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# USE OF HIGH PRESSURES FOR SOLVING PROBLEMS OF HYPERSONIC AERODYNAMICS

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*The use of high pressures in a hypersonic aerodynamic experiment is founded on physical grounds. Calculation results of Mach and Reynolds numbers reachable at the line of gas condensation are given as functions of the temperature and pressure in the plenum chamber. Approaches to solving problems of designing ultrahigh-pressure facilities that ensure outflow with pressures up to* 20,000 atm *are described. These problems include the stop of the first-stage piston at the point of maximum pressure, suppression of the reaction force, provision of normal operation of seals of the moving piston, and reduction of friction forces in the seals. The principles considered are used in an actually operating facility.*

On the occasion of the 100th anniversary of M. A. Lavrent'ev, it seems pertinent to summarize the long-term work on the physical foundations, design, and manufacturing of sources of high-pressure gases for hypersonic wind tunnels in the pages of a journal founded with the immediate participation from M. A. Lavrent'ev. This direction of experimental aerodynamics received invaluable support of M. A. Lavrent'ev in early 1970s, without which the scientific ideas would not, probably, be put into practice.

The idea of using high (up to 20,000 atm) pressures of a gas in the plenum chamber in an aerodynamic experiment was first put forward in the late 1960s by M. A. Plotnikov [1] who worked at that time in Research Institute-1 in the group of Prof. E. I. Shchetinkov, one of the founders and proponents for using supersonic combustion in air-breathing engines of hypersonic flying vehicles (HFV) for Mach numbers from 8 to 20 [2]. At the same time, the papers of Soviet and American scientists posed the question of the necessity of developing of a reentry HFV with an aircraft-type take-off and landing, capable of taking payloads to the orbit of an Earth's satellite and back. It is expected that, with the use of these vehicles, the cost of putting payloads to orbit, which is the key economic factor of using spacecraft systems in practical activity outside of the military field, will decrease by 5 to 8 times. One of the main obstacles for the use of such systems is the absence of knowledge on the complex of fundamental phenomena that accompany and ensure the HFV flight. The existing hypersonic aerodynamic facilities do not allow one to create conditions necessary for testing the corresponding HFV models [4]. This is especially true for reproduction of Reynolds numbers and ensuring test-gas purity and run time.

At the end of the 1960s, M. A. Plotnikov and his colleagues developed tables of thermodynamic functions of nitrogen [1] from which it followed that the transition to the pressure region up to 10,000–20,000 atm leads to an additional contribution to the gas enthalpy equivalent to a temperature increase by approximately 1000 K per each 10,000 atm. On the one hand, this offered a possibility of increasing the

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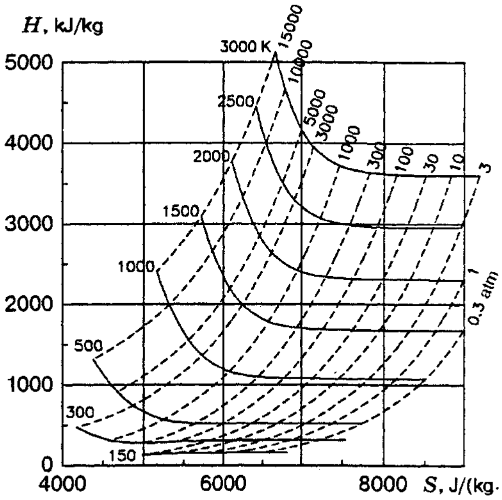

Fig. 1. Enthalpy–entropy diagram for nitrogen: the solid and dashed curves refer to isotherms and isobars, respectively.

Reynolds number due to the increase in the gas density and outflow velocity, on the other hand, this allowed one to expect an increase in Mach numbers reachable at the condensation line up to full-scale values without passing to a region of temperatures in the plenum chamber, which are incompatible with the capabilities of construction materials.

