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TRANSLATION

DESIGN OF AIRCRAFT GAS-TURBINE ENGINES

By

A. V. Shtoda, S. P. Aleshchenko, et al.

TT64-11980
625 ja
hc ~ 9.25
mf ~ 2.75

FOREIGN TECHNOLOGY DIVISION

FOREIGN TECHNOLOGY DIVISION

AIR FORCE SYSTEMS COMMAND

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FTD-TT- 63-405/1+2

UNEDITED ROUGH DRAFT TRANSLATION

DESIGN OF AIRCRAFT GAS-TURBINE ENGINES

BY: A. V. Shtoda, S. P. Aleshchenko, et al.

English Pages: 571

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KONSTRUKTSIYA AVIATSIONNYKH GAZOTURBINNYKH DVIGATELEY

Voyennoye Izdatel'stvo
Ministerstva Oborony Soyuza SSR

Moskva-1961

pages 3 - 411

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TABLE OF CONTENTS

Design of Aviation Gas-Turbine Engines (Training Manual).

Edited by Docent Candidate of Technical Sciences A.V. Shtoda

This book presents the fundamental problems in the design of gas-turbine engines.

The material of the book is presented in a form suitable for persons having a secondary school education who are familiar with the basic theory of motors.

The book is intended for engineering-technical and flying personnel of the VVS, GVF and DOSAAF. It may be found useful for auditors, registrants, and students of aviation-technical educational institutions, and also for persons interested in aviation engines.

FOREWORD

The basic engine used in aviation at the present time is the reaction-thrust gas-turbine engine.

Modern gas-turbine aviation engines, designed on the basis of advanced science and technology, are very complex machines; their application requires a fundamental understanding of the processes that take place in motors, their characteristics, the conditions of their operation, and the stresses and designs of component parts and assemblies, as well as the over-all systems which are essential to the operation of the engines. Furthermore the very rapid development of gas-turbine aviation engines causes relatively frequent displacement of the various engine models, by modifications and redesigning.

In order to facilitate understanding of the design of specific engines, suitable textbooks for engineering-technical and flight personnel of the VVS are required, in which the general problems of design of aviation engines are presented.

This book is offered as such a textbook.

The book presents not only the design versions of gas-turbine engines, and the conditions of their operation and loading due to static and dynamic forces, but also a description of the systems necessary for the operation of the engines.

The book is made up of an introduction and nine parts. The introduction contains a review of the types of aviation engines and the regions of their application, including short expositions of the schematic and design layouts of gas-turbine engines; certain fundamental concepts and definitions necessary for subsequent study of the design of the engines are given; values of the fundamental parameters of the gas-turbine engines are presented, as is a basic review of their design and development.

The first, second, and third parts describe in succession the construction of the air-flow parts of the engines, i.e., compressors, turbines, primary and secondary (afterburner) combustion chambers, and the exhaust arrangements.

Parts four and five consider the power systems of gas-turbine engines, the transmission elements, bearings, and lubrication systems; basic data on the critical speeds of rotation of rotors are also presented.

The sixth part is devoted to specific units of the turboprop engine – the hub of the propeller and the reduction gear.

The seventh part contains material on the construction of fuel pumps and atomizers, and their operation.

Basic data on systems for automatic control of gas-turbine engines is presented in the eighth part.

Finally, the ninth part is devoted to starting systems for gas-turbine engines. Basic information on starters, ignition equipment, and automatic fuel feed during starting is also presented.

From the above, it is clear that the book covers the basic problems in the design of gas-turbine engines. All the material is presented in a form suitable for persons having a secondary school education and familiarity with the basic theory of engines.

In writing the book, the authors have used the nonclassified material listed at the end. Structural diagrams of engines, examples of the construction of various parts, sketches of power systems of the motors, diagrams of lubrication systems as well as systems for fuel feed and automatic regulation, gasdynamic and design parameters, and information on the strength considerations for engines are given on the basis of material derived from foreign engines as well as the [Soviet] engines RD-10, RD-20, RD-500, RD-45, VK-1, AI-20, and certain parts of the AM-3 and AM-5. Since in most cases the original material on design was methodically reworked by the authors, reference to specific engines is largely omitted in the text as having no particular significance in the understanding of the general questions of design of gas-turbine engines.

