EXPERIMENTAL STUDY OF THE EFFECT OF SINGLE-LAYER VISCOELASTIC COATINGS ON TURBULENT FRICTION AND PRESSURE PULSATION ON A WALL

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Information is given on the change in turbulent friction and pressure pulsations on a wall with the application of single-layer monolithic coatings. The viscoelastic properties of the materials used are characterized.

The idea of using distributed damping of a wall to reduce friction is attributed to Kramer [1-3], who described and patented a method of making a flow laminar and different schemes of possible viscoelastic coatings for this purpose, including a single-layer coating. The action of the coating is interpreted as an elongation of the laminar-boundary-layer section as a result of damping of pulsations in the velocity of the viscoelastic boundary. During the same period Judge [4] proposed an analogous idea (without substantiation) in which turbulent friction is reduced while in principle retaining a turbulent flow regime by using damping facings for the surfaces of solids and pipelines. Direct measurements of flow characteristics about the viscoelastic boundary showed that both approaches are valid [5, 6].

Kramer achieved a reduction of up to 50% in hydrodynamic resistance by using two designs of facing which can tentatively be classified as three-layer coatings with a middle layer made of a material with a constant viscosity (since the middle layer is filled with a viscous fluid). The incomplete information on the properties of the materials and the fairly complicated (with regard to design) coating schemes developed by Kramer have thus far prevented determination of their frequency–phase characteristic in order to completely analyze the results he obtained. Technical difficulties arising in the development of the structurally complex Kramer facings — "columnar" and "ribbed" — prevent investigators from making them and thus repeating his tests with fixed boundary-turbulence characteristics or at least using experimental methods of polymer mechanics to study the complex compliance of the coatings.

Many of the investigators who followed the lead of Kramer and Judge (see the surveys in [7, 8]) studied the effect of damping coatings on boundary turbulence mainly under laboratory conditions [9] (a fluid-impregnated layer of elastic expanded plastic coated with a smooth thin film was secured to a hard base), the coatings here being classifiable as two-layer coatings with a main layer made of a material with a constant viscosity. Different investigators achieved a reduction of 40-60% in turbulent friction on the viscoelastic boundary. However, incomplete information on the oscillatory characteristics of the investigated damping coatings prevents analysis of the results to evaluate the available theoretical approaches [10-12]. For example, while some researchers present data on the material of the coating, they do not allow for the effect of the fluid filler — which is the main reason for the energy dissipation in the given scheme.

It is difficult to objectively study viscoelastic properties in the bending vibrations of such coatings using existing experimental methods. There are also problems in conducting hydrodynamic experiments with facings with a fluid filler which moves in accordance with the distribution of the mean excess hydrodynamic pressure and as a result forms pimples and depressions on the surface. Thus, such coatings are not suitable for tests in tubes and in zones with a nonzero pressure gradient in the case of external flow. In setting up the tests, it is necessary to carefully evacuate gas bubbles from the filler. However, due to some porosity in the film layer, air again gradually penetrates inside the coating. This makes it difficult to maintain the required stability of the viscoelastic properties of the facing.

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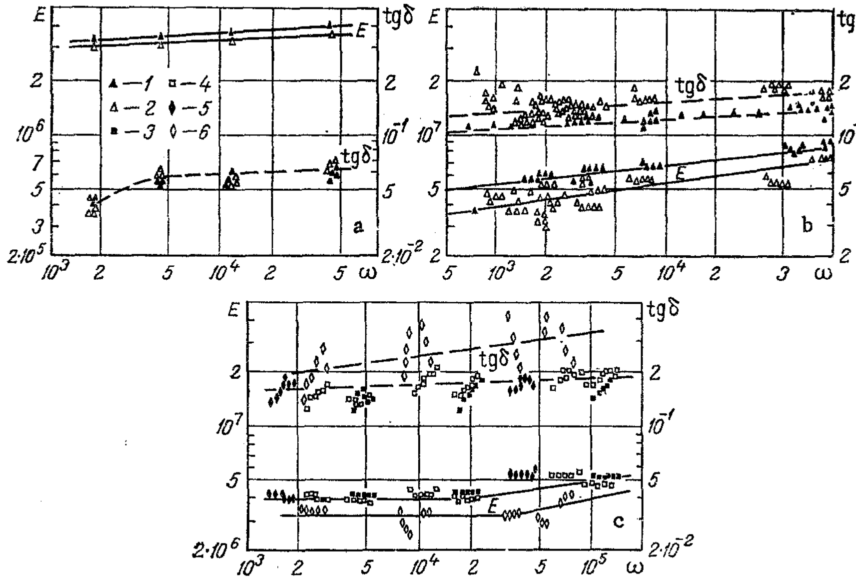

