surface, referred to the bow shock standoff distance; x, distance between the cylinder and the sphere; Δ, bow shock standoff distance; d, thickness of the reverse circulation flow region; E, A, B, C, D, e, b, c, d, coefficient matrices of the Navier–Stokes equations; X, column vector of the desired functions; u, v, longitudinal and transverse components of the gas velocity in the shock layer. Subscripts: ∞, parameters of the unperturbed uniform incident stream; 0, parameters on the stagnation line.

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ANALYSIS OF INTERACTION OF SINGLE-LAYER MONOLITHIC DAMPING COATINGS WITH TURBULENT FLOW

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The article analyzes the effect of the spectral-energy and phase–frequency oscillation characteristics of damping coatings on the friction coefficient and on turbulent pressure pulsations.

The article [1] presented data on the change of turbulent friction and pressure pulsations on the wall for ten variants of single-layer monolithic coatings made of three different materials whose thickness varied. The viscoelastic properties of the materials were ascertained in a wide frequency range with deformations corresponding to the operating conditions of the coatings on line.

The present article analyzes the obtained results. For that we used the calculation of the oscillation characteristics of coatings whose algorithm was presented in [2, 3]. In the mentioned calculation the viscoelastic dynamic properties of the coating materials are taken to be constant in the entire range of the analyzed frequencies. Such an assumption is correct if this frequency range lies in the zone of the high-elasticity plateau; this is so in the case of the data of the polymer materials (Fig. 1 in [1]).

Trifonov [4] pointed out the substantial influence of the frequency characteristics of coatings on the magnitude of the effect of interaction of the deformed wall with a turbulent flow.

The main oscillation characteristics are the phase angle Θ_p between the pulsating pressure applied to the coating and the displacement of its surface, and the coefficient of dynamism K_d which is equal to the ratio between the amplitude of the forced vibrations of the surface of the coating and the displacement under the effect of an equal but static pressure. Semenov [3] pointed out the undulating nature of the change of Θ_p in dependence on the vibra-

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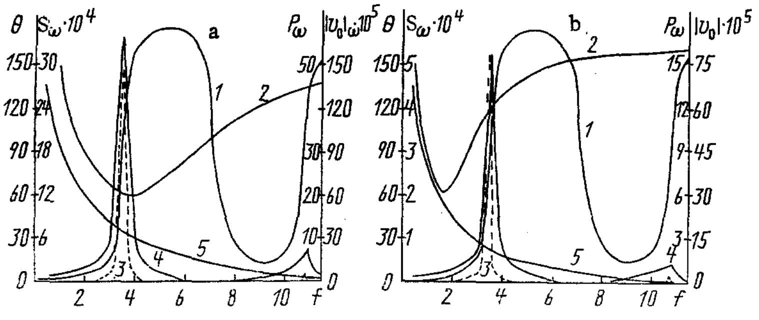

Fig. 1

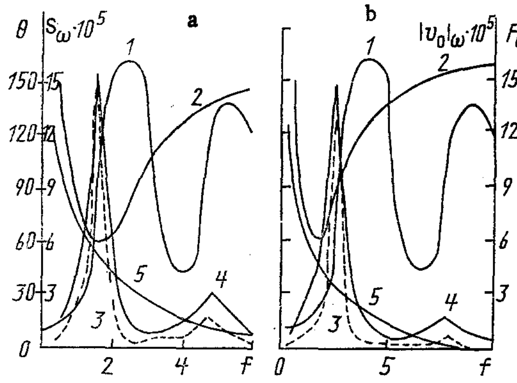

Fig. 2

Fig. 1. Dependence of the characteristics [1) Θ_p, deg; 2) Θ_0, deg; 3) $S_\omega \cdot 10^4$, J/m^2; 4) $|v_0|_\omega \cdot 10^5$, m/sec; 5) P_ω, N^2•sec/m^4] on the frequency f, kHz, with flow velocity: a) U = 10.5 m/sec; b) 15.5. The material of the coating is KLT-30A, thickness 3 mm.

Fig. 2. Dependence of the characteristics [1) Θ_p, deg; 2) Θ_0, deg; 3) $S_\omega \cdot 10^5$, J/m^2; 4) $|v_0|_\omega \cdot 10^5$, m/sec; 5) P_ω, N^2•sec/m^4] on the frequency f, kHz. U = 10.5 m/sec. The coatings are made of material No. 2 with thickness: a) 7.0 mm; b) 4.0.

tion frequency and the resonance behavior of K_d. The oscillation amplitudes Θ_p and the resonance values of K_d increase with decreasing loss coefficient of the material.