The book was written by the following Docent Candidates of Technical Sciences: Introduction, A.V. Shtoda; Part 1, A.G. Shiukov and V.S. Krasavtsev; Part 3, S.P. Aleshchenko; Parts 4 and 5, F.N. Morozov; Part 6, V.S. Krasavtsev; Part 7, A.G. Shiukov; Part 8, A.V. Shtoda, V.A. Sekistov and A.G. Shiukov; Part 9, A.Ya. Ivanov.

The authors wish to thank I.T. Denisov for a number of valuable suggestions and preferences that he supplied in his review of the manuscript.

INTRODUCTION

1. TYPES OF AVIATION ENGINES AND THE REGIONS OF THEIR APPLICATION

The engine is the basic part of the powerplant of a flying machine and has played a primary role in the development of aviation. The first flights of heavier-than-air flying craft were accomplished only after engines of high power and light weight had been developed. All of the most important advances and achievements in aviation have been related in one way or another to the appearance of new types of aircraft powerplants, or to fundamental improvements of existing types. An especially dramatic example of this is the abrupt qualitative turning point produced in aviation development by the appearance of the gas-turbine engine.

Modern aviation gas-turbine engines, being very complicated machines created on the basis of the most recent achievements of science and technology, require not only the existence of a highly developed aviation industry, but also the continuous expansion and development of many other adjacent branches of industry. The design, construction, and perfection of a modern aviation engine require from three to six years, which exceeds the period required for the design, construction, and perfection of the airframe into which the engine is to be installed.

The gas-turbine engine is one of a number of types of reaction motors.

All reaction motors may be divided into two classes:

1) rocket reaction motors;

2) air-breathing reaction motors.

Rocket reaction motors in turn are classified into two categories:

1) liquid-fuel rocket motors (ZhRD),

2) solid-fuel rocket motors (RDTT).

Air-breathing reaction motors in turn are classified into two types:

1) compressorless or direct-flow motors (PVRD) (ramjets);

2) gas-turbine engines (GTD).

At the present time, gas-turbine engines are the primary type used in aviation. Reciprocating engines, the possibilities of their further development having been exhausted, have been displaced by gas-turbine engines, which are characterized by a substantial reduction in specific weight and dimensions in comparison with the reciprocating motors, together with acceptable economy, particularly at high flight speeds.

In the relatively short period of the development of gas-turbine engines they have attained a very high level of refinement; yet they have a great deal of potential for further improvement, and for application to flight for a long time to come. In comparison with aviation engines of other types, the gas-turbine engines have their most advantageous region of application to flight in the speed range below 2.5 to 3 times the speed of sound, and at altitudes up to 25-30 km.

Modern aviation gas-turbine engines are divided into two classes – the turbojet engines (TRD) and turboprop engines (TVD).

The TRD, using for thrust production atmospheric air which passes through the engine itself, is favorably differentiated from the TVD by the fact that it does not have a propeller. Because of this fact, the TRD has a lower weight, greater operational reliability, and less operational complexity.