In 1969 it was decided to perform the corresponding experiments and use the available experience of creation of high-rate hydrodynamic equipment to develop a device that would ensure exhaustion of a gas with constant parameters at pressures up to 10,000–15,000 atm. Following the proposal of O. V. Lyzhin, the Scientific and Technical Council of TsAGI made a decision to finance these works. In spring 1972, the experiments performed at the Institute of Hydrodynamics and the special Design and Technology Institute of High-Rate Hydrodynamics of the Siberian Division of the Russian Academy of Sciences allowed us to find and put into practice physicotechnical solutions that made it possible to create such a device. Nevertheless, it was not a common belief that these solutions could be combined in an operating facility, and only the resolute interference of M. A. Lavrent'ev made possible a successful continuation of the work and its finalizing in 1975 by the creation of a record facility demonstrating all the advantages of the new approach to creating gas sources for hypersonic wind tunnels.

**Introduction.** The physical basis for using high pressures in a gas-dynamic experiment is the $(S–H)$-diagram ($S$ is the entropy and $H$ is the enthalpy). Figure 1 shows such a diagram for nitrogen, which was calculated using the semi-empirical equation of state proposed by Zykov and Sevost'yanov [5], which yields results that practically coincide with the data of [1]. For pressures higher than 1000 atm, a sharp rise of isotherms is observed with increasing pressure in a dense gas. This is evidence of an additional contribution to enthalpy made by molecular repulsion forces, which becomes apparent in the case of high densities. A dramatic decrease in gas compressibility is observed in the same range of parameters.

The Mach (M) and Reynolds (Re) numbers obtained at the line of equilibrium condensation of nitrogen were first calculated in 1969 [6]. The diagram from this paper shown in Fig. 2 indicates that an insignificant decrease in the reachable Mach numbers is observed in the low-pressure range with increasing pressure. If the pressure is higher than 2000 atm, its increase leads to a significant increase in Mach numbers. For example, on the isotherm of 1800 K, the Mach number increases from 14.6 to 17.4, which corresponds to an increase in temperature in the plenum chamber by approximately 450 K. Taking into account the nonequilibrium character of condensation with gas outflow under actual conditions, it is possible to reach M = 20 at this isotherm. Based on this fact, the value of 2000 K was assumed to be the maximum design temperature in the plenum chamber in the facility to be created. It was also decided to reach working pressures of 10,000–

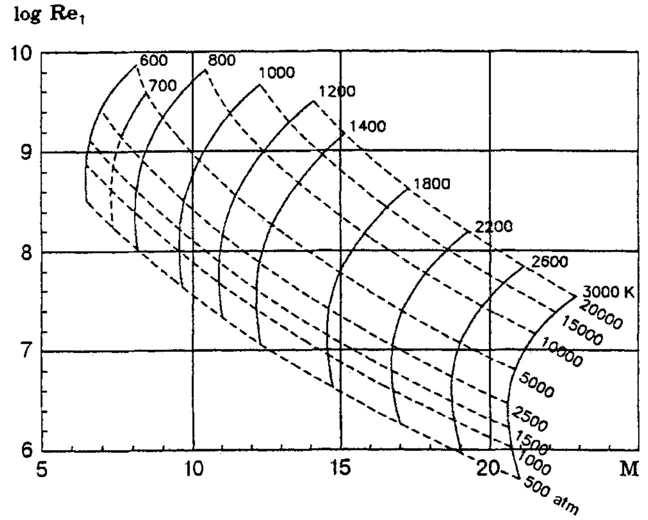


Fig. 2. Mach–unit Reynolds number diagram at the line of condensation for nitrogen: the solid and dashed curves refer to isotherms and isobars, respectively.

15,000 atm. In this case, real gas effects are clearly manifested, and the additional contribution to enthalpy is equivalent to an increase in the stagnation temperature by no less than 500 K.