Figures 1 and 2 present the phase-frequency characteristics of coatings Θ_p and Θ_0 for some typical regimes. Θ_0 characterizes the phase angle at which a neutral effect of the viscoelastic lining on turbulent friction is ensured [2, 5]. The phase angles Θ_p, when lying above the neutral curve, may lead to reduced friction, when lying below it, to increased friction.

It can be seen from the presented drawings that in all the selected regimes there is a frequency range where $\Theta_p > \Theta_0$ and where reduced friction may be expected. In some regimes (Fig. 1b) this range encompasses even the second wave of Θ_p. However, the different frequencies coming into the range of a possible positive effect influence the reduction of friction in different ways.

It follows from the condition that the liquid does not flow over the damping boundary that the rate of deflection of the surface under the effect of turbulent pressure pulsations dy/dt is equal to the normal component of the pulsating speed of the flow on the wall v_0. Since the magnitude of the Reynolds friction stresses <uv> is proportional to v, the effect of the damping coating itself on the characteristics of turbulent flow is determined in the first approximation by the magnitude of dy/dt. The weight factor, which takes the contribution of the individual frequencies into account, is therefore proportional to the component of the pulsating speed of the flow on the wall $|v_0|_\omega$.

If we represent the pulsating pressure in the form of a set of harmonic components $\varphi_T(\omega, t)$ [6] with the spectral density of distribution P_ω, we obtain that the random magnitude of deflection of the surface y also consists of a set of harmonics $y_T(\omega, t) = \varphi_T^+(\omega, t)\, K_d H/E$. Figures 1, 2 show the magnitude of $|v_0|_\omega = \omega H K_d \sqrt{P_\omega}/E$ determined as the square root of the spectral density of the strain rate of the elastic surface where the distinct resonance nature of the behavior of these weight coefficients can be seen. At the resonance frequency the sharpness of the peak (Q factor) is determined by the loss coefficient of the material of the coating, and in addition to that, the magnitude of the peak depends on the rigidity of the coating, on the resonance frequency, and on the pulsating pressure at this frequency.

Also presented are the spectral distributions of the energy dissipation rate by the wall per unit surface area $S_\omega = \omega \sin\Theta_p H K_d P_\omega/E$. For the calculations we took the spectrum of pressure pulsations measured by Willmarth and Roos [7] on a flat plate with the ratio $r_0/\delta \to 0$. Such a form of the spectrum is close to the true one because there is no averaging of the pressure pulsations over the area of the sensor. In the calculation of the spectral density of the pressure pulsations, τ_w and δ^* were calculated by formulas [8] for the coordinate x equal to the midpoint of the measuring insert; x = 1.1 m [1].

TABLE 1. Comparison of the Permissible and Dynamic Roughnesses, and also of Specific Energies Absorbed and Dissipated on Damping Walls

Coating $H \cdot 10^2$, m	Speed U, m/sec	Roughness: permissible $y_{per} \cdot 10^6$, m	Roughness: of deflection of surface $\sigma_y \cdot 10^6$, m	Specific energy: dissipated on wall E_d, J/m²	Specific energy: absorbed by wall W, J/m²	$W/E_d \cdot 10^{-4}$	Reduction of friction Ψ, %
KLT-30A; 3.0	10,5	13	2,1	84,4	$9{,}8\cdot10^{-3}$	1,2	—3
KLT-30A; 3.0	15,5	9	5,4	185	$4\cdot10^{-2}$	2,2	—14
KLT-30A; 4.0	10,5	13	4,3	84,4	$1{,}8\cdot10^{-2}$	2,1	0
KLT-30A; 4.0	13,5	10,4	6,6	140	$3{,}1\cdot10^{-2}$	2,2	—10
№ 1; 2,0	13,5	10,4	5,1	140	$2\cdot10^{-2}$	1,4	—17
№ 2; 4,0	10,5	13	0,9	84,4	$7{,}1\cdot10^{-3}$	0,84	+17
№ 2; 4,0	15,5	9	1,8	185	$1{,}8\cdot10^{-2}$	0,97	+2
№ 2; 7,0	10,5	13	1,3	84,4	$4{,}2\cdot10^{-3}$	0,50	+11

In [5, 9] the condition was introduced that the deflection of the surface must not exceed a limit value which is equal to the critical surface roughness. However, these authors calculated the deflection of the surface solely for the condition of static deformation.

