temperature of the medium; $\dot{\gamma} = dV/dy$, shearing rate; T^*, temperature of the isothermal plate. Dimensionless quantities: $W = V/U$, flow velocity; $\Theta = (T - T_l^*)/T_l^*$; $\vartheta = (T - T_u^*)/T_u^*$, temperature of the medium; $\xi = y/h$, vertical coordinate; ξ_1 and ξ_2, core boundaries; ξ_0, coordinate of the plane where the shearing stress vanishes; $\alpha = \mu_p U/(Ah)^{m/n}h$, $\beta_0 = \tau_0/Ah$, and $\varkappa = AUh^2/\lambda T^*$, parameters; $B = \varkappa/\alpha$, dissipative parameter; Nu_{is}, Nusselt numbers at the isothermal plate; $Sen = \tau_0 h/\mu_p U$, St. Venant–Il'yushin number; $Br = \mu_p U^2/\lambda T^*$, Brinkman number.

LITERATURE CITED

1. Z. P. Shul'man and V. F. Volchenok, "Generalized Couette flow of a nonlinearly viscoplastic liquid," Inzh.-Fiz. Zh., 33, No. 5 (1977).
2. Z. P. Shul'man and V. F. Volchenok, "Heat exchange in generalized Couette flow of a nonlinearly viscoplastic liquid," Inzh.-Fiz. Zh., 34, No. 6 (1978).
3. B. M. Smol'skii, Z. P. Shul'man, and V. M. Gorislavets, Rheodynamics and Heat Exchange of Nonlinearly Viscoplastic Materials [in Russian], Nauka i Tekhnika, Minsk (1970).

EFFECT OF SELF-OSCILLATIONS ON THE HYDRAULIC RESISTANCE OF A VORTEX TUBE

Yu. A. Knysh

UDC 532.526.4

It has been established experimentally that the hydraulic loss increases as tangential self-oscillations develop in a swirled flow.

Swirling of liquid and gas flows is widely used in modern technology as an effective means of intensifying the heat and mass exchange processes. It has been noted that the expenditure of energy on flow advancement increases with an increase in the heat exchange intensity. This effect is most strongly pronounced in the case of short vortex tubes with high vorticity at the inlet [1]. The increase in the heat exchange intensity and the resistance is usually explained by the effect of mass forces, which generate secondary flows and an elevated turbulence level. The other well-known characteristic of a swirled flow — its capacity for spontaneous excitation of intensive, regular velocity and pressure pulsations — is usually not taken into account. However, the results of many experiments indicate that self-oscillations are closely related to transport processes [2]. Thus, the highest energy exchange intensity and a considerable reduction in the throughput of a vortex tube are observed under conditions where the pulsation amplitude is at a maximum. The present article advances the hypothesis that self-oscillation processes are among the most important factors which determine the acceleration of heat exchange and the increase in the hydraulic resistance in a vortex tube. The experimental data given below refer to the interrelationship between oscillations and the hydraulic loss, and they support to a certain extent the above point of view.

The experimental simulator is shown schematically in Fig. 1. An endless-screw swirler 2 is mounted in a cylindrical tube 1, whose length is L and the radius r_0 = 8 mm. The dimensions of 10 different screws make it possible to vary the degree of flow vorticity in the range from A = 1.76 to A = 16. The vorticity parameter is calculated by means of the expression $A = \pi r_0^2 \sin\beta/F_{in}$ after Abramovich [3]. The swirler is fastened on a mobile hollow rod 3; by moving this rod, the distance L can be varied in the 5-400-mm range. If necessary, the central cavity of the tube can be made to communicate with the ambient by opening the valve 4 inside the through-passage in the rod. The sides of the tube are provided with holes 5 which allow a dye to be fed into the flow to make it visible or allow pressure and velocity data units to be mounted.

The water flow 6, which arrives from the delivery branch pipe 7 while the valve 4 is in the "closed" position, produces in the cylindrical tube beyond the swirler a hollow vortex 8, which performs combined rotary and translational motions. Liquid from the ambient is drawn

Kuibyshev Aviation Institute. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 37, No. 7, pp. 59-64, July, 1979. Original article submitted June 2, 1978.