Fig. 1. Dependence of the dynamic viscoelastic characteristics of the materials on frequency with different strain amplitudes (the solid curves represent Young's modulus E, the dashed curves show the loss tangent tan δ): a) vulcanized-rubber compound KLT-30A; 1) $|\varepsilon| = 0.1\%$; 2) 0.2; b) material No. 1: 1) $|\varepsilon| = 0.1\%$; 2) 0.2–3.2, c) material No. 2: 3) $|\varepsilon| = 0.03$–0.04%; 4) 0.08–0.13; 5) 0.2–0.3; 6) 0.45–1.3. E, N/m^2, ω, 1/sec.

The above problems have made it necessary to study the effect of viscoelastic coatings on boundary turbulence by using the simplest monolithic scheme (single-layer coatings). An algorithm was developed for such coatings [9] which connects the main oscillatory characteristics with the properties of the material. To verify the former, a measurement unit has been developed and methods devised for studying the viscoelastic properties of rubber with allowance for features of its use.

The problem in the present study is to obtain information on the correlation between turbulent friction and pressure pulsations on a wall on the one hand and its oscillatory characteristics on the other hand using single-layer coatings as an example. Manufacturing constraints made it necessary to use pourable compounds. Specifically, we used silicone compound KLT-30A, which is commercially produced [13]. Graphs of the dependences of the viscoelastic properties of this compund are shown in Fig. 1a. To more broadly vary the parameters of the coatings, we attempted to prepare our own pourable compositions containing vulcanized rubbers with viscoelastic characteristics which would be substantially different from those in the commercial brand.

Using low-molecular-weight silicon rubbers SKTN and SKTNF as a basis, we prepared materials Nos. 1 and 2. The dynamic properties of these materials are shown by the graphs in Fig. 1b and c. The data corresponds to a temperature of 25°C. To use the data to analyze test data for other temperatures, the Williams–Landel–Ferry formula can be used [14]. The glass point is 143°K. Other information on the characteristics of the facings tested is shown in Table 1. The coatings were applied to a replaceable insert in the model, shown in Fig. 2. To measure drag, the insert was fastened to strain-gauge balances with gaps of 0.5–1.0 mm relative to the body. The insert was located in a region of nongradient flow with zero excess hydrodynamic pressure $\bar{p}$. The scales thus measured its resistanve to friction, while flow about the insert corresponded to flow about a plate. A vibration-proofed piezoelectric transducer [15] to detect pressure pulsations was placed immediately behind the insert, still in the nongradient flow zone. The signal from the transducer was amplified by a factor of ten in a preamp and then further amplified in an amplifier with fifth-order Cauer filters, cutting off frequencies below 500 Hz and above 6 kHz. The mean square value of the pressure pulsations in this frequency range was measured by a VZ-40 voltmeter. The pressure-pulsation spectrum was analyzed on an S5-3 unit with a band of 200 Hz and recorded on a type 2305 "Bruell and Kerr" recorder.