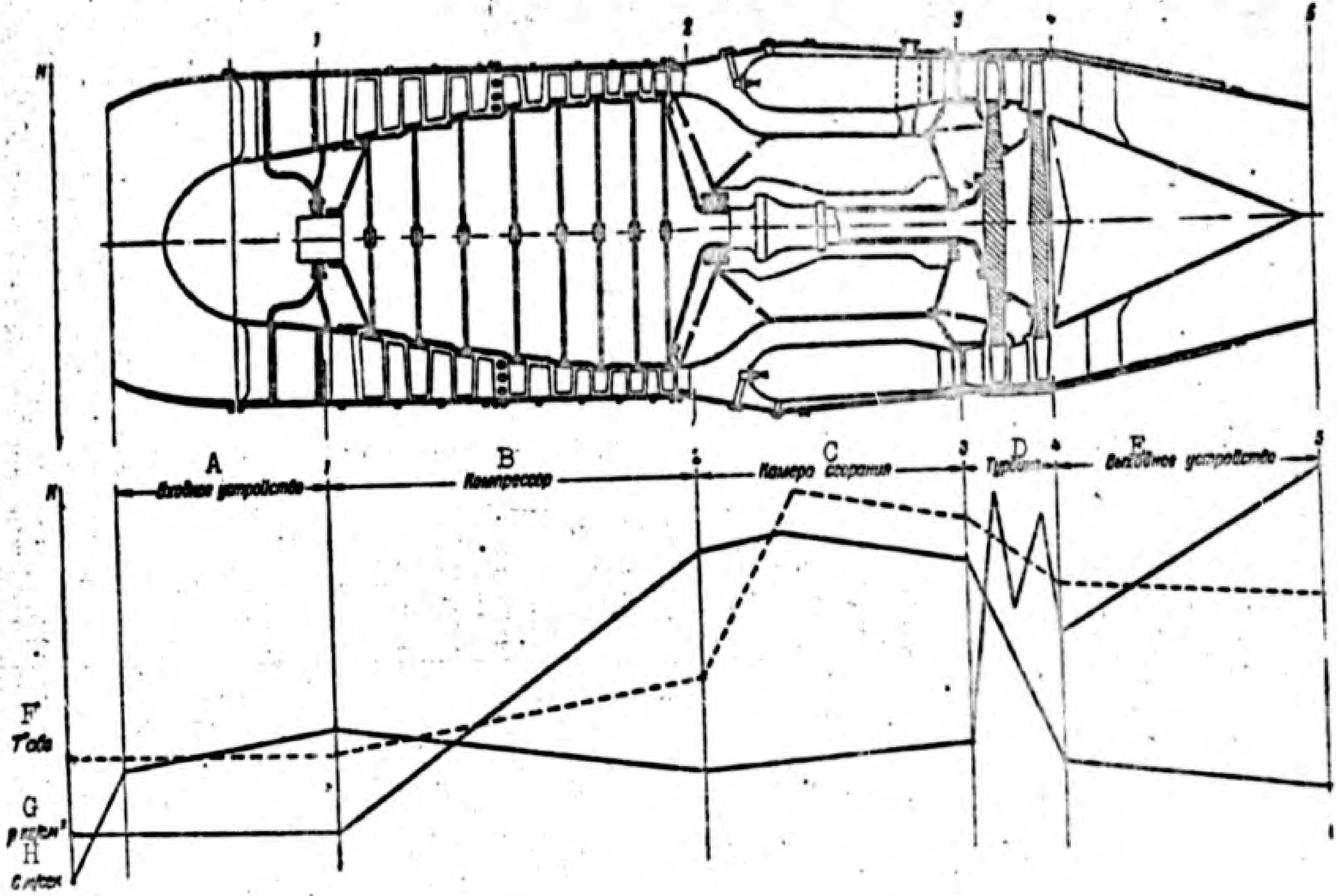

Fig. 1. Diagram of the turbojet and the nature of the variation of temperature, speed, and pressure of the air and gases along the flow passages of the engine. A) Entrance section; B) compressor; C) combustion chamber; D) turbine; E) exit section; F) T, °abs; G) p, kg/cm²; H) C, m/sec.

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TRD's are installed in fighter and bomber aircraft and transports, and in airborne guided missiles flying at high subsonic and supersonic speeds [27].

In flight at subsonic speeds, however, the TVD is more economical, by reason of its use of a propeller, than is the TRD – that is to say, it uses less fuel. This fact makes their use more expedient in subsonic transports and VTOL aircraft.

Another type of air-breathing reaction engine is the ramjet (PVRD), which compresses the air only through the action of the pressure head; this type is most conveniently used at flight speeds exceeding the velocity of sound by a factor of more than 2.5-3, and at altitudes up to 20-35 km. Since the PVRD's are incapable of independent takeoff, because they generate no thrust at low speeds, we can either use them in combination with other propulsors or provide aircraft powered by the PVRD with acceleration by means of the powerplants of special mother aircraft. PVRD's are used on winged missiles and flying targets [27], where their simplicity and low cost of construction is of overriding importance.

Rocket reaction motors – liquid-fuel rocket (ZhRD) and solid-fuel rocket engines (RDTT) are widely used in rocket engineering; in aviation they are used only as takeoff boosters and powerplant (sustainer) engines for use with high-velocity aircraft at very high altitudes at which air-breathing reaction engines - the gas-turbines or ramjets - are no longer usable [27].