0022-0841/79/3701-0796$07.50

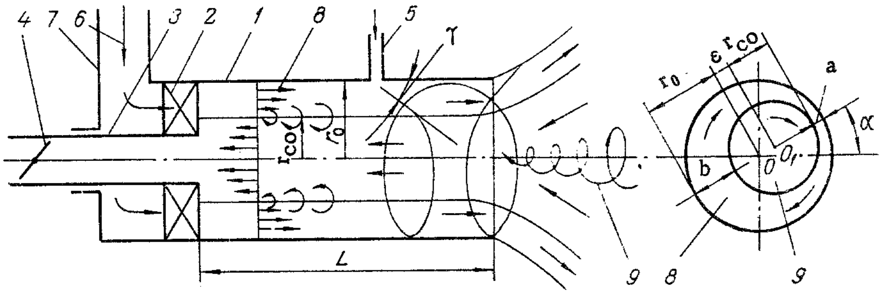

Fig. 1. Schematic diagram of the vortex tube.

into the central cavity of the vortex; the recirculating vortex core 9 is formed from this liquid. Energy and mass exchange, which sometimes acquire a strongly pronounced self-oscillatory character, occurs continuously between the basic and the recirculation vortex flows. The instability of the interaction manifests itself in the fact that the vortex core moves as a whole to the distance ε from the tube axis and performs precessional motion about it [4]. The precessing mass of the core exerts a periodic dynamic action on the basic flow. As a result, velocity and pressure pulsations, synchronous with the precession, develop in the basic flow. In correspondence with the principle of conservation of the angular momentum of circulation and the principle of continuity of flow, in places where the cross-sectional area of the flow is reduced at a certain given moment (Fig. 1, region a), the flow velocity and pressure increase relative to the mean values pertaining to the symmetric position of the vortex core. The velocity and pressure, on the contrary, decrease on the diametrically opposite side (region b). The time of one full revolution of the core's center of mass about the tube axis is equal to the period of a single oscillation of the parameters of the basic vortex.

The velocity pulsations and the precession of the core can be observed visually in a transparent tube. Small air bubbles introduced through capillary 5 or a dye delineate clearly a helical streamline, whose slope with respect to its axis is equal to γ. Under nonstationary conditions, the magnitude of the slope varies periodically in exact correspondence with the rotation angle α of the precessing core 9. The pressure and velocity data units record periodic variations of static pressure and velocity in the flow. The pressure variations have a phase lead relative to the velocity variations. In the absence of extraneous disturbances the signal has a sinusoidal shape. The oscillation amplitude is proportional to the core shift ε, while the frequency is proportional to the angular velocity of precession.

Means for amplifying or reducing the pulsation amplitude are provided in the experimental simulator in order to determine the role of self-oscillations in the rise of hydraulic resistance. Partial or complete suppression of the nonstationary interaction between the basic vortex and the core is achieved by reducing the tube length L beyond a certain limiting value or by opening the valve 4. The damping of oscillations in the first case is explained by a reduction in the length of the section of interaction between the vortex and the core. In the second case, the vortex core is formed from the liquid drawn in from the screw and moving in the direction of the screw axis. The reduction in the radial gradient of the axial velocity achieved in this manner lowers the intensity of energy and mass exchange between the vortex and the core, which reduces or completely suppresses the development of precessional self-oscillations.

Introduction of a small volume of air in the cavity of the liquid vortex also is an efficient method of eliminating self-oscillations. Since its density is lower, the air floats toward the axis and occupies the volume of the liquid core, whose mass is carried away by the basic flow beyond the confines of the tube.