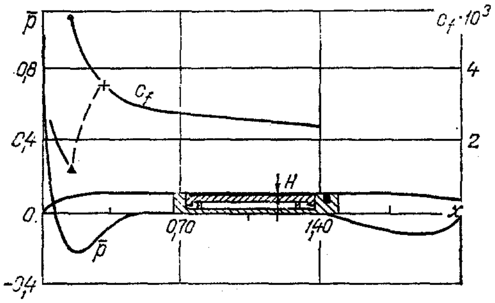

Fig. 2

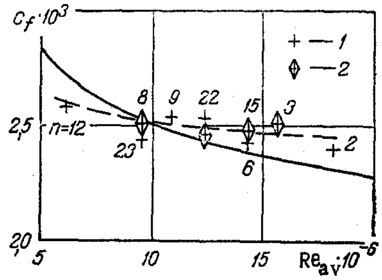

Fig. 3

Fig. 2. Diagram of model with indication of distribution of excess hydrodynamic pressure and the friction coefficient.

Fig. 3. Dependence of the friction coefficient of the hard smooth boundary on the Reynolds number. The solid curve shows results of calculation by Truckenbrodt's method; the dashed line shows an approximation of the experimental data. Ellipses denoting scatter of the test data: 1) with vortex generator; 2) without generator.

Along with the distribution of the measured excess hydrodynamic pressure, Fig. 2 shows as an example the results of calculation of the friction coefficient $c_f(x)$ for a smooth hard boundary with a mean Reynolds number $Re_{av} = 1.1 \cdot 10^7$ ($Re_{av} = ux_{av}/\nu$), where $x_{av} = 1.05$ m corresponds to the abscissa of the midpoint of the insert. Well-known theoretical methods from boundary-layer theory [16] were used to determine the friction coefficients for the initial section with laminar flow, its length (the triangles in Fig. 2 denote the friction coefficient at the point of loss of stability, while the dashed line denotes the transition from laminar to turbulent flow in the case of small external perturbations), and drag coefficients. The results of the calculations show that in the range of Reynolds number $Re_{av} = (5\text{–}20) \cdot 10^6$ developed turbulent flow is realized about the insert. The value of the friction coefficient calculated for the mean abscissa x_{av} is roughly equal to half the sum of the values found at the beginning and end of the insert (the law of change in friction along it is nearly linear — see Fig. 2). The solid line in Fig. 3 shows the results of calculation of the friction (drag) coefficient of the insert as a function of the mean Reynolds number for the range used in the tests.

We carefully measured the frictional force and pressure pulsations on the wall for the case of an insert with a hard, polished surface ($k_s < 1$ μm). Tests were performed both with a wire vortex generator (diameter 1 mm at $x = 0.025$ m) and without it. The results of friction measurement are shown in Fig. 3. The area of the signs includes the scatter of the test data on velocity and the measured friction coefficient. Its absolute error is evaluated at $5 \cdot 10^{-5}$. The figure also shows the number of tests n which characterize each mean corrected value. The comparison made provides evidence of the closeness of the calculated and measured values for the hard smooth boundary. Meanwhile, the vortex generator is seen to not have an effect. This means that there is developed and similitudinous flow in this region. For comparison with the results presented below on measurements of friction on the section with coatings, the test data for the hard polished insert was approximated by a relation (the dashed line) which differs somewhat from the theoretical relation used. A similar relation was constructed for the mean square values of the pressure pulsations as a function of the mean Reynolds number in $5 \cdot 10^6 \leqslant Re_{av} \leqslant 2 \cdot 10^7$.

