VARIATION OF TURBULENT FLOW PARAMETERS IN THE PRESENCE OF FLOW ACCELERATION AND DECELERATION

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The variation of the turbulent flow parameters along the main section of a cylindrical channel in the presence of flow acceleration and deceleration has been experimentally investigated. The results of the experiments are presented.

In power plant flow channels the motion of the working fluid is often essentially unsteady. The unsteadiness is most clearly expressed during startup and shutdown and in transient operating regimes. As many installations operate under high stresses, it is especially important to be able to predict the nature of the unsteadiness and its effect on the turbulent flow parameters in pipes and tubes.

An analysis of the literature on unsteady turbulent flow reveals that most of the available experimental data relate to the determination of the effect of the unsteadiness on the friction. This research was reviewed in [1, 2]. Information on the velocity structure of the flows is very limited [3-6], and there are few if any data on the energy losses associated with the acceleration and deceleration of the flow. Despite nearly 30 years of research (since the mid-1950s [7, 8]) many hydromechanical aspects of unsteady turbulent flow remain little investigated.

The present research involves the experimental determination of the evolution of the hydromechanical flow parameters during acceleration and deceleration of the flow, the object of investigation being the nature and extent of the deviation of a series of hydromechanical flow parameters from their standard and quasisteady values. The standard values are the parameters realized under conditions of steady flow along the main section of a hydraulically smooth pipe when $Re^{++} = u\delta_0{}^{++}/\nu$ = idem, while the quasisteady values are the corresponding steady values of the parameters for $Re = wD/\nu$ = idem. The results were represented in this way so that they could be used to check methods of calculating unsteady flows based, on the one hand, on the principles of the parametric theory of a turbulent boundary layer [4, 9, 10] and, on the other, on the general relations used in hydraulic calculations [1, 2]. This, in its turn, led to the introduction of different hydraulic parameters characterizing physically similar concepts and the use of different conditions (standard and quasisteady) for normalizing the quantities investigated.

The measurements were made on a closed-circuit installation consisting of the following principal elements: a dump tank/receiver; a pump with a steady-flow regulating system; an unsteady-flow controller; a forechamber; a working section; and a counterflow cooler. As the working fluid we used water. The inside diameter of the main flow channels was 50 mm. The unsteady-flow controller unit consisted of two pipes connected in parallel with regulating valves and a cutoff valve mounted in series with one of the pipes. To prevent vibration due to valve operation from being transmitted to the working section, a flexible sylphon bellows coupling was introduced between the unsteady-flow controller and the forechamber. The forechamber, designed to smooth the velocity profile at the entrance to the working section, had a diameter of 600 mm and was 1000 mm long. It was fitted with a conical stream divider and a perforated diaphragm and was smoothly connected to the working section by means of a Vitoshinskii nozzle.

The working section was made of Plexiglas and consisted of a cylindrical three-segment channel with an inside diameter of 59 mm and a total length of 65 diameters. An annular turbulence generator in the form of a 1.5-mm-thick disk with an inside diameter of 58 mm was mounted at the entrance. The results of the experiments on steady flows showed that on the flow rate

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interval investigated the working section was hydraulically smooth and the resistance coefficient was satisfactorily approximated by the Blasius formula.

In the experiments the principal parameters to be measured were: the mean flow velocity over the cross section w, the velocity on the channel axis u_0, the shear stresses at the wall τ_w, and the flow temperature. The mean velocity w was measured in the throat of the nozzle connecting the forechamber and the working section, and the velocity u_0 and the friction τ_w in a cross section 55 diameters from the entrance to the working section. The friction was determined by the Preston method. The temperature was measured with a laboratory thermometer graduated in 0.1°C mounted in the receiver. For measuring the velocities and friction we used L-shaped Pitot tubes and static-pressure tappings. The tubes for measuring the velocities were made of 2 × 0.15 mm noncorroding capillary. The Preston tube had an elliptical inlet and the dimensions of the outer and inner axes were 2.0 × 1.0 mm and 1.35 × 0.50 mm, respectively. The diameter of the static pressure probes was 1.2 mm.

In the steady regimes the dynamic head was measured with U-piezometers, and in the unsteady regimes with DMI-type differential pressure transducers coupled to ID-2I amplifiers. The signals were recorded on an N-115 loop oscillograph.

The dynamic characteristics of all the pipes employed were determined experimentally on a special rig. It was found that the time constant for the pipes with the greatest inertia did not exceed 0.02 sec.

In order to reduce the random component of the measurement error, each of the flow regimes investigated was repeated six times with subsequent averaging of the results. The maximum measurement error was not more than ±3% for the velocities and ±8% for the friction.

