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INFLUENCE OF THE VELOCITY OF TURBULENT FLOW ON ITS INTERACTION WITH A VISCOELASTIC COATING

V. M. Kulik and S. V. Rodyakin UDC 532.526.4

It has been shown that the maximum interaction between turbulent flow and a viscoelastic coating is observed when the coating thickness is equal to one quarter of the wavelength of the longitudinal deformation of the coating surface. The amplitudes of the longitudinal and lateral deformations are equal, which must be taken into account in analyzing the mechanism of operation of a compliant coating.

The idea of controlling the boundary layer of turbulent flow by a compliant coating was successfully realized by Kramer [1]. The results of subsequent activation of the investigations were critically considered in [2]. The impossibility of reproducing successful experiments (where a friction decrease has been recorded) is basically explained by the difference in the vibrational characteristics of coatings that have not been determined in these experiments.

The second stage of investigations of compliant coatings in turbulent flow was study of disturbances formed on their surface. The possibility of V-folds developing on the coating surface if the flow velocity exceeds the velocity of propagation of shear vibrations in the coating material has been shown theoretically in [3, 4]. The amplitude of these vibrations increases to 4% of the boundary-layer thickness with the velocity of flow. For a velocity of flow 2.8 times higher than the velocity of shear waves these disturbances lose stability, and random motion of the folds, i.e., flutter, begins. These waves have been visualized in [5, 6] and a coincidence with the calculation [4] of the flow-velocity ranges where these waves are detected has been obtained. Such disturbances of the surface (static-divergence waves) result from the instability of class C, according to the terminology of Benjamin [7]. A decrease in the Reynolds stresses and an upward shift of the logarithmic part of the dimensionless velocity profile on a coating calculated from Dunkan's theory have been obtained in [8].

However, for stable vibrations of fairly large amplitude to be formed on the coating surface it is necessary to satisfy the condition $U/C_{sh} \approx 2.5$, $C_{sh} = (E_{sh}/\rho)^{1/2}$. Thus, for the flow velocity $U = 10$ m/sec and $\rho = 10^3$ kg/m^3 a coating must possess a shear elastic modulus of $E_{sh} \approx 1.5 \cdot 10^4$ Pa (E_{sh} = 310–360 Pa in the experiment of [9]), i.e., must be in a gel-like state, which is absolutely unacceptable for practical use.

An alternative approach has been proposed in [10, 11]. For a rigid coating the amplitude of vibrations of its surface under the action of turbulent pressure pulsations becomes smaller than the thickness of a viscous sublayer, i.e., the coating remains hydraulically smooth. However, the velocity of motion of the surface in flow becomes comparable to the normal component of the pulsation velocity of the flow itself. This changes the Reynolds stress in the near-wall region, which is in proportion to the correlation of the longitudinal and normal components of the pulsation velocity and can lead to a decrease in the friction on the wall itself. Experiments with such coatings were conducted under the conditions of an open water reservoir [12] and in the cavitation tunnel of Newcastle University [13], and they have shown a decrease in the hydrodynamic friction and a reduction in the levels of pressure and longitudinal-velocity pulsations and tangential stresses.

In all the experiments in which a friction decrease was recorded one has noted a strong influence of the flow velocity on the value of the effect [1, 12–14].

The dynamic compliance of coatings deformed by sensors of different diameter has been determined experimentally in [15]. The ratio of the sensor diameter d to the coating thickness H varied from 0.3 to 4. A resonance relationship between the compliance and the deformation frequency and a strong dependence of the compliance on the

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S. S. Kutateladze Institute of Thermal Physics, Siberian Branch of the Russian Academy of Sciences, Novosibirsk, Russia; email: kulik@itp.nsc.ru. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 76, No. 6, pp. 61–64, November–December, 2003. Original article submitted March 26, 2003.

