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SUPPRESSION OF OSCILLATIONS OF PISTON-AIDED WIND TUNNELS

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The method chosen to compensate for the forces acting on the barrel in the course of compression and confinement of the test gas in the settling chamber of a hypersonic wind tunnel with a free locking piston is justified. A method is proposed to compensate for the effect of adverse factors by optimizing the mass of an auxiliary piston introduced into the system. The effectiveness of the method is validated experimentally. As a result, for the case of gas compression to 200 MPa*, the displacement of the center of mass of the barrel is reduced from* 50 *to* 0.25 mm*, which is smaller than the amplitude of elastic axial extension corresponding to the maximum pressure of the gas.*

Key words: *adiabatic compression of the gas, barrel oscillations, energy of elastic deformations.*

1. Using a pure homogeneous gas compressed in aerodynamic facilities with a free heavy piston, one can obtain not only high values of flow parameters (pressures up to 200 MPa and temperatures up to 2000 K), but also a high quality of hypersonic flows [1]. The term "free piston" means that the piston motion in the barrel is induced only by the difference in pressures of the driving and compressed gases; the expression "heavy piston" means that the piston velocity is much lower than the velocity of sound in the compressed gas.

Piston-aided gas-dynamic facilities, such as "Longshot" (Belgium) [2] and U-11 and U-7 (Russia) [3], are equipped with a settling chamber separated from the barrel by a valve assembly with a set of one-way valves. At the moment when the maximum compression of the test gas is reached in such facilities, the gas pressure in the barrel becomes higher than the pressure in the settling chamber, and the one-way valves are opened, the free piston starts accelerated motion in the opposite direction. Because of that and also because of the presence of the so-called parasitic volumes, only some part of the gas enters the settling chamber, whereas the remaining part of the gas stays in the barrel and ensures further motion of the piston in the regime of nonlinear decaying oscillations. The barrel moves in the opposite phase to the piston, and the amplitude of motion is proportional (the friction force being assumed to be small) to the ratio of the piston mass to the barrel mass. As the barrel displacements after the end of the piston stroke coincide in time with the period of gas-dynamic measurements and involve difficulties in the measurements, these displacements were called the "parasitic oscillations" [4].

These "parasitic oscillations" can be avoided by using a locking piston [5]. As the locking piston behaves as a free piston in the working stroke (gas compression), there is one opposing displacement of the facility. As the piston is fixed at the end of the working stroke and is maintained in this position by forces of friction against the barrel, the motion of the facility is terminated. The locking piston blocks the gas compressed to high parameters at the end of the barrel attached to a conical hypersonic nozzle opened into the test section of the gas-dynamic facility.

Operation of the locking piston induces a significant shift of the barrel. For instance, for a small-scale facility (inner diameter of the barrel 50 mm, mass of the barrel 150 kg, and length of the working area 1.7 m) with a locking piston having a mass of 4.25 kg, the barrel displacement is approximately 50 mm. It is impossible to prevent barrel

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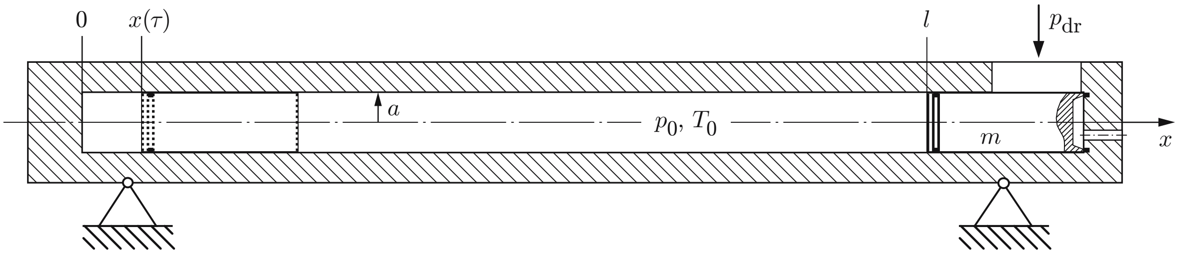

Fig. 1. Layout of a gas-dynamic facility with a free piston.

displacement by fixing it on the floor or on the foundation, because the bearing elements of the building may be overloaded: at a pressure of 200 MPa, the horizontal force for the small facility described above is approximately $4 \cdot 10^5$ N.