2. SCHEMATIC AND DESIGN ARRANGEMENTS OF GAS-TURBINE ENGINES

The thermodynamic processes taking place in a gas-turbine engine are the compression of the air, its heating, and the subsequent expansion of the gases, as a result of which energy is generated to be expended in propelling the aircraft.

The conventional turbojet engine with an axial compressor (Fig. 1) consists of the following basic parts: entrance section, compressor, combustion chamber, turbine, and exit section.

The entrance section is designed for introduction of the air into the compressor with the least possible loss of pressure and with uniform velocity distribution. The pressure losses at the intake to the compressor are differentiated into hydraulic losses (caused by friction and generation of vortices), and wave losses. At low air speeds, the former are more important, and at high speeds, the latter predominate.

For the purpose of decreasing the wave losses of TRD's designed for speeds greater than Mach 1.5 to 2, a supersonic entrance section is used (controllable, or fixed). TRD's designed for aircraft with speeds less than sonic have a subsonic entrance section such as is shown in Fig. 1.

In static operation of an engine or at low air speed, the air at the entrance section is accelerated, and its temperature and pressure are reduced. Thus, for example, in static operation of an engine, the pressure p_1 and the temperature T_1 of the air at the inlet to the compressor will be less than the pressure p_H and the temperature T_H of the air in the ambient atmosphere at a reasonable distance from the engine. The difference will be greater the greater the air speed into the compressor. If this air is slowed down, its temperature T_1^* will be found to be equal to the temperature T_H, since at the entrance section heat is neither added nor removed from the air. The pressure of the air after it has been slowed down at the compressor inlet, denoted p_1^*, would be somewhat less than p_H by reason of hydraulic losses.

In flight at speeds C_H greater than the velocity C_1 of the air into the compressor, the air in the entrance section is slowed down, and its pressure p_1 and temperature T_1 increase and become greater than the pressure and temperature of the ambient air. If the air should now be slowed down arbitrarily at the compressor entrance, then the pressure p_1^* and the temperature T_1^* would become even greater than the values of these parameters in the surrounding atmosphere.

The ratio T_1^*/T_H depends only on the Mach number $M_H = C_H/a_H$, that is, on the ratio of the flight speed C_H to the velocity of propagation of sound a_H in the atmospheric air. The greater the Mach number M_H, the more will T_1^* exceed T_H. On the other hand, the ratio p_1^*/p_H, which designates the effective pressure ratio of the air in the entrance section with respect to the decelerated parameters, depends not only on the M_H number, but also on the loss of pressure in the entrance section.

The efficiency of the entrance section can be evaluated by use of the so-called pressure recovery coefficient σ_{vkh}, which is equal to the ratio of the pressure p_1^* of the decelerated air at the compressor intake to the pressure p_H^* of decelerated atmospheric air with no losses whatsoever:

$$\sigma_{vkh} = p_1^*/p_H^*. \qquad (1)$$

The higher the flight speed, the smaller the pressure recovery coefficient, primarily by reason of the increase of wave losses. If at $M_H = 1.5$, σ_{vkh} is equal to about 0.92-0.96, then at $M_H = 3$, $\sigma_{vkh} = 0.4$ to 0.7.

The compressor functions to compress the air. This compression is necessary for better efficiency in the transformation to useful work of the heat supplied to the air downstream of the compressor. Furthermore, the compression of the air decreases its volume, which makes possible a reduction in the size of the combustion chamber and turbine.

One of the most important parameters of the compressor is the effective pressure ratio, which is denoted by π_k. It is equal to the ratio of the pressure of p_2 of the air at the outlet from the compressor to the pressure p_1 of the air at the entrance:

$$\pi_k = p_2/p_1. \qquad (2)$$

The pressures p_2 and p_1 depend on the speeds of the air at the exit and entrance of the compressor, C_2 and C_1, respectively. In this connection the compressor is generally characterized by the pressure ratio of the arbitrarily decelerated air at the exit and entrance:

$$\pi_k^* = p_2^*/p_1^*. \qquad (3)$$

The speed C_2 is ordinarily less than C_1. The air temperature at T_2^* at the exit from the compressor is greater than the temperature T_1^* at the entrance, and the more so the greater the pressure ratio.

