INFLUENCE OF JET TURBULENCE ON FLOW IN THE BOUNDARY LAYER NEAR THE WALL

I. A. Belov, G. F. Gorshkov,
V. S. Komarov, and V. S. Terpigor'ev

UDC 532.526

Results are presented of an experimental investigation of the interaction of a subsonic axisymmetric jet, within the initial section, and a flat plate mounted parallel to the jet axis. Relations are obtained for the mean and fluctuating velocities in the wall boundary layer, and the friction stress on the plate is also given.

The level of turbulence in the oncoming stream has an appreciable influence on the nature of the flow in the wall boundary layer of a plate. The influence can be twofold [1]: firstly, the presence of fluctuations in the external flow changes the position of the point of transition from laminar to turbulent flow, and secondly, there is a change in the heat and mass transfer characteristics, which leads to a change in the local values of friction stress and heat flux at the wall.

The determination of these important engineering characteristics of the interaction of the flow with a flat plate can be determined by solving the boundary layer equations in the Reynolds form. In this case, to determine the relationship between the fluctuating characteristics and the mean, one can hardly use available hypotheses as to the effective properties of the transfer of turbulent fluid, as obtained for a turbulent boundary layer with no fluctuations in the outer flow.

Therefore, the aim of the present investigation is to study the relation between the mean and fluctuating characteristics in the wall boundary layer on a flat plate washed by a jet of turbulent fluid.

The experimental arrangement consists of a continuous wind tunnel, a traverse mechanism, a constant temperature hot wire anemometer, a Pitot microtube, and a recorder (an N-105 oscillograph). The flat plate was a micarta slab of size 1000 ×400 ×8 mm, with a wedge-shaped nose. The plate was flat over its entire surface to within 0.025 mm. The plate had an aperture of diameter 3 mm to allow the anemoneter sensor to be introduced into the plate boundary layer. Traverse of the sensor through the boundary layer was accomplished using a micrometer mechanism with a resolution of 0.01 mm.

The plate was mounted with its leading edge at the nozzle exit on the jet axis. The air stream was generated by a wind tunnel consisting of an intake collector, a fan, a settling chamber, a honeycomb section, and a subsonic nozzle inlet section. The metal intake collector had a three-stage compressor. A four-blade fan, driven by a DC electric motor, delivered 0.3 m^3/sec of air at 600 N/m^2 head. A ferro-resonant stabilizer was used to decrease the effects of mains voltage fluctuations.

The experimental investigation was conducted with a flow velocity at the nozzle exit of $\langle u_a \rangle$ =5 to 30 m/sec, the nozzle exit diameter being d_a=100 mm. The mean and fluctuating velocities were measured at 15 sections on the plate surface; the total range of variation of the flow Reynolds number $R_x = \langle u_a \rangle x/\nu$, where x is the distance from the leading edge of the plate, was 0 to $3 \cdot 10^5$. To obtain the mean and fluctuating velocity profiles in the wall boundary layer, the anemometer sensor was positioned at various distances from the wall in the range y=0.05 to 10 mm, in steps of 0.05 mm. The initial position of the sensor wire relative to the wall was determined by means of a microscope. Alignment of the plate parallel to the steam axis was set by means of static pressure taps on the plate at a number of sites on the surface. For correct alignment there was no static pressure difference over the initial section of the jet.

Leningrad. Translated from Zhurnal Prikladnoi Mekhaniki i Tekhnicheskoi Fiziki, No. 6, pp. 77-82, November-December, 1972. Original article submitted June 12, 1972.

TABLE 1

Point No.	ε_δ	$R_x \cdot 10^{-5}$	Point No.	ε_δ	$R_x \cdot 10^{-5}$	Point No.	ε_δ	$R_x \cdot 10^{-5}$
1	0.018	0.184	8	0.040	2.080	15	0.072	1.000
2	0.020	0.454	9	0.045	2.340	16	0.078	1.260
3	0.024	0.725	10	0.050	2.610	17	0.083	1.530
4	0.029	1.000	11	0.056	2.880	18	0.086	1.800
5	0.031	1.260	12	0.049	0.184	19	0.089	2.080
6	0.033	1.530	13	0.056	0.454	20	0.094	2.340
7	0.035	1.800	14	0.064	0.725	21	0.097	2.610
						22	0.100	2.880