It has been established that flow stabilization achieved by means of any of the above damping methods always entails a reduction in the hydraulic resistance. As an example, Fig. 2 shows typical relationships between the hydraulic resistance coefficient and the Reynolds number for self-oscillatory and stabilized flow conditions. It is evident from the diagrams that, in the presence of oscillations in the flow, the coefficient ξ is consistently higher by 10-20%. The value of ξ is calculated by means of the well-known [5] expression $\xi = 2\Delta P/\rho W_{me}^2$.

We did not succeed in establishing a single-valued quantitative interrelationship

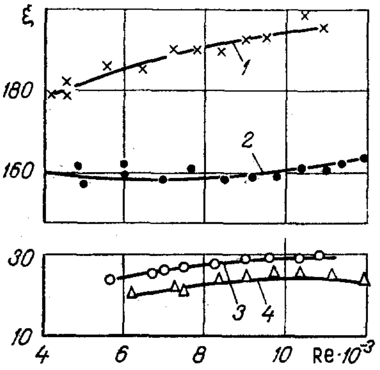

Fig. 2. Hydraulic resistance coefficient as a function of the Re number for a tube with L/d = 6 and the following flow conditions: 1) nonstationary, A = 6.07; 2) stabilized, A = 6.07; 3) nonstationary, A = 1.76; 4) stabilized, A = 1.76.

between the variation of ξ and the amplitude and frequency of oscillations. The point is that the efficiency of the action of oscillations depends to a considerable extent on the flow velocity, the degree of vorticity, the relative length of the tube, and a number of other factors. Moreover, the amplitudes of velocity and pressure pulsations along the tube are essentially unstable. The maximum value of the amplitude $\Delta P'$ is observed, as a rule, near the outlet of the tube. This was used as the reference value in processing the experimental data.

Figure 3 shows a number of characteristic relationships between the basic parameters of swirled flow and the relative length of the vortex chamber. With an increase in the length, the relative amplitude of pressure pulsations $\overline{\Delta P'} = \Delta P'/\Delta P$ first increases continuously up to a certain maximum value and then declines. The hydraulic resistance coefficient and the relative vacuum near the swirler's axis $\overline{e} = (P_a - P_{ax})/\rho W_{me}^2$ vary in a similar manner. The intensity of the reverse flow, defined as the ratio of the discharge in the flow core to the discharge in the basic vortex, $\overline{G} = G_{co}/G_v$, decreases monotonically.

Figure 4 shows the curves of these parameters as functions of the Reynolds number for a vortex tube with the length $\overline{L} = L/d = 6$ and the geometric characteristic A = 1.76. The Reynolds number is calculated with respect to the mean-discharge velocity and the tube diameter. Tests of other variants of the experimental tube have shown similar results. In the range of the investigated combinations of dimensions, an increase in the oscillation amplitude is always accompanied by an increase in the hydraulic loss. In the range of the falling branch of the amplitude curve, the coefficient of hydraulic resistance varies slightly. Our experimental results can be interpreted in the following manner.

The hydraulic resistance during the motion of a swirled flow within a cylindrical tube is composed of the loss due to friction between the liquid and the side walls of the channel and the loss due to kinetic energy dissipation during interaction between the basic vortex and the core. As was shown in [6], the relative share of the loss due to friction against the walls is relatively small. Therefore, the sharp increase in the hydraulic resistance as oscillations develop is most probably caused by a change in the character of interaction between the flows. The mutually opposite flows of the liquid in the axial direction create the necessary prerequisites for the development of an anomalously high turbulence initially at the interface between the flows. The core precession, by increasing periodically the radial gradients of the axial velocity, stimulates a strengthening of the regular, large-scale high-intensity turbulent vortices, which subsequently spread virtually over the entire cross section of the flow. Through this vortex system, a maximum amount of kinetic energy is transferred from the basic flow to the core and is then dissipated in the ambient. In this, the positive pressure gradient in the axial region of the tube (characterized by the value $e = P_a - P_{ax}$) increases continuously, while the reverse flow velocity in the core rises, which causes an increase in hydraulic resistance to the axial motion of the basic vortex.