Ordinarily it is the intention that the compressor should operate in such a fashion that the energy supplied to it will be utilized as completely as possible for compression of the air, without large losses. The degree of compressor perfection is evaluated by use of the shaft-power efficiency factor η_{ek}^*, equal to the ratio of the work expended in compressing the air without heat transfer between the external medium or hydraulic losses (adiabatic compression), to the total (effective) mechanical work supplied to the compressor.

The fuel fed to the combustion chamber burns there and this is accompanied by a substantial increase in the temperature of the air at nearly constant pressure. The average speed of the air in the combustion chamber is not large. The pressure along the combustion chamber falls by reason of hydraulic losses and the increase in speed due to the volume increase on heating. The speed of the gases at the outlet of the combustion chamber is denoted by C_3. The pressure and temperature of the gases at the outlet from the chamber, after slowing down to stagnation, are designated by p_3^* and T_3^*. From the combustion chamber, the gases enter the gas turbine.

The turbine in a TRD has the function of driving the compressor and auxiliary power drives. The turbine extracts a part of the energy of the gases, so that the pressure p_4^* and temperature T_4^* are lower than at the entrance. The velocity C_4 of the gases at the turbine outlet is greater than C_3 but not very much greater, so as to avoid large hydraulic losses in the turbine. The ratio of pressure of the gases at the turbine entrance to that at the exit is designated the expansion ratio of the gases. If we specify the ratio of pressures in the gas flow, the expansion ratio of the gases in the turbine is denoted by π_t.

$$\pi_t = p_3/p_4. \qquad (4)$$

If, however, we take the ratio of pressures of the decelerated gas, the expansion ratio of the gases with reference to deceleration parameters is denoted by π_t^*:

$$\pi_t^* = p_3^*/p_4^*. \qquad (5)$$

The work which might be produced by the gas in the turbine upon its expansion from the pressure p_3^* to p_4 without any losses is called the adiabatic-expansion work, or the design heat drop [available heat difference] of the turbine. The design heat drop is greater, the higher the temperature T_3^*. In practice there are losses due to friction (hydraulic losses) and losses due to the exit speed C_4 (not equal to zero), so that the actual (effective) work at the turbine shaft is less than the design heat drop. The ratio of the effective work at the turbine shaft to the design heat drop is called the shaft-power efficiency of the turbine and denoted η_{et}. By definition it follows that η_{et} depends on the magnitude of the outlet speed C_4. At present it is considered that the heat difference is determined by the deceleration parameters of the flow, not only at the entrance to the turbine (p_3^* and T_3^*), but also at the exit (p_4^* and T_4^*). Since $p_4^* > p_4$ the design heat drop according to the deceleration parameters is less than the design heat drop according to the nondeceleration parameters at the exit, and therefore the shaft efficiency of the turbine based on the deceleration parameters η_{et}^* is somewhat larger than η_{et}.

From the turbine, the gas comes to the exit section, consisting of an exhaust tube and nozzle.

The converging nozzle should be differentiated from the Laval nozzle. In the converging nozzle (see Fig. 1) the exit cross section has a minimum area F_5, which is always less than the cross section at the turbine outlet. Because of this fact, the pressure p_4 of the gases is increased, and the speed C_4 is reduced; consequently, the losses in the turbine and exhaust tube are reduced and it is possible to regulate the turbine power by changing the cross sectional area F_5 of the nozzle.

If the pressure p_4^* is greater than atmospheric pressure p_H by no more than a factor of 1.85, the smallest exit cross section of the nozzle will be characterized by a pressure $p_5 = p_H$, and the greatest speed of the gas C_5 will be less than sonic speed in the gas at temperature T_5. The gas will then be completely expanded in the exit section.

If, however, the pressure p_4^* exceeds the atmospheric pressure p_H by more than a factor of 1.85, the exit cross section becomes critical, and the velocity of the gas becomes equal in it to the velocity of sound, the pressure taking a value that is greater than the atmospheric pressure. In this case, the convergent nozzle does not effect complete expansion of the gas, and a loss in thrust is the result. A Laval nozzle is used to avoid large thrust losses in aircraft with high supersonic speeds. In such nozzles, in addition to the convergent part, there is a diverging part (Fig. 2). The exit cross section of the nozzle F_5 is larger than the critical cross section F_{kr}. In the diverging part of the nozzle, the excess pressure acts to increase the thrust. The greater the flight speed, the more the cross-sectional area F_5 should exceed F_{kr}. In this connection it is desirable that the nozzle have an adjustable diverging part [27].