The test section of the facility can be rigidly connected to the nozzle; in this case, however, it should be borne in mind that the shear-induced acceleration of the tested models and gauges can reach 3000 m/sec^2 and more. A flexible telescopic connection between the nozzle and the test section, the working stroke being greater than the displacement length, can also be provided in the facility. In this case, the center of mass of the facility has to be on the axis of symmetry of the conical nozzle.

If the shear accelerations and telescopic connections are inadmissible, it is possible to construct a facility with active suppression of barrel oscillations, similar to that considered in the present paper.

2. Displacements of piston-aided facilities in the course of gas compression can be prevented by providing simultaneous opposing motion of identical pistons in identical barrels, which would ensure a constant coordinate of the common center of mass. This principle was realized in the test-gas source designed for an AT-303 wind tunnel. In this wind tunnel, the gas is compressed by identical opposing pistons in two barrels with their axes aligned along one straight line [6]. Locking pistons may be used instead of usual pistons in a facility similar to AT-303, if the piston are set into motion simultaneously.

A principal advantage of facilities with locking pistons over valve-aided systems with usual pistons is the fact that the pistons remain motionless after the first cycle of compression in the first case. The use of valve systems with usual pistons in a symmetric case leads to dynamic disbalance and oscillations of the facility because the "parasitic oscillations" of the pistons become asymmetric.

Apart from pistons moving toward each other, it is possible to use identical locking pistons moving away from each other. In this case, compression results in formation of two (instead of one) identical volumes of the compressed gas at the opposite ends of the facility (the total mass of the gas is the same as that in the previous scheme). For gas-dynamic experiments in the test section, however, only one volume is used; the second volume and, hence, the second barrel are used only to eliminate the displacement.

To reduce the axial size and increase the efficiency of using the compressed gas, it seems reasonable to use an asymmetric scheme of a dynamically balanced facility. The possibility of this engineering solution is predicted by the equation of motion of a heavy piston during gas compression in a motionless barrel of a gas-dynamic facility shown schematically in Fig. 1. The piston is considered to be a solid cylinder of mass m and radius a, which moves without friction under the action of gas pressure along the x axis (in the direction of decreasing coordinate). The strains of the facility elements are small ($\varepsilon \approx 10^{-3}$), and the force of friction of the piston on the barrel is approximately 10^{-2} of the value of the accelerating (decelerating) force; in the first approximation, therefore, these quantities can be neglected. In the initial position, the end face of the piston is located in the cross section $x = l$, the barrel volume ahead of the piston is closed in the cross section $x = 0$, and the initial parameters of the compressed gas are $p(l) = p_0$ and $T(l) = T_0$.

When the barrel volume behind the piston is rapidly connected at the initial time ($t = 0$) with a receiver of a much greater volume (the driving gas pressure is $p_{\rm dr} > p_0$), the piston, which was at rest, is set into accelerated motion. The experience of working with facilities of this type shows that the gas pressure $p(x)$ can be assumed to follow the equilibrium adiabat curve with accuracy sufficient for practical purposes of calculating the heavy piston dynamics. The reasons are not only the subsonic velocity of the piston (not more than 100 m/sec), but also by a small time of growth of the compressed gas temperature (such that heat loss can be neglected). For $p(x) = p_{\rm dr}$, the piston velocity reaches the maximum value, the acceleration changes its sign, and gas compression is continued at the expense of the kinetic energy accumulated by the piston. At the moment of piston locking ($t = \tau$), it has the maximum acceleration, and the velocity acquires a zero value persisting at $t > \tau$.

The change in the coordinate of the end face of the piston in the time interval from 0 to τ is described by the equation

$$m \frac{d^2x}{dt^2} + \pi a^2 [p(x) - p_{\rm dr}] = 0$$

with the initial conditions $x(0) = l$ and $p(x(0)) = p_0$.

Introducing the dimensionless coordinate $\xi(t) = x/l$, we obtain

$$lm \frac{d^2\xi}{dt^2} + \pi a^2 [p(\xi) - p_{\rm dr}] = 0 \qquad (\xi(0) = 1, \quad p(\xi(0)) = p_{\rm dr}),$$

where ξ is a quantity inverse to the degree of gas compression ahead of the piston.

For different piston systems with identical values of the parameters a, $p_{\rm dr}$, p_0, and T_0, the dependences $\xi(t)$ coincide if the barrel length and the piston mass of each system satisfy the relation

$$lm = \text{const}, \tag{1}$$

which can be called the condition of simultaneous motion of the pistons (simultaneous changes in their dimensionless coordinates).