**Choice of the Layout. Construction and Operation of the Stage of Preliminary Compression. Operating Principle of the Facility.** From the very beginning, it was decided to avoid using high-temperature heaters and to employ only adiabatic compression for reaching the necessary parameters of the gas. This decision guarantees a high homogeneity of the gas temperature. The monotonic increase in the parameters and the short time of gas preparation and its interaction with the walls ensure an almost complete absence of contamination, which arises in other facilities due to gas dissociation and entrainment of the material of the heating elements and walls. In addition, to maintain the outflow parameter constant during the overall test regime, it was decided that the gas should be forced out from the plenum chamber using a special piston-multiplier.

The calculations showed that a pressure of 15,000 atm at a temperature of 2000 K can be obtained by compressing the gas from the initial state with a pressure of 10.8 atm at room temperature (293 K). The overall degree of volume compression is 63.7, and only a 2.26-fold compression is needed to increase the pressure from 2000 to 15,000 atm. To obtain a pressure of 10,000 atm at the same temperature, it is necessary to compress the gas from the initial state with a pressure of 7 atm. In this case, the overall degree of compression is 82.5, and the factor of 2.03 refers to the pressure increase from 2000 to 10,000 atm.

Such a low compressibility of the gas at pressures above 2000 atm allows us to divide the compression process between two stages. The first stage is a system of adiabatic compression with a free heavy piston [7], and the other is a pressure multiplier, which ensures an approximately twofold compression of the gas with its subsequent forcing out from the plenum chamber. The choice of the intermediate pressure of 2000 atm is also determined by the fact that the construction and operating principles of hermetic seals between the moving elements within the range up to 2000 to 3000 atm are significantly different from those used at higher pressures.

One serious problem arising in interaction of two stages of the facility is related to the fast recoil of the first-stage piston, which is caused by a restoring force of 2 tons acting on each squared centimeter of the butt-end surface of the piston when it stops. The calculations showed that the entire precompression cycle in the first stage (with a piston diameter of 50 mm and its mass of about 6 kg) takes 33 msec, and the last doubling of pressure occurs during less than 1 msec [8]. Obviously, a twofold decrease in pressure during the recoil occurs during the same time, which is insufficient for starting and actuating the second stage. The use of a non-return valve separating the gas in the plenum chamber from the first stage would be highly undesirable. The experience of operation of the Long Shot facility in Belgium is available, wherein a device consisting of 40 individual valves is used to keep the gas in the plenum chamber [9]. Though the wind tunnel operates only with inert nitrogen, the valves have to be replaced after several runs because of the intense erosion.

It was assumed from the very beginning of investigations that the would-be facility, which was called A-1, will operate with air at least in some regimes. Therefore, it was necessary to find a principally new approach. The problem was solved by means to a specially developed self-wedged system [10], which allows one to stop the first-stage piston at the point of the maximum pressure and maintain the gas in the compressed state for a long time. The piston leans against the smooth walls of the barrel and becomes absolutely free when the pressure is removed. The high stresses in the barrel, piston, and wedges can be made lower than the gas pressure by choosing a correct length of the piston. The self-wedged system simultaneously offered a solution for the problem of suppression of oscillations that should arise when the first-stage piston stops.

The moment of piston stopping is associated with another problem: appearance of a reaction force reaching 40 tons. For the chosen position of the first stage, this force will lead to lateral displacements of the facility, which cannot be allowed in an aerodynamic experiment. The transfer of this force to the foundation would lead to strong vibrations and destruction of the building. The momentum generated by the reaction force was almost completely suppressed by a special compensator, which is a continuation of the barrel at the tail part of the precompression stage of the same cross-sectional area but smaller length. The second self-wedged piston, whose mass is greater than the mass of the main piston, is located in this part of the barrel. The operating principle of the compensator is based on the similarity of motion of two pistons under the action of identical pressures, which act on an identical surface, under the condition that the ratio of piston masses is inversely proportional to barrel lengths. To clarify this condition, we consider the equation of motion of the piston under the action of gas-pressure forces in the receiver and plenum chamber. The friction force is ignored since it is small as compared to gas-pressure forces. Introducing the dimensionless coordinate of the piston $\xi = x/l$, we obtain

$$ml \frac{d^2\xi}{dt^2} = [P_{\rm rec} - P_{\rm g}(\xi)]S,$$

where $l$ is the total length of the piston stroke, $P_{\rm rec}$ and $P_{\rm g}$ are the gas pressures in the receiver and under the piston, $m$ is the piston mass, and $t$ is the time.