The integral characteristics of the velocity profile δ^+, δ^{++} were calculated from the assumption that the velocity profile could be approximated by the power-law dependence

$$u/u_0 = (y/R)^n \tag{1}$$

and from the obvious relations

$$\bar{\delta}^+ = \frac{\delta^+}{R} = \int_0^1 \left(1 - \frac{u}{u_0}\right)\left(1 - \frac{y}{R}\right) d\left(\frac{y}{R}\right) = \frac{(n+1)(n+2)-2}{2(n+1)(n+2)}, \tag{2}$$

$$\bar{\delta}^{++} = \frac{\delta^{++}}{R} = \int_0^1 \left(1 - \frac{u}{u_0}\right)\left(1 - \frac{y}{R}\right)\frac{u}{u_0} d\left(\frac{y}{R}\right) = \frac{3n}{2(n+1)(2n+1)(n+2)}, \tag{3}$$

$$\bar{\delta}^+ = 0.5(1 - w/u_0). \tag{4}$$

The parameters characterizing the friction and hydraulic losses were determined from the following expressions [2, 4]:

$$C_f = \frac{2\tau_w}{\rho u_0^2}, \quad \lambda = \frac{8\tau_w}{\rho w^2}, \tag{5}$$

$$\zeta = \lambda - (\alpha - 1)\frac{2D}{w^2}\frac{dw}{dt} - \frac{D}{w}\frac{d\alpha}{dt}, \tag{6}$$

where

$$\alpha = \frac{\int_{F_0} \rho u^2 dF}{\rho F_0 w^2} = \frac{u_0^2}{(2n+1)(n+1)w^2}. \tag{7}$$

It should be noted that there is a certain imprecision in the naming of the coefficients C_f, λ, ζ. Accordingly, we shall agree to call C_f the coefficient of friction, λ the resistance coefficient, and ζ the energy loss coefficient. The first two coefficients characterize the action of the surface friction forces and are a measure of the friction losses. The coefficient ζ determines the total losses of mechanical flow energy per unit length per unit time divided by the kinetic flow energy calculated from the flow-mean velocity. In accordance with the definition, it can be represented in the form of the relation [2, 11]

$$\zeta = \frac{8G}{\pi\rho w^3 D}, \tag{8}$$

$$G = \frac{2Q\tau_w}{R} - \rho(\alpha - 1) Q \frac{dw}{dt} - \frac{1}{2}\rho Q w \frac{d\alpha}{dt} - 2\pi\rho \frac{d}{dt}\int_0^R e r dr. \tag{9}$$

The investigations carried out in [11] lead to the conclusion that in many practical cases it is possible to neglect the last term in Eq. (9), which characterizes the variation of the kinetic energy of turbulence e with time. Accordingly, by eliminating it from (9) it is not difficult to obtain (6).

The coefficient of friction C_f is one of the principal parameters in boundary-layer calculation methods, while λ and ζ figure in hydraulic calculations.

The standard values of the coefficient of friction and the displacement thickness were determined from the expressions

$$C_{f0} = \frac{0.0213}{(\mathrm{Re}^{++})^{0.235}}, \quad \tilde{\delta}_0^+ = \frac{0.151}{(\mathrm{Re}^{++})^{0.07}}, \tag{10}$$

obtained by approximation of the results of measurements made by the authors in steady regimes and the results of processing the data provided in [12].

The quasisteady analogs of the coefficients λ and ζ were found from the relation

$$\zeta_{qs} = \lambda_{qs} = \frac{0.3164}{\mathrm{Re}^{0.25}}. \tag{11}$$

The variation with time of the measured values of u_0, w, and τ_W and the relative values of the investigated parameters

$$\Psi = \frac{C_f}{C_{f0}}, \quad \Delta = \frac{\tilde{\delta}^+}{\tilde{\delta}_0^+}, \quad \Lambda = \frac{\lambda}{\lambda_{qs}}, \quad Z = \frac{\zeta}{\zeta_{qs}} \tag{12}$$

as the flow is accelerated and decelerated is shown in Fig. 1. The information in Fig. 1 relates only to the first part of the transient processes. In both cases, at t > 1.6 sec, periodically damped oscillations with a damping time of about 8-10 sec were observed.

In order to determine the relative order of the terms on the right side of Eq. (6), in Fig. 1 we have included data on the variation of the last two terms:

$$Z_N = (\alpha - 1)\frac{2D}{\lambda_{qs}}\frac{1}{w^2}\frac{dw}{dt}, \quad Z_\alpha = \frac{1}{\lambda_{qs}}\frac{D}{w}\frac{d\alpha}{dt}. \tag{13}$$