Since the turbulent friction and pressure pulsation on the viscoelastic boundary depend not only on the flow parameters modeled by the Reynolds number but also on the characteristics of the damping coating, we present the following primary data for convenience of analysis of these tests: the velocity of the water flow u, the ambient temperature t, the number of the coating described above. We will present the results of the study of ten coatings. Table 2 shows the following values: ψ_τ — the change in the friction coefficient of a coating insert c_{fd} compared to a smooth, hard boundary (in percents):

$$\psi_\tau = 100 \cdot (c_f - c_{fd})/c_f,$$

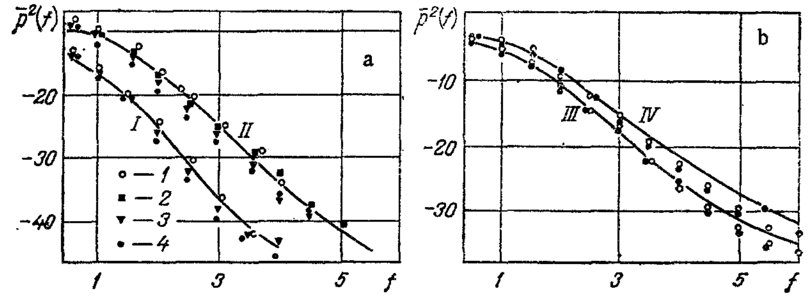

Fig. 4. Spectra of pressure pulsations on a wall for a hard and a compliant surface (f, kHz; $p^{-2}(f)$, dB): I) u = 10.5 m/sec; II) 13.5; III) 15.5; IV) 16.5; the solid curves show an approximation of the test data for the hard boundary; the points are for the coatings: 1) coating No. 4; 2) No. 5; 3) No. 6; 4) No. 7 (test No. 7).

TABLE 1. Characteristics of the Coatings

Number of coating	Material of the coating	$\rho \cdot 10^{-3}$	μ	$E_0 \cdot 10^{-6}$	H	k_s	Notes
1	KLT-30A (1978 batch)	1,23	0,46	2,2	1,5	41	
2	KLT-30A (1978 batch)	1,23	0,46	2,2	2,0	48	
3	KLT-30A (1978 batch)	1,23	0,46	2,2	1,4–2,0	12	Coating with variable thickness
4	KLT-30A (1978 batch)	1,23	0,46	2,2	3,0	19	
5	KLT-30A (1979 batch)	1,23	0,46	1,6	4,0	19	
6	№ 1	4,00	0,50	0,47	2,0	20	
7	№ 2	2,14	0,47	3,7	4,0	16	Isolated cavities on the surface
8	№ 2	2,14	0,47	3,7	2,5	5	
9	№ 2	2,24	0,47	3,7	4,4	3	
10	№ 2	2,14	0,47	3,7	7,0	5	Same

ψ_p — the change in the mean-square pressure pulsations (in the overall frequency band) (also in percents):

$$\psi_p = 100 \cdot ([p] - [p]_d)/[p].$$

The number of tests n for each velocity regime is also indicated. The error of the values shown is about 5%. It should be noted that although the comparison was made with a hydraulically smooth hard boundary, the roughness of coatings Nos. 1-7 was greater than the limiting permissible level in keeping with present representations for a hard boundary [16] at velocities u ~10.5 m/sec.

The following can be concluded from examining the experimental results shown on the change in average friction and mean-square pressure pulsations as a result of the viscoelastic boundary (both positive and negative).

A single-layer monolithic coating made of a highly elastic material can react very sharply to certain test conditions. For example, coatings Nos. 7-8, tested at two flow temperatures differing by two degrees, exhibited effects of opposing signs. We see a sawtoothed dependence of the effect of the change in friction on velocity for coating No. 10. This phenomenon is evidently connected with a sharp, sawtoothed change in the frequency–phase characteristic of a single-layer monolithic coating made of a material with insufficiently high mechanical losses [9].

We may also note as an interesting finding that the effects of a change in friction and the mean square value of the pressure pulsations nearly coincide in the frequency range from 500 to 6000 Hz, i.e., the Kreichnan coefficient [17] evidently retains its value with the introduction of a viscoelastic boundary.