Obviously, if the pistons are accelerated by the same gas pressure in the receiver $P_{\rm rec}$ and the pistons start simultaneously and have identical initial gas pressures $P_{\rm g}$ and areas $S$, then the values of $\xi$ and $P_{\rm g}$ for both pistons are identical at the same times for identical values of $ml$. The equality of the pressures $P_{\rm g}$ means the equal and simultaneous action of oppositely directed forces acting on the facility.

The ratio of the masses in the actual construction was about 10. In this case, the additional energy of the receiver spent on operation of the compensator was less than 10% of the useful energy. In the course of experiments, by choosing exactly the compensator piston mass, we succeeded in decreasing the shift of the center of mass of the first stage of the facility to 0.25 mm, which is four times as small as its longitudinal elastic deformation under the action of working pressures.

The general layout of the gas-dynamic facility A-1 is shown in Fig. 3. The facility consists of an aerodynamic nozzle, a plenum chamber with a side window, and a pressure multiplier with small and large pistons and a shank. The latter is in contact with a fast-response valve separating the receiver from the large cylinder of the multiplier. The side window of the plenum chamber is connected to the stage of preliminary compression equipped by two pistons, a receiver, and a starting valve.

The large cylinder of the multiplier is divided by a calibrated orifice (throttling jet nozzle) into two cavities filled by a liquid, which is isolated from the gas by a floating dividing piston.

To ensure a given law of pressure variation in the plenum chamber, the shank of the multiplier has a contoured section located inside the jet nozzle through which the driving liquid is injected. As the piston

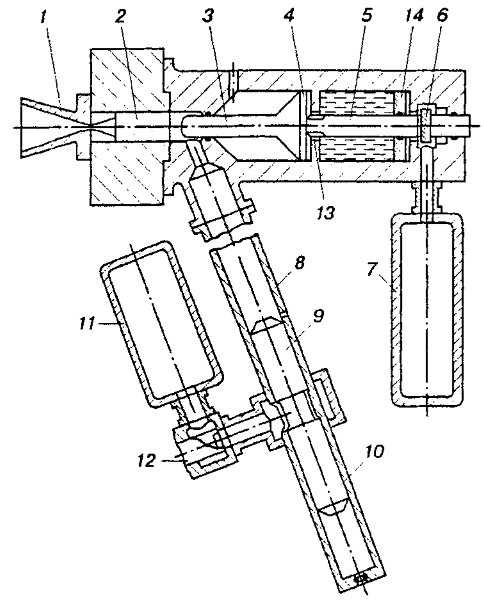


Fig. 3. Layout of the A-1 facility: aerodynamic nozzle (1); plenum chamber (2); small (rod) and large pistons of the multiplier (3 and 4, respectively), shank (5), fast-response valve (6), receiver of the pressure multiplier (7), stage of preliminary compression (8), light and heavy pistons of the stage of preliminary compression (9 and 10), receiver of the first stage (11), starting valve (12), throttling jet nozzle (13), and dividing piston (14).

moves toward the nozzle, the cross-sectional area of the jet nozzle changes. For example, to provide a rapid compression and a constant pressure during forcing the gas out through the nozzle, the shank should have two sections in the form of cylinders of different diameters.

Initially, the piston of the pressure multiplier is located in the extreme right position so that the plenum chamber is connected to the barrel of the first stage of adiabatic compression through the window in the side wall. The shank of the piston leans against the valve pressed to the multiplier body by the gas from the receiver. In the stage of preliminary compression, the light and heavy pistons are pressed to each other by an identical initial pressure of the gas in the plenum chamber and in the cylinder of the compensator. Necessary pressures are generated in the plenum chamber and receivers.