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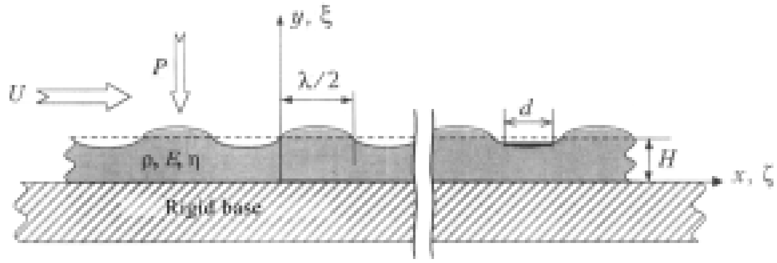

Fig. 1. Deformation of a compliant coating in the turbulent flow (on the left) and when the complex compliance is measured (on the right). $P = P_0 \exp[(i(\omega)t - 2\pi x/\lambda)]$.

ratio of d to H were established. This allowed the assumption explaining the reason for the dependence of the efficiency of the decrease in friction by coatings on the velocity. This hypothesis has been developed further in the present work.

Figure 1 shows an instantaneous pattern of deformation of a viscoelastic-material layer under the action of turbulent pressure pulsations. A two-dimensional stationary idealization of the real process is considered. The region of measurement of the dynamic compliance is singled out in the figure on the right. The maximum value of the compliance is attained when $d/H = 1.3$ [15]. The sensor diameter is somewhat smaller than half the wavelength of the vibrations formed; we have assumed that

$$(\lambda_0/2)/H \approx 2\,. \tag{1}$$

The wavelength of deformation of the surface at the resonant frequency λ_0 is related to the flow velocity by the relation $\lambda_0 = U_c/f_0$, where U_c is the convective velocity of motion of structures forming pressure pulsations on the wall with a frequency f_0, which is equal to the resonant frequency of the coating. In [16], it is shown that $U_c = (0.6–0.8)U$. Therefore, the flow velocity for which a viscoelastic coating will interact with pressure pulsations to the largest extent is equal to

$$U = \frac{4Hf_0}{(0.6–0.8)}\,. \tag{2}$$

Why $\lambda_0/2 = 2H$? To answer this question we have analyzed the amplitude of displacement of a coating ξ at different distances from the wall y; it is described by the equation [17]

$$\frac{d^2\xi}{dy^2}(1 + i\eta) = -\omega^2 \frac{\rho}{E}\xi$$

with the following boundary conditions:

$$\xi = 0 \quad \text{for} \quad y = 0\,; \quad \frac{d\xi}{dy}(1 + i\eta) = -P_0/E \quad \text{for} \quad y = H\,.$$

The results of calculation of the displacement amplitude for coating layers at different distances from the rigid base are presented in Fig. 2a. For the first resonance, which is at the frequency

$$f_0 = \frac{\sqrt{E/\rho}}{4H}\left[\frac{2\,(1+\eta^2)}{1+\sqrt{1+\eta^2}}\right]^{1/2},$$

the coating thickness is equal to one-quarter of the length of the standing wave of compressive–tensile deformation in the coating material (curve 2). The displacement of the coating is always equal to zero on the rigid base (for $y = 0$); there is a "node" of vibrations here. On the exterior surface (for $y = H$) in the case of resonance there is a "crest."

