together with a criterial equation for calculating the heat transfer in the separation zone.

The calculation of the separation point of a turbulent boundary layer has been the subject of a number of studies ([1–4], etc.). An analysis of these studies shows that a number of methods have not been experimentally evaluated owing to the lack of reliable test data.

In connection with the calculation of the separation point of a laminar boundary layer, it has been established that heat transfer between the wall and the moving flow is a complicating factor which may delay (cooling effect) or promote (heating effect) separation.

We have experimentally investigated the friction and heat transfer associated with the flow of heated air in diffusers with cooled walls with separation of the turbulent boundary layer. In generalizing the experimental data, we also employed the data of [5, 6].

The experiments covered the range

$$\mathrm{Re} = 1.7\cdot 10^5 - 9.45\cdot 10^5\,;\quad \bar{T}_w = 0.5 - 1.$$

The axisymmetric diffusers exhibited divergence angles of 8, 12, and 16°.

The experiments were performed in the intermittent wind tunnel described in detail in [5]. A feature of the experimental zones is the large number of control points, which made possible the more accurate fixing of the position of the separation point. The diffusers had cooling jackets divided longitudinally into sections. The diffuser walls were water cooled. Each section had its individual cooling system. There were 13 sections in each diffuser.

During the experiment, we measured the velocity and temperature distribution over the cross section of the boundary layer, the wall temperature, and the static pressure at the wall and in the flow in the control sections of the experimental zone. The location of the separation point was determined from the shape of the velocity profile and the presence of reverse flow near the wall.

From the measured velocity and temperature profiles, we calculated the integral characteristics of the boundary layer (up to the separation point): the momentum thickness θ; the displacement thickness δ^*; and the energy thickness φ.

From the measured rates of flow of the cooling water and its temperature rise in each section, we calculated the mean heat flows within a given section and then determined the specific heat flow to the wall $q_w = Q/F$.

The heat fluxes calculated from the balance were used to determine the Stanton number. The zones were investigated up to the point of boundary layer separation, so that the physical features of the separation flow in the diffusers could be fully established.

It has been shown that the longitudinal pressure gradient has only a slight effect on the temperature distribution upstream from the separation point. The temperature fields are conservative with respect to a positive longitudinal pressure gradient and are deformed only slightly even in the separation zone, where the velocity profile changes very sharply. This can be partly attributed to the fact that the longitudinal pressure gradient does not enter directly into the energy equation, whereas it plays a controlling part in the equation of motion.

In generalizing the experimental data, we used the so-called Reynolds analogy factor

$$k = \frac{c_f}{2\,\mathrm{St}}\,, \tag{1}$$

where c_f is the coefficient of friction.

In Fig. 1 the modified Reynolds analogy factor is shown as a function of the longitudinal pressure gradient.

The experimental relation obtained is satisfactorily described by the expression

$$k = \left(1 - \frac{\Gamma}{\Gamma_{cr}}\right)^4 \mathrm{Pr}^{0.43}, \tag{2}$$

where $\Gamma = (\theta/\rho_1 U_1^2)(dU_1/dx)\,\mathrm{Re}_\theta^{0.25}$ is the form parameter of the pressure gradient; $\Gamma_{cr} = (\theta_0/\rho_1 U_1^2)\times$ $\times\,(dU_1/dx)\,\mathrm{Re}_{\theta_0}^{0.25}$ is the form parameter of the longitudinal pressure gradient calculated from the momentum thickness θ_0 in the immediate vicinity of the separation point; U_1 is the velocity outside the boundary layer within a given section of the diffuser channel.

As may be seen from expression (2), when $\Gamma \to 0$ and $\mathrm{Pr} = 1$, $k \to 1$.

As $\Gamma \to \Gamma_{cr}$, $k \to 0$, which corresponds to the physical model of a flow with a positive longitudinal pressure gradient.