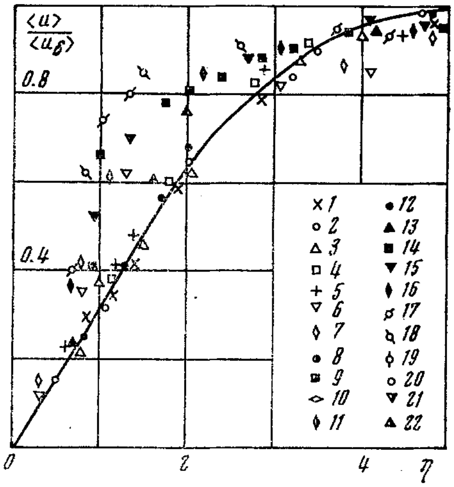

Fig. 1

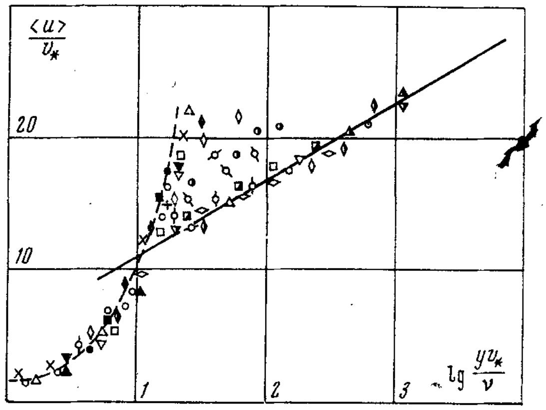

Fig. 2

The sensitive element of the anemometer probe was a gold-plated tungsten wire of length 1.5 mm and diameter 6.5 μ, which was spot-welded to the ends of nickel needles to form a U-shaped probe. The anemometer had a linearizing circuit to give an output signal proportional to the flow velocity [2]. For the chosen degree of wire overheat factor $n=2$ the anemometer time constant $M=4 \cdot 10^{-6}$ sec gave it a frequency range of 0 to 25 kHz. The experimental results were corrected for large signal distortion by the method of [3]. When the wire was very close to the wall the wall cooling effect was accounted for [4]. The anemometer was used to measure both fluctuating and mean longitudinal velocity in sections along the jet axis of symmetry; the mean velocity was also measured with the Pitot microtube.

The total measurement errors for the mean and fluctuating velocities were 5 and 15%, respectively. Because hot wire anemometer measurements are inaccurate in flows with low velocity and high turbulence level, the jets used in the experiments had a turbulence intensity at the nozzle exit of $\varepsilon_a = \sqrt{\langle u_a'^2 \rangle}/\langle u_a \rangle =$ 1.5 and 4.6%. In the latter case the converging part of the nozzle was equipped with a turbulence generating grid of cell diameter 10 mm, with blockage factor of 0.7.

Results are presented below of the measurements for a total Reynolds number $R_a = \langle u_a \rangle d_a/\nu = 4.9 \cdot 10^4$. The experiments at different R_a showed that variation of R_a over a wide range has little effect on the measured values.

From the measurements of the intensity of turbulence $\sqrt{\langle u_\delta'^2 \rangle}/\langle u_a \rangle$ in the potential core of the free stream, and including the jet interacting with the plate, it was established that the intensity of turbulence in the jet core is larger than the corresponding levels measured in the two-dimensional stream behind the turbulence generating grid in the flow over a flat plate in [5] and is of the same order as the corresponding values in free axisymmetric jets [6].

Figure 1 shows results of measurement of the distribution of mean velocity $\langle u \rangle$ through the flat plate boundary layer. Here $\eta = \sqrt{\langle u_\delta \rangle / \nu x}$ is the Blasius variable, and $\langle u_\delta \rangle$ is the velocity at the outer edge of the boundary layer (in our experiment $\langle u_\delta \rangle = \langle u_a \rangle$). The point number and the corresponding values of ε_δ and R_x are given in Table 1.