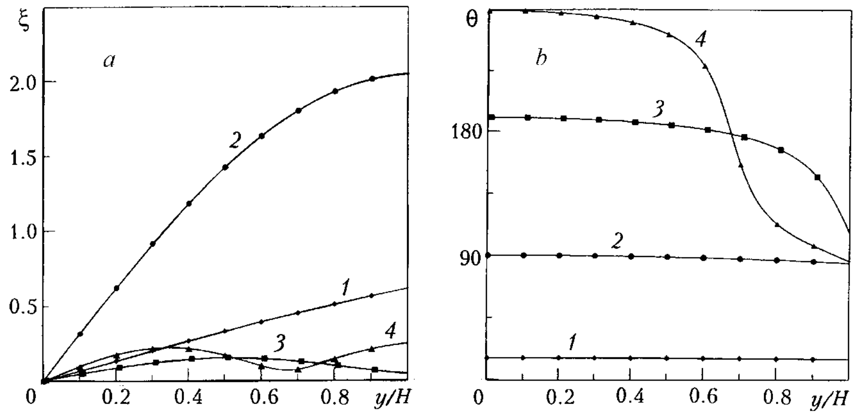

Fig. 2. Change in the deformation amplitude over the coating thickness (a) and in the phase lag of the deformation behind the pulsation pressure (b) (E = 10^6 Pa, ρ = 10^3 kg/m^3, $H = 5 \cdot 10^{-3}$ m, P = 100 Pa, η = 0.2): 1) $f = f_0/2$; 2) f_0 = 1573 Hz; 3) $2f_0$; 4) $3f_0$. ξ, μm; θ, angular degrees.

The compliance of the coating will be minimum at the frequency $2f_0$ (curve 3), since, when $y = H$, we will have the node of the standing wave and the crest will be inside the coating when $y/H \cong 0.5$. At the frequency $3f_0$ (curve 4), we have $H = 0.75\lambda_0$ and a second resonance much weaker than the first one is realized.

Figure 2b shows the phase lag of the vibrations of the coating layers behind the pulsation pressure applied as a function of the distance from a rigid base. To the first-resonance frequency (curves 1 and 2), the coating vibrates in a single phase over the entire thickness. This phase grows from 0 to 90° as the frequency increases from 0 to f_0. At frequencies higher than the first-resonance frequency, different layers of the coating vibrate with different phases (curves 3 and 4). Phase changes are the largest in the upper part of the coating for $y/H > 0.5$.

Since one utilizes rubber-like materials which are incompressible, in practice, for most of the coatings, vertical deformations (over the coating thickness) become horizontal (propagating along a coating). Therefore, shear waves and compression–expansion waves are radiating from the site of local periodic deformation. It is logical to assume that the velocities of propagation of disturbances along the coating and across it will be approximately the same near the site of local deformation; hence the wavelengths will also be equal.

A necessary requirement for the efficient interaction between a coating and turbulent flow is the attainment of the maximum deflections of its surface under pressure pulsations. Clearly, this condition is attained when the crest of the deformation wave on the coating surface is in the normal direction to the wall, i.e., $H = \lambda_0/4$.

The amplitude of deformation of a compliant coating in operation in the flow with initiated disturbances has been measured experimentally in [9]; there (Fig. 17), the resonance dependence of the amplitude on the frequency of the disturbing action and on the flow velocity has been noted and the wavelength near the resonance region has been measured (Table 2). The ratio λ/H = 3.3–4.5 has been obtained.

By analogy with capillary-gravitational waves on the liquid surface [18], the trajectory of motion of each point of a coating will generally represent an ellipse. The capillary and viscous forces in the liquid must be replaced by multicomponent elastic forces which cause shear and compressive–tensile deformations. A description of the deformation of a coating under the action of a traveling pressure wave was proposed for the first time in [19]. This analysis is very difficult and calls for knowledge of the coating parameters (modern experimenters do not have it).

Figure 3 shows the interrelation of vertical and horizontal displacements of the coating surface. A stationary two-dimensional pattern of deformations is considered. The displacement amplitudes are much smaller than the coating thickness.

Since the coating material is incompressible in practice, the volume of swelling of the surface above the level $y = H$ is equal to the volume of a horizontally shifted substance, i.e.,