It is easy to see that in the notation of (12), (13), Eq. (6) has the form

$$Z = \Lambda - Z_N - Z_\alpha. \tag{14}$$

Since in actual calculations it is often necessary to use the equation of motion in the form

$$\rho\frac{dw}{dt} = -\frac{dP}{dx} - \frac{2\tau_w}{R}, \tag{15}$$

Fig. 1 also contains information on the relative magnitude of the terms. For this purpose Eq. (15) was represented in the dimensionless form

$$V_t = \Pi - \Lambda, \tag{16}$$

where

$$V_t = \frac{\rho}{k}\frac{dw}{dt}; \quad \Pi = -\frac{1}{k}\frac{dP}{dx}; \quad \Lambda = \frac{1}{k}\frac{2\tau_w}{R}; \quad k = \frac{2\tau_{w,qs}}{R} = -\left(\frac{dP}{dx}\right)_{qs}. \tag{17}$$

The longitudinal pressure gradient dP/dx in (17) was calculated from Eq. (15), and its quasisteady analog from the expression

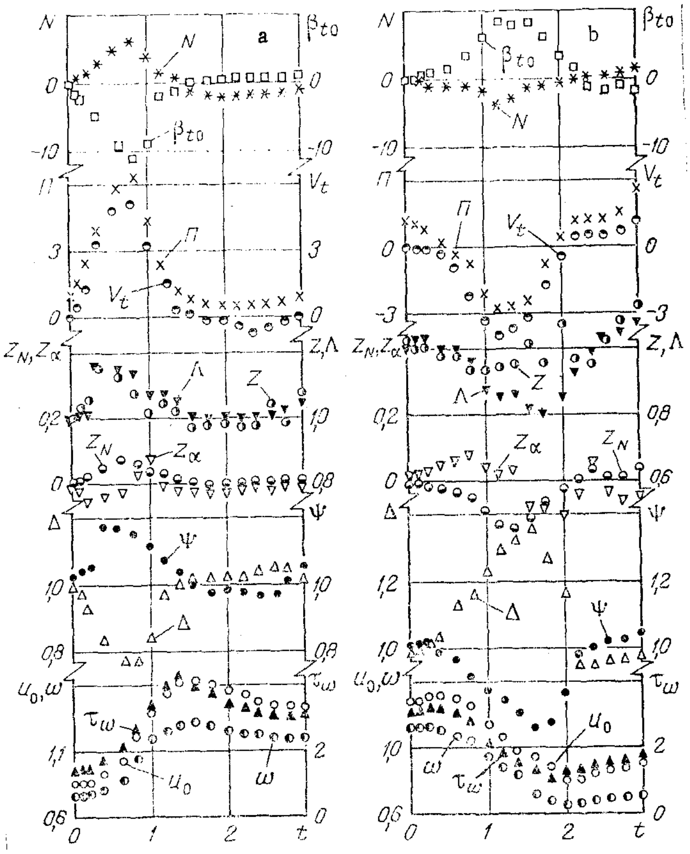

Fig. 1

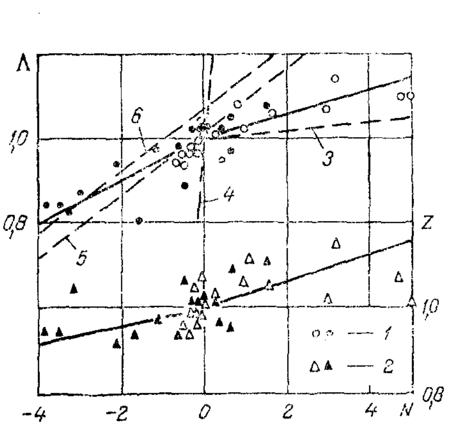

Fig. 2

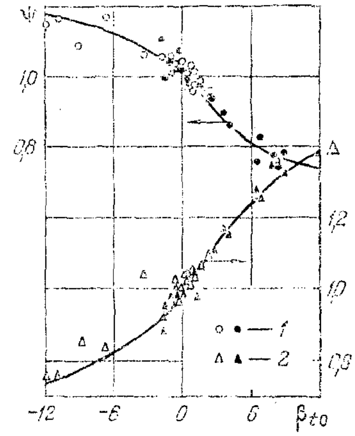

Fig. 3

Fig. 1. Variation of flow parameters in the presence of acceleration (a) and deceleration (b) of the flow. u_0, m/sec; w, m/sec; τ_w, N/m^2.

Fig. 2. Effect of unsteadiness on the resistance and energy loss coefficients: 1) Λ; 2) Z; white symbols) acceleration regime; black symbols) deceleration regime. Dashed curves represent the approximations: 3) [8]; 4) [13, 14]; 5) [2] (K = 0); 6) [2] (K = 200).