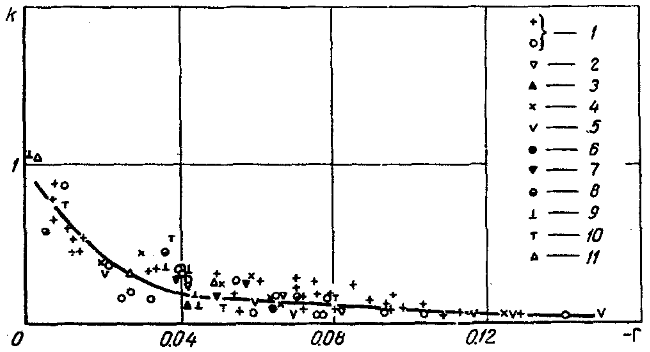

Fig. 1. Reynold's analogy factor as a function of the form parameter of the longitudinal pressure gradient: 1) $\beta = 12°$, $\mathrm{Re}_0 = (1.9-5.5) \cdot 10^5$; (2–11) $\beta = 16°$; 2) $\mathrm{Re}_0 = 0.91 \cdot 10^4$; 3) $1.14 \cdot 10^5$; 4) $1.88 \cdot 10^5$; 5) $1.95 \cdot 10^5$; 6) $2.41 \cdot 10^5$; 7) $2.44 \cdot 10^5$; 8) $2.97 \cdot 10^5$; 9) $3.08 \cdot 10^5$; 10) $3.74 \cdot 10^5$; 11) $3.84 \cdot 10^5$.

To use this relation to determine Γ_{cr}, it is necessary to find the functional dependence of Γ_{cr} on the Reynolds number and on the temperature factor, since at a fixed value of dp/dx the position of the separation point depends on these quantities.

In Fig. 2 we have plotted the experimental relation between Γ_{cr} and $(\mathrm{Re}_{\theta_0}, \bar{T}_w)$. This relation is satisfactorily described by the expression

$$\Gamma_{cr} = \frac{a - \lg \dfrac{\mathrm{Re}_{\theta_0}^{0.5}}{\bar{T}_w^{0.5}}}{b}, \tag{3}$$

where $\mathrm{Re}_{\theta_0} = U_1\theta_0/\nu_1$; $a = 1.88$; $b = 2.7$.

Expression (3) reflects the physical model of flow under the conditions in question.

Thus, as the cooling rate increases, separation is delayed, which corresponds to the conclusions of [6, 7] relating to a laminar boundary layer.

Using (2) and (3), we can write

$$k = \left[1 - \frac{(-\Gamma)}{\left(a - \lg \dfrac{\mathrm{Re}_{\theta_0}^{0.5}}{\bar{T}_w^{0.5}}\right) \Big/ b}\right] \mathrm{Pr}^{0.43}. \tag{4}$$

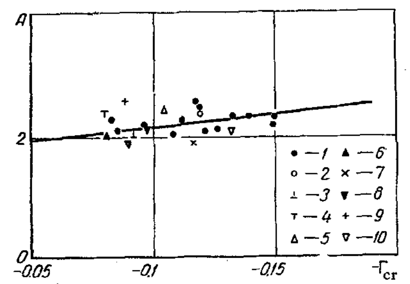

Fig. 2. Relation between Re_{θ_0}, $\bar{T}_w$, and Γ_{cr} ($A = \lg(\mathrm{Re}_{\theta_0}^{0.5}/\bar{T}_w^{0.5})$: 1) $\beta = 12°$, $\mathrm{Re}_0 = (1.9-5.5) \cdot 10^5$, (2–10) $\beta = 16°$; 2) $\mathrm{Re}_0 = 1.01 \cdot 10^5$; 3) $1.12 \cdot 10^5$; 4) $1.14 \cdot 10^5$; 5) $1.95 \cdot 10^5$; 6) $2.17 \cdot 10^5$; 7) $2.30 \cdot 10^5$; 8) $2.58 \cdot 10^5$; 9) $2.97 \cdot 10^5$; 10) $3.00 \cdot 10^5$.

At the separation point,

$$\frac{-\Gamma b}{a - \lg \dfrac{\mathrm{Re}_{\theta_0}^{0.5}}{\bar{T}_w^{0.5}}} = 1. \tag{5}$$

Thus, condition (5) makes it possible to calculate the separation point of a turbulent boundary layer by successive approximations, using the author's published work [5] on the calculation of friction in diffusers with cooled walls.

No less important a problem is the calculation of the heat transfer in the separation zone and the subsequent eddy zone in diffusers.

Whereas, up to the separation point of the turbulent boundary layer, it is possible to use the Prandtl–Kármán mixing length hypothesis and obtain a closed method of calculating the friction and heat transfer, using the integral relations of momentum and energy and introducing additional experimental relations, in the separation zone this approach is not applicable.

We have made an experimental investigation of the heat transfer in the separation zone of a turbulent boundary layer.

In Fig. 3 the variation of the heat-transfer coefficient along the channel is given for two regimes.

It is interesting to note the increase in the intensity of heat transfer at the separation point and its subsequent decline in the separation zone of the turbulent boundary layer.