The single-valuedness of the function $p(\xi)$ implies simultaneous growth rates of pressures ahead of the pistons in different barrels and, hence, simultaneous action of forces on these barrels. Thus, if Eq. (1) is satisfied, simultaneous actuation of pistons of different masses in the opposite directions in two coaxially attached barrels with identical cross sections but different lengths (the so-called common barrel) is expected to lead to a simultaneous increase and mutual elimination of forces acting on the barrel. During gas compression in such a facility, the center of inertia of the common barrel has to remain at rest.

Providing identical initial parameters of the gas (p_0 and T_0) in the volumes ahead of the pistons is not a complicated technical problem. Simultaneous acceleration of the pistons is ensured by rapid operation of the actuating valve with a sufficiently large cross-sectional area, which drives the gas at a pressure $p_{\rm dr}$ from the driving receiver into the space between the pistons.

3. As the forces $F(t)$ acting on the pistons are identical (with accuracy to the sign), the absolute value of the momentum of the main piston P in the absence of friction is equal to the momentum of the compensator piston $P_{\rm comp}$. By definition,

$$P(t) = \int_0^t F(t)\,dt = \pi a^2 \int_0^t [p(t) - p_{\rm dr}]\,dt = P_{\rm comp}(t); \tag{2}$$

$$W(t) = \frac{P^2}{2m}, \qquad W_{\rm comp}(t) = \frac{P_{\rm comp}^2}{2m_{\rm comp}} = \frac{P^2}{2Nm}. \tag{3}$$

It follows from Eqs. (3) that the kinetic energy of the main piston $W(t)$ is N times ($N = m_{\rm comp}/m = l/l_{\rm comp}$) the kinetic energy of the compensator piston $W_{\rm comp}(t)$ (i.e., the energy and the work of compression are proportional to the compressed volumes, as it could be expected).

In piston-aided facilities, the friction forces of the pistons are proportional to the pressure difference, because the determining factor is dry friction of sealing rings on the barrel; the intensity of dry friction is calculated in accordance with the Coulomb law. If the structure of the seals and the sealed perimeter of the main and auxiliary pistons are identical, the absolute values of the friction forces are also identical: $f = f_{\rm comp} = \delta F$. Assuming that the proportionality factor δ is a small positive number of the order of 10^{-2} and taking into account friction, we can transform Eqs. (2) to

$$\int_0^t (F - f)\,dt = (1 - \delta) \int_0^t F(t)\,dt = (1 - \delta) P(t) = (1 - \delta) P_{\rm comp}(t).$$

Despite the changes in the momentums, they are equal to each other at each time instant, and simultaneous operation is not violated. If the vectors of the friction forces are drawn from the center of inertia of the common barrel lying on the axis of symmetry, a symmetric pair is formed. Hence, under the action of symmetric friction forces, the center of inertia of the barrel also remains at rest.

As δ is a small quantity, Eq. (3) predicts equal relative changes in the kinetic energy of the pistons because of identical friction forces:

$$\frac{\Delta W}{W} = \frac{\Delta W_{\rm comp}}{W_{\rm comp}} = -2\delta + \delta^2 \approx -2\delta.$$

Hence, the relative losses of the kinetic energy of each piston are approximately equal to the doubled relative loss of momentum. Obviously, an increase in the friction force leads to a smaller work of compression and, hence, to somewhat lower gas parameters reached. This reduction can be compensated by increasing the driving pressure $p_{\rm dr}$ by $f/(\pi a^2)$.

4. In the case of asymmetric friction of the pistons, the balance is violated and the center of mass of the common barrel is shifted. This shift, however, can be minimized.

Let the friction of the working piston of mass m on the barrel correspond to the case $f = \delta F$, where the momentum of the working piston is determined by Eq. (4), and the oppositely directed friction force of the compensator is $f'_{\rm comp} = (\delta + \gamma)F$ (γ is a small quantity of the same order as δ). When the facility is in operation, the center of inertia of the common barrel is accelerated under the action of the resulting force $\Delta F = \gamma F$ directed toward the compensator. The force driving the compensator piston is smaller than the force driving the working piston by the same value ΔF (absolute value). As a result, the following difference in momentums is obtained in the case of asymmetric friction:

$$\Delta P_{\rm comp} = -\gamma P. \qquad (4)$$

In contrast to the case of symmetric friction, the velocity of the auxiliary piston decreases by the relative quantity γ, and the system operation is no longer simultaneous: the maximum compression point is first reached by the working piston and then by the auxiliary piston. As a result, the common barrel is first shifted forward and then backward.