To start the facility, the starting valve of the first stage opens access of the gas from the receiver into the cavity between the first-stage pistons. The pistons start to move in the opposite directions, compressing the gas, each in its own barrel. The simultaneous motion of the pistons eliminates the lateral impulse acting on the facility in the case of motion of only one piston. At the end of the compression path, the pistons are wedged and close the gas in the plenum chamber with a pressure of about 2000 atm and a temperature up to 1500 K. The compressed gas acting from the plenum chamber on the multiplier piston provides a force that opens the fast-response valve.

The gas from the receiver of the pressure multiplier pushes the dividing piston, which makes the liquid move from one cavity of the large cylinder to the other through the jet nozzle and sets the piston of the multiplier into motion. Moving to the nozzle, the piston separates the gas in the plenum chamber from the first-stage barrel, compresses it to a given value, and forces it out through the nozzle.

High pressures at the low-pressure side of the multiplier (up to 1000–1500 atm) ensure an almost inertia-free motion of its piston. The maximum velocity is reached during a time of about 1 msec, and the whole cycle of test-gas preparation lasts about 70 msec. The small time of gas residence in a state with high parameters allows one to work with an open nozzle, i.e., avoid using any valves or closing devices that contact the heated gas flow. The calculations showed that the total losses in the test gas do not exceed 20%.

**High-Pressure Unit.** One serious problem that arose in the process of creation of the A-1 facility was the seal between the small cylinder of the multiplier (plenum chamber) and the piston (rod). First of all, the seal should not be exposed to the thermal action of the gas heated to a high temperature. This means that the seal should be located on the cylinder rather than on the rod. In this case, it is possible to organize cooling of the gas contacting the seal in a narrow gap between the plenum-chamber wall and the rod.

The displacement $u$ of the wall of the thick-walled cylinder with an internal radius $r$ under the action of an internal pressure $P$ in the absence of external pressure and axial stresses is determined by the relation $u/r \geqslant (P/E)(1+\mu)$, where the equality sign corresponds to the case of infinite radius of the external wall.

The main formulas for rod deformations have the form $(\Delta r/r)_{\text{int}} = -(P/E)(1-2\mu)$ for sections in the plenum chamber (triaxial compression) and $(\Delta r/r)_{\text{ext}} = +P\mu/E$ outside the plenum chamber (uniaxial compression) [11]. For high-quality steels, Young's modulus is $E = 2.1 \cdot 10^6$ kgf/cm$^2$ and Poisson's ratio $\mu$ is within 0.25–0.30. Therefore, according to the above-given formulas, for pressures of about 20,000 atm, the relative increase in the diameter of the inner surface of the cylinder is about 1.2% even for infinitely thick walls.

The diameter of the part of the rod located in the plenum chamber under conditions of triaxial compression, vice versa, decreases by approximately 0.4%. Outside the plenum chamber, the rod loaded only by uniaxial longitudinal compression becomes "thicker" by 0.3%. Thus, the total change in the rod diameter is about 0.7%. The consequence of disagreement in deformations is the appearance of a gap between the cylinder and the rod, whereas absolute leakproofness is necessary, otherwise the leak of a hot gas will lead to failure of the seal material or, in the case of working with air, to its burnout. The difficulties in the creation of a movable seal capable of working under these conditions become clear. The leakproofness can be ensured only under the condition that the contact pressures between the cylinder and the rod in the closing part of the seal are always greater by a certain value than the gas pressure, which reaches 20,000 atm.

Under these conditions, the use of traditional seals with glands, sleeves, and sealing and antiextrusion rings could not be successful. A steel–steel contact is needed to ensure strength. In this case, the friction coefficient is 0.2–0.3. The estimates show that, at these contact pressures, the operation of friction forces upon mutual motion of the rod and the cylinder for an expected velocity of about 3 m/sec will heat the contact surfaces up to a temperature much greater than the melting point of all known metals and alloys.