In accordance with the laws of describing random variables, the magnitude of the dynamic roughness is the rms deflection of the surface. Since the spectrum of the magnitude of displacement of the surface is $P_y(\omega) = K_d^2H^2P_\omega/E$, the dispersion of displacement of the surface is $\sigma_y^2 = \int_{-\infty}^{\infty} P_y(\omega)\,d\omega = 2\int_0^{\infty} (K_d^2H^2P_\omega/E)$. Hence, the rms displacement of the damping surface under the effect of turbulent pressure pulsations has the form

$$\sigma_y = \sqrt{\overline{y(t)^2}} = \frac{\sqrt{2}H}{E}\left(\int_0^{\infty} K_d^2 P_\omega d\omega\right)^{0,5}.$$

The derived formula is correct for all types of damping coatings, and not only for single-layer monolithic ones. However, in distinction to the thoroughly investigated classical types of roughness, e.g., sandy roughness, the manifestation of roughness due to the deflection of a compliant surface has some characteristic features. The main difference is that this roughness manifests itself only in the range of the resonance frequencies of the coating. The regularities of how such roughness affects the turbulent characteristics of the flow may differ from the classical ones, and so far they have not been investigated at all.

Table 1 presents the calculated data on the permissible values of roughness and on the rms values of deflection of the surface for the described regimes. The permissible roughness was determined by the formula $y_{per}v^*/\nu = 5$. The value of δ_y was obtained by the method of numerical integration. It can be seen from Table 1 that the dynamic roughness is smaller than the permissible static roughness of the surface for all the analyzed regimes. The relative deformation for these regimes lies within the range $2\cdot10^{-4}$–$2.6\cdot10^{-3}$, which confirms the correctness of the requirement that the coating materials should be tested with small values of relative deformations.

Let us now analyze the results of hydrodynamic tests of a series of damping coatings jointly with the spectra of pressure pulsations on the wall, measured directly behind the coatings (Fig. 4 in [1]).

Tests of coatings of material KLT-30A with loss coefficient $\eta = 0.05$ [1, 10], 3.0 mm thick, at a speed of 10.5 m/sec (Fig. 1a) yielded some increase of friction and of the level of pressure pulsations. The possible positive effect was imperceptible (the minimum of the phase-frequency characteristic Θ_0 lies to the left of the region of resonance interaction) and was more than outweighed by the negative effect of roughness. Roughness, whose manifestation in our case is described by the regularity of the transient regime ($5 \le yv^*/\nu \le 70$), yields an increase of friction of ~10% [8]. On the spectrum of the pressure pulsations we noted an increase of spectral density to frequencies of ~3.5 kHz, i.e., to the commencement of the region of the predicted positive effect. The same coating but at a higher speed u =

15.5 m/sec (Fig. 1b) yielded a greater increase of friction and pressure pulsations. With increasing speed the minimum of the phase frequency characteristic of the curve of neutral effect is shifted to the right, and it becomes extended along the frequency axis. A variation of speed does not affect the width of the frequency band of resonance interaction or the resonance frequency itself. Regardless of the fact that the phase-frequency conditions for the manifestation of the positive effect became optimal, an increase of dynamic roughness and reduction of the permissible level of roughness apparently exerted a greater influence. However, on the spectrum of pressure pulsations we found a decrease of spectral density at frequencies above 3.5 kHz (of the peak of resonance interaction and the commencement of the region of expected positive effect).

In the second region of possible positive effect at frequencies of ~11 kHz, which contains the second maximum of the phase—frequency oscillation characteristic of the coating, the rate of deflection of the surface, and consequently the effect of the damping coating on the characteristics of flow are about one order of magnitude smaller than at the first frequency of resonance interaction. In the region of the high-frequency part of the pressure pulsations their spectral density is low, and the damping action is therefore ineffective. Moreover, it is possible that the mechanism of operation of the damping coating in the region of dissipative frequencies differs from the mechanism of operation in the region of energy-carrying frequencies.

The characteristics of coatings of material with a loss coefficient η = 0.2 and with a thickness of 4.0 and 7.0 mm are presented in Fig. 2. For the first regime the minimum of Θ_0 lies slightly to the left of the region of resonance interaction. The width of this region, determined at half the height of the resonance peak, is equal to ~900 Hz. Regardless even of the fact that the magnitude of the roughness is comparable with the permissible one, with this regime we found a decrease of friction and of the level of pressure pulsations by 17 and 16%, respectively. The spectral level of the pressure pulsations was lower than for a hard surface when the frequencies were higher than 2 kHz, analogously to the preceding cases.