This was previously noted by Leont'ev in connection with separated flows in nozzles with $M > 1$. Obviously, at the separation point, vortex formation sharply reduces or totally destroys the dynamic and thermal boundary layer and reduces the thermal resistance of the layer, at the same time increasing the flow turbulence, which causes an intensification of the molar heat transport.

The experimental data were generalized in criterial form.

The effect of the longitudinal gradient was taken into account by introducing the Euler number

$$\mathrm{Eu} = \frac{\Delta p}{\rho_0 U_0^2}.$$

In Fig. 4 we have plotted the relation

$$\mathrm{St} = f(\mathrm{Re}_x;\ \mathrm{Eu};\ \bar{T}_w),$$

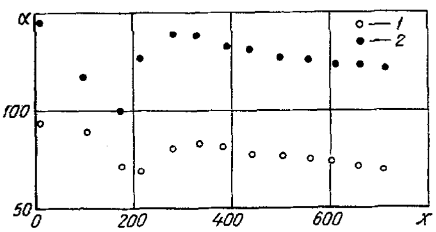

Fig. 3. Distribution of local values of the heat-transfer coefficient (W/m^2 · deg) along the length of the diffuser (mm): 1) $\beta = 16°$; $\mathrm{Re}_0 = 2.41 \cdot 10^5$; 2) $\beta = 16°$; $\mathrm{Re}_0 = 3.84 \cdot 10^5$.

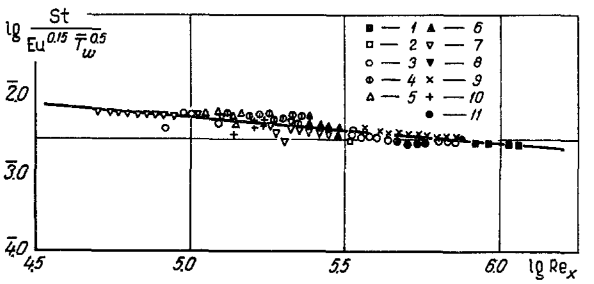

Fig. 4. The dependence of $St/Eu^{0.15}\bar{T}_w^{0.5}$ on Re_x: 1) $\beta = 12°$; $Re_0 = (1.9-5.5)\cdot 10^5$; (2–11) $\beta = 16°$; 2) $Re_0 = 0.91\cdot 10^4$; 3) $1.01\cdot 10^5$; 4) $1.88\cdot 10^5$; 5) $2.30\cdot 10^5$; 6) $2.58\cdot 10^5$; 7) $2.97\cdot 10^5$; 8) $3.00\cdot 10^5$; 9) $3.08\cdot 10^5$; 10) $3.84\cdot 10^5$; 11) $4.20\cdot 10^5$.

where $Re_x = xU_1/\nu_1$; x is the linear dimension along the generatrix of the diffuser channel.

The experimental curve in Fig. 4 is satisfactorily described by the power law equation

$$St = 0.0082\, Re_x^{-0.422}\, Eu^{0.15}\, \bar{T}_w^{0.5}\,. \tag{6}$$

As the characteristic temperature, we have taken the temperature of the flow core.

Thus, the criterial equation (6) makes it possible to calculate the heat-transfer coefficient in the separation zone of a turbulent boundary layer within the range of the experiments conducted.

NOTATION

Δp is the static pressure drop at the wall between the initial and the following section of the channel; ρ_0 and U_0 are, respectively, the static density and the flow velocity at the diffuser inlet; β is the diffuser divergence angle; $Re_0 = \rho_0 U_0 d_0/\mu_0$; d_0 is the diameter of the diffuser inlet section; μ_0 is the dynamic viscosity of air at the diffuser inlet.

REFERENCES

1. F. K. Goldschmid, Journ. of Aircraft, 2, no. 2, 1965.
2. L. G. Loitsyanskii, Mechanics of Liquids and Gases [in Russian], GITTL, 1957.
3. L. E. Kalikhman, Turbulent Boundary Layer on a Curved Surface in a Gas Flow [in Russian], Oborongiz, 1956.
4. S. S. Kutateladze and A. I. Leont'ev, Turbulent Boundary Layer of a Compressible Gas [in Russian], Izd. SO AN SSSR, Novosibirsk, 1962.
5. A. I. Leont'ev, A. N. Oblivin, and P. N. Romanenko, PMTF, no. 5, 1961.
6. C. B. Cohen and E. Reshotko, NACA Report, 1294, 1956.
7. K. Illingworth, Mekhanika 1, no. 29, 1955.

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