Thus, it was necessary to solve three problems: avoid an increase in the internal diameter of the cylinder, compensate for rod deformation ensuring contact pressures greater than the gas pressure, and sharply decrease friction. These problems were solved by using a cylinder with a variable external radial support and using a thin indium coating as an antifriction layer.

A segmented high-pressure cylindrical vessel with variable external support (Platen's scheme) differs from the known constructions [12, 13] in having a high strength and the best dynamic characteristics. The frequency of eigenoscillations is maximum with a minimum effect of friction forces.

The layout of the high-pressure stage with the use of such a vessel applied in the A-1 facility [14] is shown in Fig. 4. The pressure from the cavity of the large cylinder of the multiplier through the connecting channel is transferred to the cavity of external support. Movable sealing of the rod upon gas compression is ensured by squeezing the internal cylinder by the segmented cylinder under the action of the external pressure of the liquid in the cavity of external support. The stress on the internal surface of the cylinder is usually removed by variable external support, which ensures a high strength of the vessel. However, if the pressure on the external surface of the internal cylinder $P_b$ satisfies the relation [15]

$$P_b \geqslant P_a\left\{1 - \frac{\mu}{2}\left[1 - \left(\frac{a}{b}\right)^2\right]\right\}$$

($a$ and $b$ are the internal and external radii of the cylinder and $P_a$ is the internal pressure), the appearance of a gap between the rod and the cylinder is prevented, i.e., the leakproofness is ensured. This is sufficient for operation of a stationary seal.

In the case of a moving piston, the friction should be reduced even more. This is achieved by introducing an antifriction material (indium) into the contact zone between the rod and the cylinder. If a decrease in

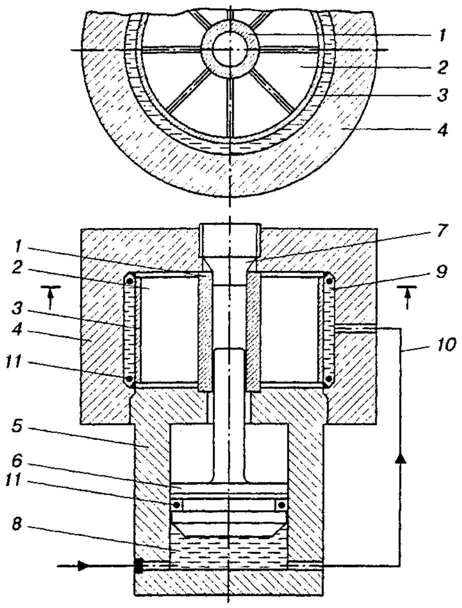


Fig. 4. Layout of a multiplier with hydromechanical support: *top figure*: cross section marked by arrows in the bottom figure; *bottom figure* (longitudinal section): 1) internal cylinder; 2) segmented cylinder; 3) thin shell; 4) cage; 5) large cylinder; 6) multiplier piston; 7) nozzle plug; 8) cavity of the large cylinder of the multiplier; 9) cavity of external compression; 10) connecting channel; 11) seals.

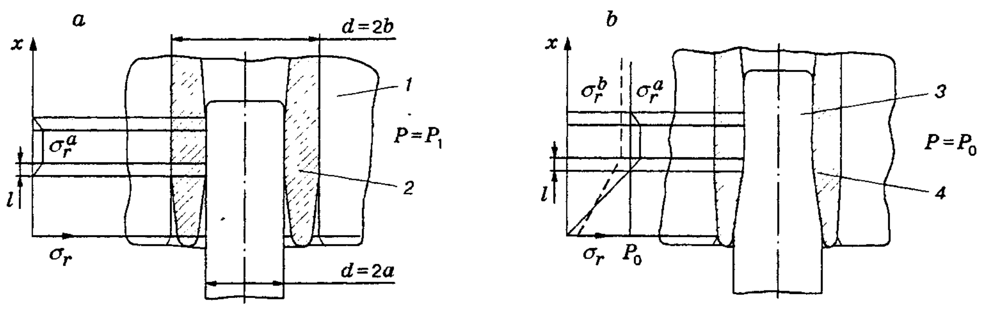


Fig. 5. Contact stresses and deformations of elements in the case of sealing with external support for the initial (a) and final (b) stages of compression; 1) support element (segment); 2) cylinder; 3) rod; 4) layer of a soft metal (indium).