An increase of the thickness of the coating from 4.0 to 7.0 mm led to reduced width of the resonance region by a factor of ~1.5. And though in this case the minimum of the phase-frequency characteristic of neutral effect and the peak of resonance interaction lie at the same frequency, and the sum of static and dynamic roughness does not exceed the maximally permissible value, i.e., the coating is hydrodynamically smooth, a test of this coating showed that friction was reduced by 11% only. It seems that the width of the band of interaction frequencies is one of the important factors influencing the effectiveness of operation of a damping coating.

In Table 1 we compare the specific energies dissipated on the wall and absorbed by the coatings. The specific energy dissipated on a section of wall with length L is equal to $E_d = \tau_w L$. The specific energy absorbed by the wall was determined by the method of numerical integration by the formula $W=(L/U_c)\int_0^\infty S_\omega d\omega$, where L = 0.59 m is the length of the coating, U_c = 0.8 U. It can be seen from Table 1 that the energy absorbed by the wall constitutes only a small fraction of the turbulent energy dissipated on the wall:

$$W/E_d = (0,5 \div 2,2)\,10^{-4}.$$

A comparison of the proportion of turbulent energy absorbed by the damping wall to the energy dissipated on the same length with the effect of lowering friction shows that there is no direct correlation between them. On the contrary: coatings that reduce friction absorb a smaller proportion of turbulent energy than coatings that increase friction. This conclusion shows the unsoundness of the explanation of the effect of reduced friction by damping coatings solely by the energy absorption of the wall [9].

For effective operation of damping coatings in a turbulent flow it is necessary to ensure such viscoelastic characteristics of the material of the coating and to choose such a velocity regime that the region of positive effect obtained from phase—frequency notions and the band of frequencies of resonance interaction, which should be as wide as possible, lie in the region of energy-containing frequencies. In addition to that, the total effect of static and dynamic roughness of the coating has to be smaller than the maximally permissible one to ensure hydrodynamic smoothness. To satisfy these conditions, mutually contradictory requirements would have to be met.

It follows from the above examination of the operation of monolithic single-layer damping coatings that:

1. The interaction of coatings with turbulent pressure pulsations is of a resonance nature. The first resonance frequency $f_0 \simeq \sqrt{E/\rho}/4H$ (see [3]) is mainly responsible for the interaction. The width of this band of active influence Δf_0 is determined by the loss coefficient $\Delta f_0 H/\sqrt{E/\rho}$ = const. Hence it follows that if the effect is to be enhanced, more rigid materials with lower density have to be chosen and the thickness of the coating has to be reduced. These same requirements are also indispensable for reducing dynamic roughness.

2. This is in contradiction to the condition of increasing the degree of interaction of the damping wall with the turbulent flow $|v_0|_\omega = \omega H K_d \sqrt{P_\omega}/E$.

3. If the band of interaction frequencies is to lie in the region of the energy carrying frequencies, the inequality $\sqrt{E/\rho}/H < 2U/\pi\delta^*$ has to be fulfilled; this is in agreement with the requirement 2.

4. The condition of optimality of the phase-frequency characteristic of neutral effect requires that $2 \cdot 10^{-2} < \pi\nu f_0/v^{*2} < 6 \cdot 10^{-2}$ [3], which imposes additional constraints on the selection of properties of the material of the coating, or in the work on an actual coating it leads to a selective dependence on the speed.

NOTATION

Θ_0, Θ_p, phase angles; K_d, coefficient of dynamism; v, u, normal and longitudinal components, respectively, of the pulsating speed of the flow; t, time, ω, angular frequency; f, cyclic frequency; y, deflection of the surface; σ_y, dispersion of the deflection; E, ρ, η, modulus of elasticity, density, and loss coefficient, respectively, of the material of the coating; H, thickness of the coating; x, coordinate of the midpoint of the coating; P_ω, S_ω, $P_y(\omega)$, spectra of the pressure pulsations, of the energy dissipation rate, of the deflection of the surface, respectively; r_0, radius of sensitivity of the pressure-pulsation sensor; δ, thickness of the boundary layer; δ^*, thickness of displacement; U, flow rate; ν, kinematic coefficient of viscosity of water; v^*, dynamic velocity; τ_w, friction stress on the wall; L, length of the damping wall; E_d, dissipated specific energy; W, absorbed specific energy; u_c, convective speed; f_0, resonance frequency; Δf_0, resonance width; ψ, reduction of the friction coefficient; $\varphi_T(\omega, t)$, $y_T(\omega, t)$, harmonics of pressure pulsations and of deflection of the surface, respectively.

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