Correcting the mass of the auxiliary piston obtained from Eq. (1), one should be careful to avoid violating the simultaneous motion condition owing to asymmetric friction of the pistons.

In Eq. (3), the kinetic energy is a function of two variables: momentum and mass. Using the expression of the full differential via the partial derivatives and increments of the variables, we can obtain expressions for small changes in the kinetic energy of the compensator piston $W_{\rm comp}$ corresponding to asymmetric friction in some particular cases:

$$\Delta W_{\rm comp} = \frac{P}{m_{\rm comp}}\,\Delta P - \frac{P^2}{2m_{\rm comp}^2}\,\Delta m_{\rm comp}. \qquad (5)$$

Substituting the parameters for the above-described case, where the compensator piston mass is fixed at a level corresponding to Eq. (1) [i.e., $\Delta m_{\rm comp} = 0$, and the momentum acquires the increment predicted by Eq. (4)] into Eq. (5), we find the change in the kinetic energy of the piston:

$$\Delta W'_{\rm comp} = -\gamma P^2/m_{\rm comp} = -2\gamma W_{\rm comp}.$$

If the compensator piston mass is changed by $\Delta m = -\gamma m_{\rm comp}$, system operation becomes much more synchronous, because the acceleration of the compensator piston at the stage of its acceleration to the maximum velocity reaches a value practically corresponding to symmetric friction. By substituting this increment of mass and also the increment of momentum (4) into Eq. (5), we obtain the change in the kinetic energy of the compensator piston

$$\Delta W''_{\rm comp} = -\gamma W_{\rm comp}.$$

Substituting a halved value of the mass $\Delta m_{\rm comp} = -2\gamma m_{\rm comp}$ into Eq. (5), we obtain the following equation for the same increment of momentum:

$$\Delta W^*_{\rm comp} = 0.$$

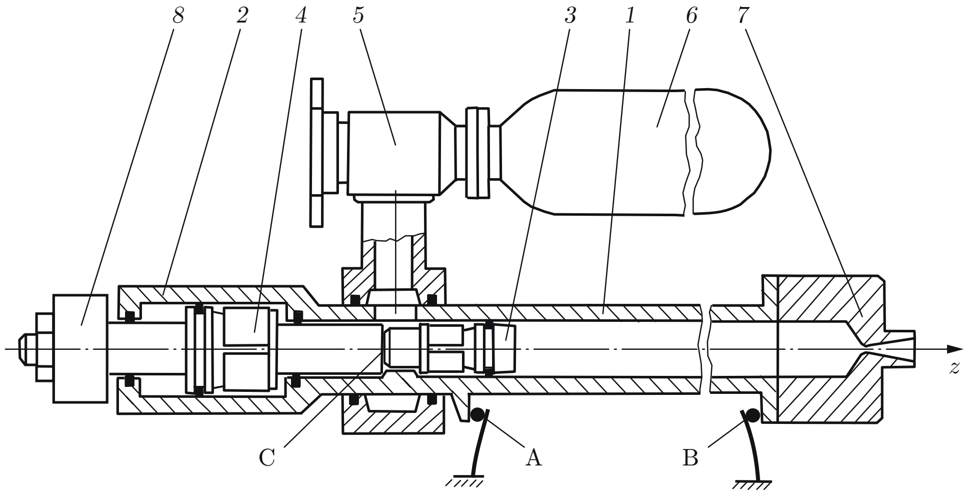

Fig. 2. Layout of a gas-dynamic facility with a momentum compensator: 1) main barrel; 2) compensator barrel; 3) main piston; 4) auxiliary piston; 5) actuating valve; 6) receiver; 7) settling chamber; 8) control load; the letters denote the gauge of displacement of the center of mass of the common barrel (A), the gauge of displacement of the settling chamber (B), and the plane of contact of the pistons (C).

In this case, the kinetic energy of the compensator piston is sufficient to obtain the maximum parameters of the gas identical to those in the test section. With a decrease in mass, however, the velocity of the auxiliary piston increases; hence, the gas in the compensator is compressed faster than the gas in the test section. A further decrease in mass $m_{\rm comp}$ does not seem to be reasonable.