Fig. 3. Effect of unsteadiness of the coefficient of friction and displacement thickness: 1) Ψ; 2) Δ; white symbols) acceleration regime; black symols) deceleration regime; the curves represent calculations [4, 10] for $Re^{++} = 3.5 \cdot 10^3$.

$$\left(\frac{dP}{dx}\right)_{qs} = -\frac{\lambda_{qs}}{D}\frac{\rho w^2}{2}. \quad (18)$$

Moreover, Fig. 1 also shows the variation with time of the unsteadiness parameter N, most often used in processing data on the coefficient Λ [1, 2], and the parameter β_{t0}, used in unsteady boundary layer calculation methods [4, 10]:

$$N = \frac{2D}{\lambda_{qs}}\frac{1}{w^2}\frac{dw}{dt}, \quad \beta_{t0} = -\frac{2\delta}{C_{f0}}\frac{1}{u_0^2}\frac{du_0}{dt}. \quad (19)$$

An analysis of the results presented in Fig. 1 shows that, in the case of acceleration, the velocity profile is fuller than for steady flow. As a result, the displacement thickness decreases. At the same time, the wall friction and the energy losses increase, and there is a sharp increase in the absolute value of the longitudinal pressure gradient. In the case of deceleration, the qualitatively opposite variation of the principal hydromechanical parameters is observed: the velocity profile is less full, the displacement thickness increases, and the friction and energy losses decrease. An analysis of the order of magnitude of the terms of Eq. (14) shows that $|Z_N| \approx |Z_\alpha| \approx (0.05\text{-}0.2)Z$, the contribution of the terms Z_N and Z_α to the overall energy loss balance often being qualitatively opposite in character. This leads to $Z \approx \Lambda$ for acceleration and $Z \approx (1\text{-}1.2)\Lambda$ for deceleration of the flow.

The variation with time of the terms of Eq. (16) clearly illustrates the fact that under unsteady flow conditions the longitudinal pressure gradient is to a considerable extent determined by the inertia term. Situations where the longitudinal pressure gradient takes positive values ($\Pi < 0$) are possible.

Generalized information on the effect of the unsteadiness parameter on the coefficients Λ and Z is presented in Fig. 2. The experimental data on the resistance coefficient can be generalized by means of the expression

$$\Lambda = 1 + aN, \quad (20)$$

where a = 0.03 for acceleration and a = 0.05 for deceleration of the flow. It should be noted that the data of various authors for the coefficient a in Eq. (20) cover a broad range. The minimum values are: for acceleration a = 0.01 [8], for deceleration a = 0.07 [2]. The maximum value of the coefficient for unsteady flow is a = 0.64 [13, 14].

In a number of studies it is pointed out that the quantity Λ should be determined not only by the parameters containing the first derivative dw/dt, but also by the parameters that take into account the derivatives of second and even higher orders. In [2], the following complex is proposed as such an additional parameter:

$$K = \frac{D^2}{32\nu}\sqrt{\frac{1}{|w|}\left|\frac{d^2w}{dt^2}\right|}. \quad (21)$$

In view of the smallness of the parameter K in the experiments ($K \leq 200$) and the very considerable random component of the error in determining Λ, an attempt to process the experimental data in the form $\Lambda = \Lambda(N, K)$ did not lead to positive results. Clearly, for such measurements ($K \leq 200$) approximations of type (20) are the most appropriate.

In order to estimate the possible range of variation of the coefficient Λ with the parameter K, in Fig. 2 we have plotted the values of Λ for K = 0 and K = 200 calculated in accordance with the method proposed in [2].

The best correlation between the calculated and experimental values of Λ is observed for decelerated flow ($N < 0$).

Approximation of the energy loss coefficient data in the form

$$Z = 1 + bN \quad (22)$$

gives the following values of the coefficient b: for acceleration b = 0.03, for deceleration b = 0.02.

The effect of the parameter β_{t0} on the coefficient of friction and the displacement thickness is illustrated in Fig. 3. The same figure shows the results of calculating the friction law for an unsteady turbulent boundary layer by the method of [10] using the relation for the shear stress profile in the form given in [4]. The correspondence of the calculated and experimental values of Ψ and Λ can be regarded as perfectly satisfactory.

The results of the investigation show that for flow acceleration we get the following relations: $\lambda > \lambda_{qs}$, $\zeta > \zeta_{qs}$, $C_f > C_{f0}$, $\delta^+ < \delta_0^+$. Deceleration of the flow leads to the qualitatively opposite relations, the effect of unsteadiness of Λ, Ψ, and Δ being more strongly expressed for deceleration than for acceleration of the flow. However, considerable deviation of the coefficient ζ from ζ_{qs} is observed in accelerated flows.

NOTATION

P, pressure; ρ, density; ν, kinematic viscosity; u, local velocity, u_0, velocity on the pipe axis; w, the mean velocity over the cross section; τ_w, wall shear stresses; r, radial coordinate; R, pipe radius; D = 2R; y, coordinate normal to the pipe wall; y = R − r; x, longitudinal coordinate; $Q = F_0 w$, flow rate; $F_0 = \pi R^2$; δ, boundary layer thickness.

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