In the above-described experiment we also measured the spectra of the pressure pulsations beyond an insert both with a hard polished surface and with damping coatings. Some results are shown in Fig. 4, which gives values of the level (in decibels) of the spectral density as a function of frequency in the range from 500 Hz to 6 kHz. The solid lines show

TABLE 2. Experimental Data on the Change in Integral Characteristics by the Coating

Number coatings	Number tests	t	n, ψ_τ, ψ_p	u 6,7	9,0	10,5	12	13,5	15,5	16,5
1	1	6	n	—	—	1	—	2	2	—
			ψ_τ	—	—	—10	—	—14	—17	—
2	2	6	n	—	—	2	—	2	—	—
			ψ_τ	—	—	+8	—	+5	—	—
3	3	12	n	—	—	2	—	2	2	—
			ψ_τ	—	—	—2	—	—18	—23	—
			ψ_p	—	—	0	—	—10	—12	—
4	4	12	n	—	2	2	—	2	2	2
			ψ_τ	—	+6	—3	—	—7	—14	—17
			ψ_p	—	0	—7	—	—12	—12	—8
5	5	12	n	—	—	3	—	4	—	—
			ψ_τ	—	—	0	—	—10	—	—
			ψ_p	—	—	—	—	—8	—	—
6	6	12	n	—	4	3	—	3	3	—
			ψ_τ	—	0	—3	—	—17	—25	—
			ψ_p	—	—	—6	—	—16	—25	—
7	7	12	n	—	—	2	—	3	4	2
			ψ_τ	—	—	+17	—	+4	+2	—4
			ψ_p	—	—	+16	—	+9	0	—2
	8	14,0 ÷ 14,5	n	9	—	14	—	6	3	—
			ψ_τ	+12	—	+2	—	—6	—9	—
8	9	16	n	—	—	11	—	8	9	6
			ψ_τ	—	—	—12	—	—10	—18	—15
	10	14	n	5	—	11	—	8	9	6
			ψ_τ	+14	—	+10	—	+4	+4	+6
9	11	14,0 ÷ 15,0	n	10	—	10	16	10	3	4
			ψ_τ	+17	—	+9	+10	+3	+1	+5
10	12	14	n	17	10	9	6	—	—	—
			ψ_τ	+15	+6	+13	+6	—	—	—

measured dependences for the hard surface. Comparing the results shown in the figures and in Table 2, we note that in a number of tests the spectral density of the pressure pulsations at frequencies greater than 2 kHz decrease near the viscoelastic wall by 2-3 dB, despite an increase in friction and the mean square value of the pulsations.

NOTATION

E_0, initial quasiequilibrium Young's modulus, N/m²; E, dynamic Young's modulus, N/m²; tan δ, tangent of angle of mechanical losses in the material; ω, angular frequency, 1/sec; f, cyclic frequency, kHz; $|\varepsilon|$, amplitude of strain under dynamic loading; μ, Poisson's ratio; ρ, density of the material, kg/m³; H, thickness of the coating, mm; k_s, mean-square roughness of the surface, µm; u, flow velocity, m/sec; t, water temperature, °C; n, number of tests; x, abscissa, m; p, relative excess hydrodynamic pressure; c_f, friction coefficients for a hard smooth boundary; c_{fd}, friction coefficient of an insert with a coating; $\psi_\tau = 100 (c_f - c_{fd})/c_f$; $Re_{av} = ux_{av}/\nu$, mean Reynolds number, corresponding to the abscissa of the midpoint of the insert; ν, kinematic viscosity of water, m²/sec; [p], mean-square pressure pulsation for the hard smooth boundary; $[p]_d$, mean square pressure pulsation for the coated insert; $\psi_p = 100 ([p] - [p]_d)[p]$; $\overline{p^2}(f)$, spectral density of the pressure pulsations on the wall, dB.

LITERATURE CITED

1. M. O. Kramer, "Boundary-layer stabilization by distributed damping," J. Aeronaut. Sci., 24, No. 6, 459-460 (1957).
2. M. O. Kramer, "Boundary-layer stabilization by distributed damping," J. Am. Soc. Nav. Eng., 72, No. 1, 25-33 (1960).
3. M. O. Kramer, "Boundary-layer stabilization by distributed damping," Nav. Eng. J., 74, No. 2, 341-348 (1962).