It should be noted that the above-described methods of influencing the flow for damping

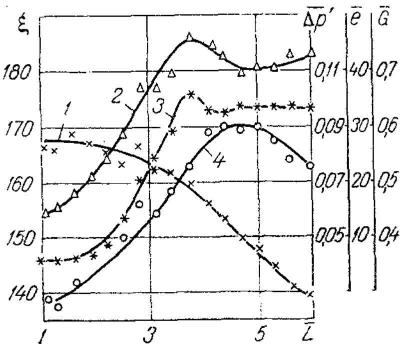

Fig. 3

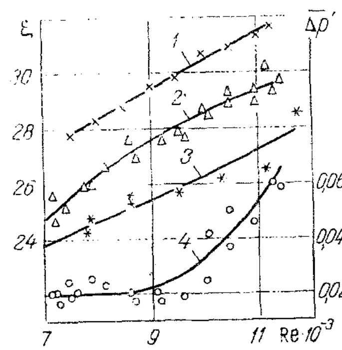

Fig. 4

Fig. 3. Relationships between the relative length of the vortex tube and the following quantities: 1) $\bar{G} = G_{co}/G_v$; 2) $\xi = 2\Delta P/\rho W_{me}^2$; 3) $\bar{e} = (P_a - P_{ax})/\rho W_{me}^2$; 4) $\overline{\Delta P}' = \Delta P'/\Delta P$; A = 6.07; Re = 8000.

Fig. 4. Relationships between the Re number and the following quantities: 1) $\bar{G}$; 2) ξ; 3) $\bar{e}$; 4) $\overline{\Delta P}'$; A = 1.76; L/d = 6.

purposes also affect the degree of vorticity. Thus, both the vorticity and the hydraulic loss can be reduced to a certain extent by increasing the length of the vortex chamber and providing an axial supply of the liquid. If the flow then remains a steady-state flow, the drop in the hydraulic resistance, as is known [6], is small. With the development of regular velocity pulsations, the hydraulic resistance displays a virtually jump-like increase by an amount proportional to the amplitude and the frequency of oscillations, i.e., oscillations and the degree of vorticity exert opposite influences on the hydraulic loss in a swirled flow when the length of the vortex chamber is varied. The effect of oscillations is predominant. In these experiments, we did not succeed in determining separately the contributions of these factors, since the development of oscillations is intimately related to changes in the most important hydrodynamic characteristics — the turbulence intensity and the distribution of the mean velocity field.

Nevertheless, on the basis of the qualitative results obtained, we reach the conclusion that oscillation damping constitutes a highly efficient means of reducing the hydraulic loss in some vortex apparatus. For instance, simultaneous supply of an amount of air through the central opening in the swirler resulted in a 10% reduction in the hydraulic resistance of a cyclone separator [4]. In another case [7], this method was used to increase the throughput of a swirl gas nozzle (by more than 15%). An efficient method of damping pulsations in the flow is to use the separating insert for cyclones described in [8], which prevents direct contact between the basic vortex and the core, thus eliminating the possibility of unsteady interaction between them. Implementation of this proposal would make it possible to reduce by 20-40% the pressure loss in a cyclone dust extractor without impairing substantially the degree of air purification.

NOTATION

L, tube length; r_0, d, radius and diameter of the tube, respectively; F_{in}, cross-sectional area of the swirler's inlet channels; r_{co}, radius of the vortex core; β, γ, and α, slope of the helical curve of the screw, streamline angle, and rotation angle of the core's center of mass, respectively; ε, shift of the core center relative to the tube axis; ξ, coefficient of hydraulic resistance; P_{in}, P_{ax}, and P_a, pressure before the tube inlet, near the axis, and in the ambient, respectively; ρ, W_{me}, density and mean-mass velocity of the liquid, respectively; ΔP, ΔP', total pressure drop and amplitude of pressure oscillations, respectively; $\bar{e}$, relative vaccuum in the axial region; Re, Reynolds number.