Thus, to compensate for the effect of adverse factors caused by asymmetric friction of the pistons, it is reasonable to reduce the mass of the compensator piston $m_{\rm comp}$ calculated by Eq. (1). The optimal mass $m^*_{\rm comp}$ is in the range

$$(1-2\gamma)m_{\rm comp} \leqslant m^*_{\rm comp} \leqslant (1-\gamma)m_{\rm comp}. \tag{6}$$

The value of $m^*_{\rm comp}$ can be determined more exactly by choosing an appropriate mass of the compensator piston in a particular facility.

5. Based on the above-described reasoning, we designed and manufactured a gas-dynamic facility with a momentum compensator, which is schematically shown in Fig. 2. The main barrel of radius $a = 25$ mm with a cavity of length $l = 1.71$ m contains the working (main) locking piston of mass $m = 4.25$ kg. The main barrel is coaxially attached to the compensator barrel of length $l_{\rm comp} = 0.143$ m containing the auxiliary locking piston. The cross-sectional area of the ring-shaped cavity of the compensator is $\pi a^2 = 19.6$ cm^2; hence, the volume of the working cavity of the barrel is 3360 cm^3, and the volume of the ring-shaped cavity of the compensator is 280 cm^3, i.e., the ratio of the lengths, masses, and volumes is $N = 12$.

In the initial state, the cylindrical cavity of the barrel and the ring-shaped cavity of the compensator are filled by a compressible gas with the parameters $p_0 = 1$ MPa and $T_0 = 290$ K; the action of pressure in the cavities ensures the initial force contact of the pistons in the plane C (see Fig. 2). After operation of the fast-response actuating valve, the receiver supplies the driving gas to the barrel cavity between the pistons. Being compressed by the main piston, the test gas is sustained by the latter in the settling chamber whose axial nozzle orifice is so small that gas exhaustion does not affect the dynamics of its compression. To optimize the mass of the compensator piston, its structure includes a control load. The dynamics of the facility during gas compression was studied in experiments with an initial value of the driving gas pressure in the receiver $p_{\rm dr} = 13$ MPa, which ensured compression of the gas (with the above-indicated parameters) to pressure of 200 MPa and higher. In some experiments, the mass of the control load of the compensator piston was changed.

The displacement of the facility elements in the course of gas compression was registered with the use of elastic steel plates clamped on a motionless foundation. The natural frequencies of the plates were 590 and 610 Hz. The signal was formed by a bridging scheme of resistance strain gauges glued onto the plates; after amplification,

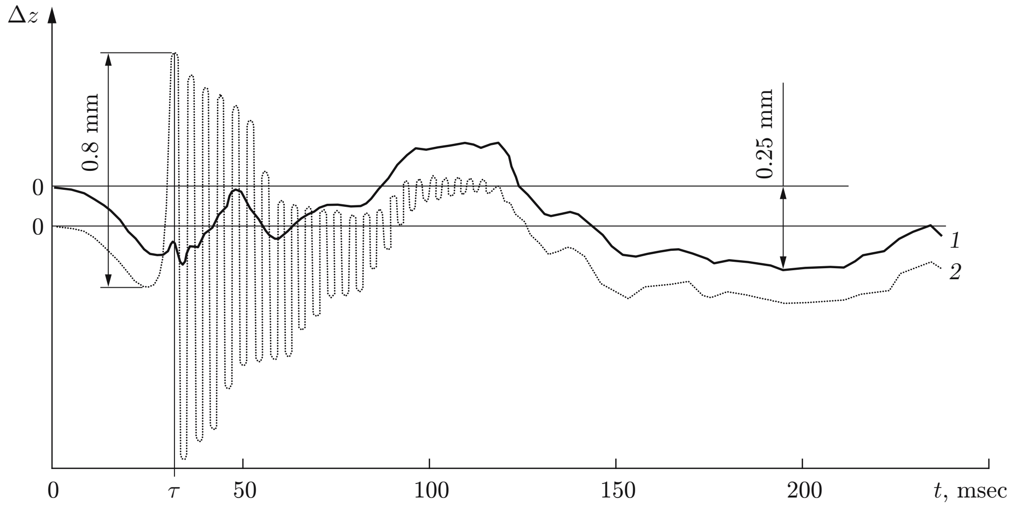

Fig. 3. Oscillograms of motion of the center of mass of the barrel (1) and settling chamber (2) during gas compression to a pressure of 200 MPa.

the signal was fed to an oscillograph. The gauge A registered the change in the coordinate in the plane of the center of mass of the common barrel. The gauge B recorded the settling chamber displacement (see Fig. 2).