the radial stress (contact pressure) $\sigma_r^a$ close to linear is ensured on the contact surface in the axial direction outward from the plenum chamber, it can be expected that the indium layer remaining after loading, whose shear strength is $\tau_{\rm sh}$, will have a roughly constant thickness $\delta \geqslant \tau_{\rm sh}|dx/d\sigma_r^a|$.

A solution of a similar problem in the 2D formulation for an "ideally plastic long strip" was found by Prandtl [16] and refined by V. V. Sokolovskii [17]. The necessary dependence of the normal stress on the coordinate is ensured by special contouring of the input part of the cylinder, namely, by making bevels [14], which constrict the cylinder wall toward the input butt-end face. This constriction is chosen to be equal to the sum of the differences in the strains of the rod, cylinder wall, and supporting segment upon loading in zones where the stresses are $\sigma_r^a = 0$ and $\sigma_r^a \geqslant P_a$. The difference in rod strain is compensated by the internal bevel, and that between the segment and the cylinder by the external bevel. Figure 5 shows the distributions of contact stresses and strains (for illustration purposes, the deformations and bevels are greatly enlarged).

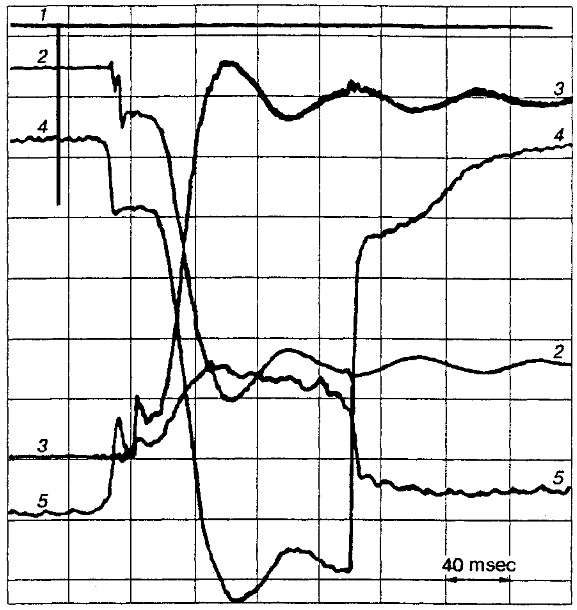


Fig. 6. Pressure oscillograms in the units of the A-1 facility in the dynamic regime (time marker 40 msec): 1) signal of actuation of the starting valve; 2) pressure in the large cylinder of the multiplier; 3) pressure in the cavity of external support (reverse polarity signal); 4) pressure in the plenum chamber (maximum value 8900 atm); 5) gas temperature.

If an indium layer of nonzero thickness is retained, the friction force tangent to the unit surface $\sigma_f$ cannot exceed the ultimate shear strength $\tau_{sh}$ whose value for indium is about 30 kgf/cm$^2$. This means that the seal works under conditions of a constant friction force, in a regime where the friction coefficient decreases with increasing contact pressure. The estimates show that the increase in temperature due to the work of friction forces is about 40°C under these conditions.