LITERATURE CITED

1. V. K. Shchukin, Heat Exchange and Hydrodynamics of Internal Flow in Mass Force Fields [in Russian], Mashinostroenie, Moscow (1970).
2. Yu. A. Knysh and S. V. Lukachev, in: Reports at the Eighth All-Union Acoustic Conference [in Russian], Acoustic Institute, Academy of Sciences of the USSR, Moscow (1973).
3. G. N. Abramovich, Applied Gas Dynamics [in Russian], Gostekhizdat, Moscow (1951).

4. Yu. A. Knysh and S. V. Lukachev, in: Transactions of the Kuibyshev Aviation Institute [in Russian], Vol. 67 (1974).
5. I. E. Idel'chik, Manual on Hydraulic Resistances [in Russian], Mashinostroenie, Moscow (1975).
6. Yu. I. Khavkin, Centrifugal Sprayers [in Russian], Mashinostroenie, Leningrad (1976).
7. V. T. Aniskin, Yu. A. Knysh, Yu. L. Kovylov, and V. P. Lukachev, Ground Utilization of Aircraft Engines in the People's Economy, Information Literature at an Interindustry Exhibition (VIMI) [in Russian], Vol. 1 (1975).
8. A. A. Petrov, in: Mechanical Purification of Industrial Gases [in Russian], Mashinostroenie, Moscow (1974).

EFFECT OF AXIAL CURVATURE ON AERODYNAMIC CHARACTERISTICS OF PLANAR CHANNELS

A. S. Mazo UDC 533.6.01

The effect of axial curvature on average characteristics of a turbulent flow of incompressible liquid in the gap between two coaxial cylinders is studied.

Calculation of aerodynamic characteristics of channels with cylindrical walls for laminar flow was performed in [1]. Experimental studies of turbulent flow in curvilinear channels were performed in [2-5]. Both averaged characteristics [2-5] and turbulent flow structure [3] were studied. However experimental data have been obtained only for relatively slightly curved channels ($r_m > 4.5H$). At the same time, technological applications employ highly curved channels (see, e.g., [6]).

The present study is a systematic calculation of the effect of axial curvature on the averaged characteristics of a turbulent flow in a channel with cylindrical walls, these characteristics being the velocity profile, tangent stress coefficient, and resistance coefficient, for various Reynolds numbers Re. For the sake of definiteness we will consider a stabilized flow, in which the velocity does not change along the axial coordinate. Such a flow is defined by one geometric parameter, the relative curvature r_m/H (or r_1/r_2), and in the case of an incompressible liquid, by one regime parameter, the Reynolds number Re.

It is known that curvature has a significant effect on turbulent exchange. At the present time various methods have been proposed to consider the effect of curvature on turbulent friction. In [7] an analysis and calculated comparison was made of various semiempirical turbulence hypotheses for a stabilized flow in a round channel. It was shown that the generalized theory of the Prandtl displacement path

$$\tau_t = \rho l^2 \alpha^2 |\theta| \theta, \tag{1}$$

$$\theta = \frac{du}{dr} - \frac{u}{r} \tag{2}$$

produces satisfactory results on the whole. Prandtl's formula may be obtained from the turbulent energy balance equation, if we neglect convective (absent in stabilized flows) and diffusion terms and also take $\nu_t = Cle^{-1/2}$. Since the term $\tau_t\theta$ is written exactly, this expression considers the major effect of curvature, i.e., additional generation of turbulent energy at the concave wall, and suppression of the same at the convex wall.

As is well-known (see, e.g., [9]) the displacement path length is weakly dependent on flow conditions: for a boundary layer on a plane plate and for developed flow in a tube the expressions for displacement path length practically coincide, and have little effect on Re and the axial pressure gradient. Also, as follows from physical considerations [9] and directly from measurement [3], the integral scale of turbulence (and consequently, the displacement path length) increases somewhat at the concave wall and decreases somewhat at the convex wall. However at present there are no systematic experimental data on the effect of curvature on the turbulence scale.

Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 37, No. 1, pp. 65-71, July, 1979. Original article submitted August 22, 1978.

 0022-0841/79/3701-0800$07.50