As the sealed perimeter of the compensator piston is twice the sealed perimeter of the working piston, the friction force of the compensator piston is twice the friction force of the working piston. The absolute value of the sum of the vectors of the piston friction forces acting on the common barrel is γF. By estimating the friction forces of bronze sealing rings, we obtain $\gamma = 0.02$.

As was indicated above, the mass of the working piston is $m = 4.25$ kg; it follows from Eq. (1) that the mass of the compensator piston in the first approximation is $m_{\rm comp} = 12m = 51$ kg. Registration of displacements showed that the stroke of the working piston takes a shorter time than the stroke of the compensator piston, as it could be expected for this value of the compensator piston mass. As a result, the center of mass of the common barrel is first shifted forward (toward the settling chamber) and then backward (toward the compensator). A decrease in the compensator piston mass by 2 kg in accordance with Eq. (6) does not exert any significant changes in the dynamics of barrel motion. A decrease in the compensator piston mass to 44 kg leads to the opposite effect: the maximum pressure is first reached in the compensator and then in the settling chamber, and the facility moves first backward and then forward.

By means of targeted regulation of the compensator piston mass, we determined its value close to the optimal one: $m^*_{\rm comp} = 45.8$ kg. This value corresponds to the oscillograms in Fig. 3, which are shifted along the ordinate: oscillogram 1 shows the displacements of the center of mass of the common barrel $\Delta z_1(t) = z_1(t) - z_1(0)$, and oscillogram 2 shows the displacements of the settling chamber $\Delta z_2(t) = z_2(t) - z_2(0)$. The positive direction of Δz corresponds to the direction of motion of the working piston.

Oscillogram 2 reveals an increase in elastic deformation of the barrel and the axial tensile force proportional to elastic deformation during gas compression from the time $t = 0$ to the moment of locking of the pistons $t = \tau$ (in Fig. 3, $\tau = 31$ msec). The maximum value of the axial tensile force is determined by the greatest value of pressure (at $t = \tau$) and reaches approximately $4 \cdot 10^5$ N. An analysis of the oscillograms shows that the elastic stretching of the barrel at $t = \tau$ is $\Delta L = 0.8$ mm. Calculating the ratio of the maximum values of the force and barrel stretching at the locking moment, we find the barrel rigidity $k = 5 \cdot 10^8$ N/m, which almost coincides with the result predicted by a fairly simple calculation of this quantity.

When the piston is locked, redistribution of axial loads occurs, because the piston locks the settling chamber like a plug. Static stretching of the barrel under the action of the pressure of the compressed gas is retained only on a small segment between the zone of piston locking and the settling chamber; the major part of the barrel is unloaded like a spring with an instantaneously removed load. At $t > \tau$, the settling chamber–piston–barrel system passes to the regime of decaying elastic oscillations with a frequency of 330 Hz and a period of oscillations $\theta \approx 3$ msec. As the shape of the axis of the decaying sinusoid 2 is identical to the shape of oscillogram 1, the resultant pattern of decaying oscillations is somewhat distorted.

An analysis of oscillogram 1 shows that the center of mass of the common barrel moves backward with acceleration of the order of 1 m/sec^2 at the initial stage under the action of the resulting friction force. When the displacement reaches 0.2 mm, the motion direction is changed. Some fluctuations of the center of mass with an amplitude of about 0.05 mm are registered in the vicinity of the point $t = \tau$. In the interval $\tau < t < \tau + \theta/2$, the velocity of the center of mass acquires the maximum value, which is substantially smaller than 0.1 m/sec; hence, the kinetic energy of the common barrel with a mass of 150 kg is less than 1 J. The maximum displacement of the center of mass was 0.25 mm during the measurement time equal to 200 msec.

6. The experimental value of the optimal mass of the compensator piston $m^*_{\rm comp}$ is smaller than the value obtained in the first approximation by 10.2%. This difference is significantly greater than the decrease in mass following from inequalities (6) (4%). Such a large difference is caused by conversion (at high pressures) of the energy of the compressed gas to the energy of elastic deformation of the elements of gas-dynamic facilities. This fact can be supported by appropriate estimates. In this case, friction is neglected, because its influence on the energy parameters of the system is insignificant.