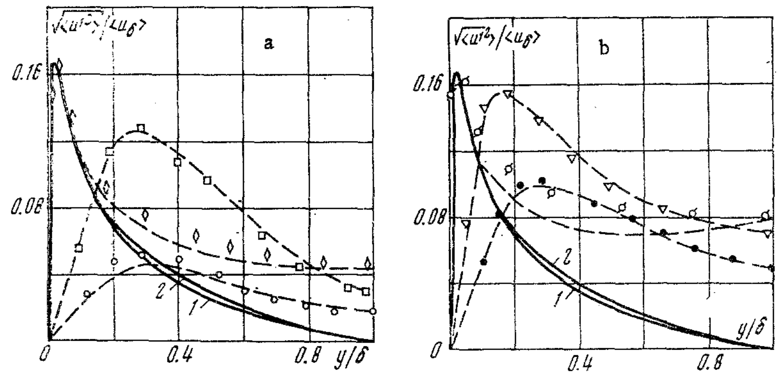

Fig. 3

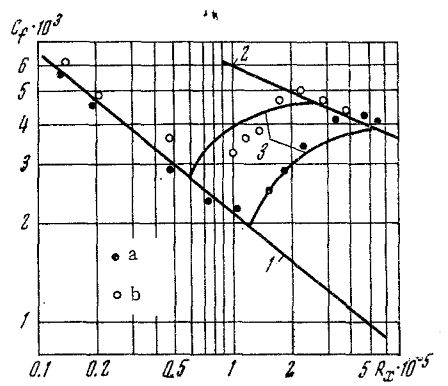

Fig. 4

The results obtained show that the mean velocity field in the wall region is similar to that near the plate leading edge and is described by the well-known Blasius profile for the laminar boundary layer (solid curve). With increase of Reynolds number R_x the velocity profile becomes fuller, which is evidence of transition of laminar flow to turbulent in the boundary layer. For $\varepsilon_a = 1.5\%$ the start of transition occurs at $R_x = 1.3 \cdot 10^5$, and for $\varepsilon_a = 4.6\%$, at $R_x = 5 \cdot 10^4$. The decrease in critical value $R_x{}^*$ in comparison with the data of [7] may be due to the increase in the intensity of turbulence along the jet axis.

In order to determine the boundary for transition to fully-developed flow turbulence the mean velocity profile was expressed in the coordinates of the distribution law for velocity in the turbulent boundary layer (Fig. 2). Here $v_* = \sqrt{\tau_w/\rho}$ is the dynamic velocity, τ_w is the friction stress at the wall, and the notation here and below corresponds to that used previously in Fig. 1.

The velocity profile in the laminar sublayer is described quite well by the relation (broken line)

$$\langle u\rangle / v_* = y v_* / \nu \tag{1}$$

In the fully turbulent region the mean velocity profile has the form (solid line)

$$\langle u\rangle / v_* = 5.75 \lg y v_* / \nu + 5.5 \tag{2}$$

It follows from the data presented in Fig. 2 that the transition to the fully developed turbulent boundary ends at $R_x = 2.5 \cdot 10^5$ for $\varepsilon_a = 1.5\%$, which is much less than the Reynolds number corresponding to the end of transition at low turbulence level [7]. It should be noted that when the degree of turbulence in the oncoming stream is relatively high ($\varepsilon_a = 4.6\%$) the scatter in the experimental values increases. This makes it difficult to construct the velocity profiles.

Figure 3 shows the distribution of longitudinal root mean square velocity fluctuations $\sqrt{\langle u'^2\rangle}$ in the plate boundary layer for the two levels of the turbulence intensity at the nozzle exit, $\varepsilon_a = 1.5\%$ (Fig. 3a) and $\varepsilon_a = 4.6\%$ (Fig. 3b). The data presented show that the velocity fluctuations in the jet penetrate deep into the plate boundary layer and create conditions for accelerated transition from laminar to turbulent flow. The maximum values of the velocity fluctuations occur in the immediate vicinity of the wall; in the laminar flow region at a distance on the order of $y/\delta = 0.2$ to 0.3, where δ is the boundary layer thickness, and in the turbulent flow region, at distance of $y/\delta < 0.05$. Figure 3 also shows the calculated velocity fluctuations $\sqrt{\langle u'^2\rangle} = l\,(\partial u/\partial y)$ using the relation for the mixing length proposed in [8] (curve 1)

$$\begin{gathered}\bar{l} = k\xi - (2k - 3k_0)\xi^2 + (k - 2k_0)\xi^3 \\ k = 0.4; \quad k_0 = 0.07, \quad \xi = y/\delta, \quad \bar{l} = l/\delta\end{gathered} \tag{3}$$

and using the Spalding relation [9] (curve 2)

$$\begin{gathered}l = ky\left[1 - \exp\left(-\frac{yv_*}{\nu A_*}\right)\right] \text{ for } \frac{y}{\delta} \leqslant \frac{\xi}{k} \\ l/\delta = \xi \text{ for } y/\delta > \xi/k, \quad k = 0.4, \quad \xi = 0.09, \quad A_* = 27\end{gathered} \tag{4}$$