Figure 5 shows the elements of the segmented cylindrical vessel for the minimum and maximum values of pressure; the angles and relative deformations are greatly enlarged for illustration purposes. Figure 5a shows the initial stage: the moment of locking the test gas. The stress $\sigma_r^a$ is generated by the preliminary tension $\sigma_0$, which ensures a contact pressure slightly greater than the gas pressure $P_1$ after gas compression in the first stage: $\sigma_{r\ \min}^a = \sigma_0 + P_1 = \sigma_0 + 2000$ atm. Figure 5b shows the final stage. The multiplication factor of the segmented support is chosen such that the maximum value of the contact load due to the preliminary tension $\sigma_0$ and external loading is supported at a level higher than the gas pressure in the plenum chamber $P_0$: $\sigma_{r\ \max}^a = \sigma_0 + P_1 + P_0$. With increase in pressure, the contact zone is expanded and covers the entire input part of the cylinder. A smooth decrease in stresses along the rod increases the workability of construction elements, since large gradients of normal stresses cause large shear stresses, which destroy the cylinder and the rod. The rod turns out to be the weakest construction element in the layout used; the part of the rod outside the high-pressure cylinder is in the state of uniaxial compression. The maximum pressure reached in static experiments without residual deformation was 25,000 atm.

**Test Results and Utilization.** The above-described principles were used in the operating model facility A-1, which reproduces all the design parameters.

Figure 6 shows the results of pressure measurement in the units of the facility in the dynamic regime. The experiment was performed for a 0.34-mm diameter of the nozzle throat. The decaying oscillatory process visible in the oscillograms is associated with the phenomena that occur in the pipeline supplying a high-pressure gas from the receiver to the low-pressure side of the multiplier and is not related to the processes in the facility itself.

The simultaneous variation in pressure in all units of the facility is worth noting. The first peak in all oscillograms is associated with the operation of the stage of preliminary compression, which increases the pressure in the plenum chamber to approximately 1500 atm. In the oscillogram of the external-support pressure, this peak is registered with a certain delay: some time is needed for the displacement of the multiplier piston and overflow of the liquid related to its compressibility and deformation of the segments of external support. The run time in this experiment was about 90 msec. At the end of it, when the rod comes into contact with the nozzle plug, a sharp decrease on the pressure oscillogram in the plenum chamber is observed. At this moment, the pressure probes in the cavity of external support and in the large cylinder of the multiplier simultaneously register a small peak on pressure oscillograms in the cavity of external support and the large cylinder of the multiplier. Since no noticeable overflow of the liquid occurs in the system of external support in this situation, desynchronization of the signals is not observed.

An analysis of oscillograms shows that the friction force of the multiplier piston at the end of motion does not exceed 6% of the acting force. The system ensures reliable sealing in the case of piston motion with a velocity up to 3 m/sec. No leaks of the gas through the side surface of the plug and the cylinder were observed. Control checks after several years of use revealed a satisfactory state of the elements of the multiplier operating under conditions of high mechanical and thermal stresses. The results of static and dynamic tests confirmed the effectiveness of the sealing method proposed and the correctness of the estimates made.

**Conclusions.** Obtaining pressures up to 10,000 atm in the operating facility despite its small dimensions (plenum-chamber volume of 40 $cm^3$) made it record-setting in terms of the hypersonic flux density and reproducible Reynolds numbers. The presence of a multiplier forcing out the gas under different pressures in the plenum chamber and throat diameters allows one to obtain run times from 20 to 250 msec in the A-1 facility [18].

The principles applied and the absence of elements operating beyond the limit of elasticity allows one to pass easily to large-scale facilities and ensure an almost unlimited number of operation cycles. Since the moment of its creation in 1975 [19], the A-1 facility has been used for laboratory studies of the properties of dense hypersonic flows and development of experimental methods capable of working in this range of parameters. Using the A-1 facility, probes for pressure measurement under conditions of the plenum chamber of hypersonic wind tunnels for a pressure range from 3000 to 10,000 atm and higher have been developed [20]. The method of direct measurement of the hypersonic flow velocity by means of electric discharge tracking [21] has been improved and adapted, which allowed experimental verification of the validity of the equations of state of the gas, proposed in [1, 5], and methods of calculation of heat losses in the channel of a high-pressure aerodynamic facility [22]. The search for materials for nozzle throats and testing of their strength has been performed and has continued [23].

The experience obtained in the creation of the A-1 facility was used in designing [24] and creating [25] a semi-industrial wind tunnel AT-303 at the Institute of Theoretical and Applied Mechanics of the Siberian Division of the Russian Academy of Sciences.

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