$$\int_0^{\lambda_0} \xi(x)\, dx = \int_0^{H} \zeta(y)\big|_{x=0}\, dy\,.$$

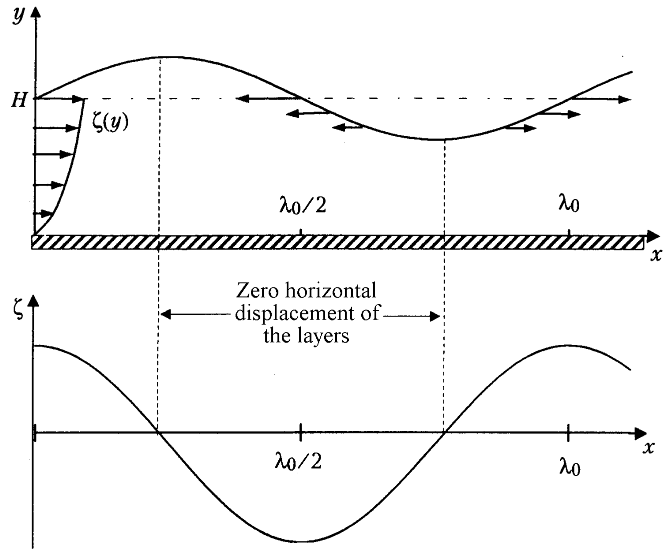

Fig. 3. Vertical (at the top) and longitudinal (at the bottom) displacements of the coating surface. The arrows show the value and direction of the displacement over the coating surface (on the left) and on its exterior surface (on the right). $y = H + \xi_{amp} \sin (2\pi x/\lambda_0)$ and $\zeta = \zeta_{amp} \cos (2\pi x/\lambda_0)$.

Elementary computations show that in the case $\lambda_0 = 4H$ the amplitudes of horizontal and vertical displacements are equal; consequently, the velocities of horizontal and vertical motions of the coating surface are the same in amplitude and are shifted by a phase angle of 90^o.

The Reynolds stresses in the near-wall region are determined by the correlation of the longitudinal and lateral velocities

$$\tau \sim \langle (u_0 + u_w)\,(v_0 + v_w) \rangle = \langle u_0 v_0 \rangle + \langle u_0 v_w \rangle + \langle v_0 u_w \rangle + \langle u_w v_w \rangle\ . \tag{3}$$

Here u_w and v_w allow for the additional influence of the compliant wall on the undisturbed turbulent flow (u_0, v_0). The last term in the given formula is equal to zero because of a phase shift of 90^o between u_w and v_w. Analysis of this formula shows that to understand the mechanism of decreasing the friction by monolithic compliant coatings one must take equal account of the vertical displacements of the coating surface (second term on the right-hand side of formula (3)) and the horizontal displacements of the surface (represented by the third term), which have not been taken into account as a rule.

From [11] it follows that for actual coatings that have shown a 16% decrease in the friction [12] the velocity of motion of the wall amounts to 4% of the maximum value of v_0 for $y_1^+ \cong 30$. Here y_1^+ denotes the dimensionless distance from the rigid surface in flow. It has been determined in [20] that the longitudinal velocity pulsations on a rigid wall are several times larger than the lateral pulsations: $(\overline{u_0^2})^{1/2} \approx 2.5(\overline{v_0^2})^{1/2}$ for $y_1^+ > 30$ (this relation becomes even larger for $y_1^+ < 30$). The additional term $\langle v_0 u_w \rangle$ obtained significantly increases the explained result of the interaction between the coating and the flow.

NOTATION

C_{sh}, velocity of the shear waves, m/sec; d, sensor diameter, m; E_{sh} and E, shear and elastic moduli of the coating material, Pa; f, vibration frequency, sec^{-1}; f_0, frequency of the first resonance, sec^{-1}; H, coating thickness, m; i, imaginary unit; P, pressure pulsation with a frequency ω, Pa; P_0, amplitude of a pressure pulsation, Pa; t, running time, sec; U, flow velocity, m/sec; U_c, convective velocity, m/sec; u_0 and v_0, longitudinal and normal velocities of undisturbed flow, m/sec; u_w and v_w, longitudinal and normal flow velocities induced by a compliant wall, m/sec; x and

y, longitudinal and normal coordinates, m; y_1^+, dimensionless distance from a rigid wall; η, loss factor of the coating material; λ, wavelength, m; λ_0, resonant wavelength, m; θ, phase lag of the deformation behind the pressure, angular degrees; ρ, density of the coating material, kg/m^3; τ, Reynolds stress, Pa; $\omega = 2\pi f$, circular frequency; ζ and ξ, longitudinal and lateral deformations, m. Subscripts: sh, shear; c, convective; w, wall; amp, amplitude.

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