4. J. F. Judge, "Turbulence drag drastically reduced," Missiles Rockets, 6, No. 5, 42-43 (1960).
5. V. V. Babenko and L. F. Kozlov, "Experimental study of the hydrodynamic stability of a laminar boundary layer on an elastically damping surface in a water flow," in: Bionics [in Russian], Naukova Dumka, Kiev (1972), pp. 22-23.
6. E. F. Blick, R. R. Walters, R. Smith, and H. Chu, "Compliant coating skin friction experiments," AIAA Paper, No. 69-165 (1969).
7. B. N. Semenov, "Effect of elastic coatings on a turbulent boundary layer," in: Investigations of Boundary-Layer Control [in Russian], ITF Sib. Otd. Akad. Nauk SSSR, Novosibirsk (1981), pp. 57-76.
8. D. M. Bushnell, J. N. Hefner, and R. L. Ash, "Effect of compliant wall motion on turbulent boundary layers," Phys. Fluids, 20, No. 10, Pt. II, S31-S48 (1977).
9. B. N. Semenov, "Analysis of the strain characteristics of viscoelastic coatings," in: Hydrodynamics and Acoustics of Boundary-Layer and Free Flows [in Russian], ITF Sib. Otd. Akad. Nauk SSSR, Novosibirsk (1981), pp. 57-76.
10. B. N. Semenov, "Interaction of an elastic boundary with the viscous sublayer of a turbulent boundary layer," Zh. Prikl. Mekh. Tekh. Fiz., No. 3, 58-62 (1971).
11. D. T. Tsahalis, "On the theory of skin friction reduction by compliant walls," AIAA Paper, No. 77-686 (1977).
12. G. F. Trifonov, "Theoretical study of the effect of the characteristics of an elastic boundary on turbulent flow," in: Effect of Polymer Additions and Surface Elasticity on Boundary-Layer Turbulence [in Russian], ITF Sib. Otd. Akad. Nauk SSSR, Novosibirsk (1978), pp. 75-85.
13. Silicone Products Made in the USSR (Handbook – Catalog) [in Russian], Khimiya, Moscow (1970).
14. J. D. Ferry, Viscoelastic Properties of Polymers, Wiley (1970).
15. V. M. Kulik and B. N. Semenov, "Measurement of pressure pulsations on a wall washed by dilute solutions of polyethylene oxide," in: Effect of Polymer Additions and Surface Elasticity on Boundary-Layer Turbulence [in Russian], ITF Sib. Otd. Akad. Nauk SSSR, Novosibirsk (1978), pp. 51-56.
16. G. Schlichting, Boundary Layer Theory, McGraw-Hill (1968).
17. I. Ya. Miniovich, A. D. Pernik, and V. S. Petrovskii, Hydrodynamic Sources of Sound [in Russian], Sudostroenie, Leningrad (1972).

STABILITY OF LAMINAR FLOW OF A LIQUID AND GAS IN A HORIZONTAL CHANNEL

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It is shown that the transition from laminar to wave flow depends on the Froude number and the ratio of the equivalent thicknesses of the liquid and gas.

The interaction of a gas flow with a liquid during laminar movement in a channel is important for the design of various heat-exchanger apparatus used in power and chemical engineering. Existing regime charts and their modifications [1, 2] were obtained from visual observations and, as noted by the authors themselves, are qualitative in nature.

To more objectively classify flow regimes for two-phase flows, the study [3] proposed the use of spectral characteristics of the pressure pulsations or shear stresses on the wall. These oscillations are the result of characteristic instabilities corresponding to different modes of motion. In some cases [4], the appearance of waves and the subsequent transition to a slug regime of flow is identified with Kelvin–Helmholtz instability. Such an approach does not consider the inertial characteristics of the liquid and energy dissipation.

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