Estimating the maximum kinetic energy of the working piston during adiabatic compression of the gas to a pressure of 200 MPa with allowance for real gas effects at high densities [7], we obtain 18.5 kJ (the corresponding piston velocity is 93 m/sec).

At $t = \tau$, the test gas occupies the minimum volume; high radial loads and, hence, radial deformation of the barrel are focused in the settling chamber whose length is smaller than the barrel length by an order of magnitude. Thus, the main part (≈90%) of the energy of elastic deformation is the energy of axial stretching of the barrel at a pressure of 200 MPa, which is equal to $k(\Delta L)^2/2 = 0.16$ kJ. Assuming that the total elastic energy of the barrel is 0.18 kJ, we can conclude that approximately 1% of the kinetic energy of the piston becomes converted to the elastic energy of the working barrel.

As the ratio of the piston masses in the first approximation is $N = 12$, it follows from Eq. (3) that the kinetic energy of the compensator piston is 1/12 of the kinetic energy of the main piston (i.e., approximately 1.5 kJ). The main "absorber" of the elastic energy in the compensator structure is the axial stem bracing the elements of the compensator piston. The calculation show that the elastic energy of the stem is 0.09 kJ for the maximum parameters of compensator piston deceleration, and the elastic energy of the compensator as a whole is approximately 0.11 kJ, i.e., about 7% of the kinetic energy of the compensator piston become converted to the energy of elastic deformation of the compensator.

Thus, owing to elastic deformations, the relative "losses" of the kinetic energy of the pistons differ by an order of magnitude. If the coefficients of the relative "elastic" losses of the kinetic energy of the pistons are identical, their dimensionless coordinates change simultaneously. In the real facility considered (see Fig. 3), the difference in these coefficients is approximately 7% − 1% = 6%, which leads to system desynchronization. It follows from Eq. (5) that this difference can be compensated by increasing the kinetic energy of the compensator piston through reducing its mass by 6%, with identical momentums of the pistons.

If the piston mass is reduced by 4% more, the first inequality in (6) predicts that the total decrease in the mass of the auxiliary piston is approximately 10%, which is in good agreement with the experimentally obtained value of 10.2%.

7. Simultaneous operation of the pistons is also affected by other factors, for instance, the difference in relative heat losses in the main and auxiliary barrels, though friction and elasticity are the main reasons for violation of simultaneous motion. It should be noted that the adverse effect can be compensated in all cases by changing the mass of the auxiliary piston.

Let us consider why a facility with the ratio of masses and axial sizes of the cavities ahead of the pistons $N = 12$ and with the diameter of the cavity of the compensator barrel being greater than the diameter of the main barrel was not created. In this case, possibly (without the ring-shaped cavity), the friction forces will be symmetric. For instance, a twofold increase in the compensator barrel diameter leads to a fourfold increase in its area; hence, the initial and final pressures are reduced by a factor of 4, because the forces acting from the gas should be retained.

At pressures up to 50 MPa, it is possible to use a system with an increased diameter of the compensator barrel, because the gas can be considered as an ideal gas at these pressures (deviations of gas parameters from the classical equations do not exceed 1%). At a pressure of 200 MPa, the real gas effects have to be taken into account. As an illustration, let us estimate the forces acting on the pistons from the gas in a facility with a doubled diameter of the compensator barrel. In the initial state, these forces are identical if the gas pressure in the main barrel is 0.8 MPa and the gas pressure in the compensator barrel is 0.2 MPa. After adiabatic volume compression of the gases in both cavities by a factor of 35, the pressure in the main barrel increases to 200 MPa, and the pressure in the compensator barrel increases to 29 MPa. We can easily see that the equilibrium is violated after adiabatic compression, because the force acting on the working piston is greater than the force acting on the compensator piston by a factor of 1.72. This fact was the reason why the design of facilities with different cross-sectional areas of the barrels was canceled.

Thus, the research performed confirms that the method of active suppression of "parasitic oscillations" in gas-dynamic facilities with a free piston is fairly effective. Using a compensator for momentum, one can reduce undesirable displacements of the barrel of the facility during its operation by two orders of magnitude and almost eliminate them, because the barrel displacement becomes smaller than the amplitude of its elastic deformation.

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