The velocity fluctuation profiles constructed using the above relations for variation of mixing length through the boundary layer are in good agreement with experiment only very close to the wall. With increasing distance from the wall the divergence between theory and experiment increases. Analysis of the experimental profiles of mean and fluctuating velocity in the boundary layer yields the following approximation for the velocity fluctuations in the boundary layer:

$$\sqrt{\langle u'^2\rangle} = l_* \frac{\partial \langle u\rangle}{\partial y} + \sqrt{\langle u_\delta'^2\rangle}\ \frac{y}{\delta}\left(l_* = l\left[1-\exp\left(-2.5\,\frac{R_x}{R_x^*}\right)\right]\right) \tag{5}$$

where l is the mixing length as defined by Eq. (4), and R_x^* is the critical Reynolds number.

The results of calculations of velocity fluctuations using Eq. (5) are shown in Fig. 3 (broken lines).

Figure 4 shows the dependence of the friction factor C_f on R_x (the points denoted by the letter *a* correspond to $\varepsilon_a = 1.5\%$, and those denoted by the letter b correspond to $\varepsilon_a = 4.6\%$). Again the Blasius law for the laminar boundary layer is shown (curve 1)

$$C_f = 0.664 \cdot R_x^{-0.5} \tag{6}$$

and the relation for the turbulent boundary layer (curve 2)

$$C_f = 0.0592 \cdot R_x^{-0.2} \tag{7}$$

The friction stress was calculated directly from the measured velocities in the boundary layer. It can be seen from Fig. 4 that the friction factor is in good agreement with C_f as calculated from Eqs. (6) and (7) for the laminar and turbulent flow regions. In the transition region the behavior of the local friction factor can be described with the aid of relations analogous to the Prandtl–Schlichting formula for the mean friction factor

$$C_f = 0.455\,(\lg R_x)^{-2.58} - AR_x^{-1} \tag{8}$$

The value of A depends on the location of the point of transition of the laminar flow to turbulent, and is therefore a function of the degree of turbulence of the oncoming stream approaching the plate. The results of the present work give

$$A = \exp(9.07 - 1.59\varepsilon^{0.38}) \tag{9}$$

where ε is the intensity of turbulence at the outer edge of the boundary layer at the point corresponding to R_x^*. The results of calculating C_f from Eqs. (8) and (9) are shown in Fig. 4 for $\varepsilon_a = 1.5$ and 4.6% (curve 3).

LITERATURE CITED

1. J. Kestin, P. F. Maeder, and H. H. Sogin, "The influence of turbulence on heat transfer to cylinders near the stagnation point," Z. Angew. Math. und Phys., 12, No. 2 (1961).
2. J. Cont-Bello, Turbulent Flow in a Channel with Parallel Walls [Russian translation], Mir, Moscow (1968).
3. J. O. Hinze, Turbulence, McGraw-Hill (1959).
4. E. U. Repik and V. S. Ponomareva, "Investigation of the influence of proximity to the wall on the readings of a hot wire anemometer in a turbulent boundary layer," Izv. SO AN SSSR, Ser. Tekh. Nauk No. 3, No. 13 (1969).
5. G. H. Junkhan and G. K. Serovy, "Effects of free-stream turbulence and pressure gradient on flat plate boundary layer velocity profiles and on heat transfer," Trans. ASME, Ser. D. J. Heat Trans, 89, No. 2 (1967).
6. A. S. Ginevskii and K. A. Pochkina, "The influence of initial turbulence on the characteristics of an axisymmetric immersed jet," Inzh. Fiz. Zh., 12, No. 1 (1967).
7. G. Schlichting, Boundary Layer Theory, McGraw-Hill (1968).
8. S. S. Kutateladze, Wall Turbulence [in Russian], Izd.-vo Novosib. Gos. Un-ta, Novosibirsk (1970).
9. S. V. Patankar and D. B. Spalding, Heat and Mass Transfer in Boundary Layers, Intext Press (1971).