In recent years extensive use has been made of engines with boosted (augmented) thrust obtained by burning fuel behind the turbine in a special afterburner chamber (see Fig. 2). These engines are designated TRDF for short. Since the pressure is less in the afterburner than in the regular combustion chamber, the use of the fuel for production of thrust is less economical in the former than in the latter. In aircraft with large supersonic capability, however, this difference is decreased. The advantage of the TRDF consists in the fact that it is possible to increase thrust without significant increase of weight and size of the engine. For the purpose of avoiding large expenditures of fuel, it is customary to permit use of the afterburner only for short periods of time.

Along with TRD's of the design already considered, some use has been made of bypass [turbofan] engines, called DTRD, for short. These engines ordinarily have several axial compressor stages, which are used (Fig. 3) for feeding air not only through the main duct, but also through an auxiliary [bypass] duct. These stages of the compressor have the function of a fan that increases the mass of air handled by the engine, so that such engines are sometimes called turbofan engines. Because of the increase in the mass of air handled [i.e., because of an increase in the air flow rate], the design thrust of the engine may be obtained with smaller average speeds at the outlet of the engine, which

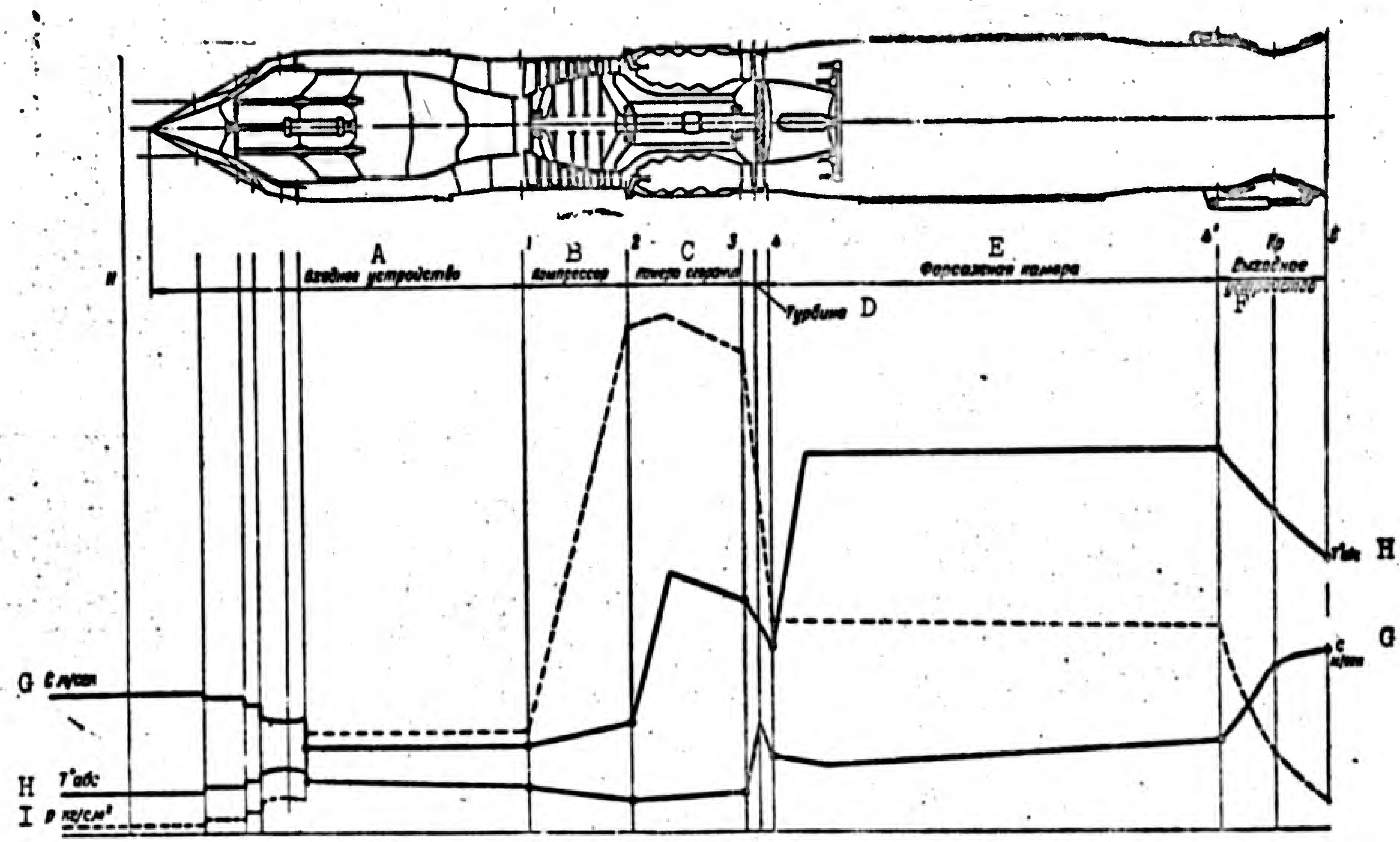

Fig. 2. Sketch of TRDF and the nature of the variation of temperature, speed, and pressure of the air and gases along the gas passages of the engine in flight [27]. A) Entrance section; B) compressor; C) combustion chamber; D) turbine; E) afterburner; F) critical exit section; G) C, meters/sec; H) T, °abs; I) p, kg/cm^2.

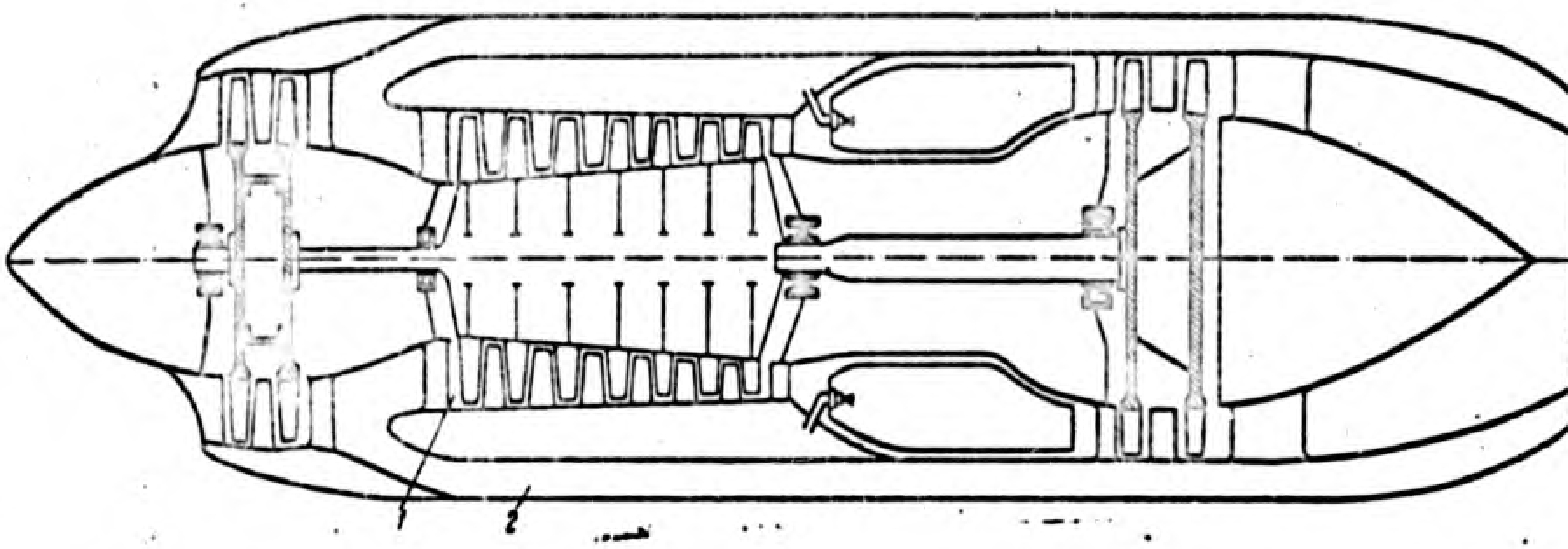

Fig. 3. Diagram of turbofan engine. 1) Inner (basic) duct; 2) outer duct.

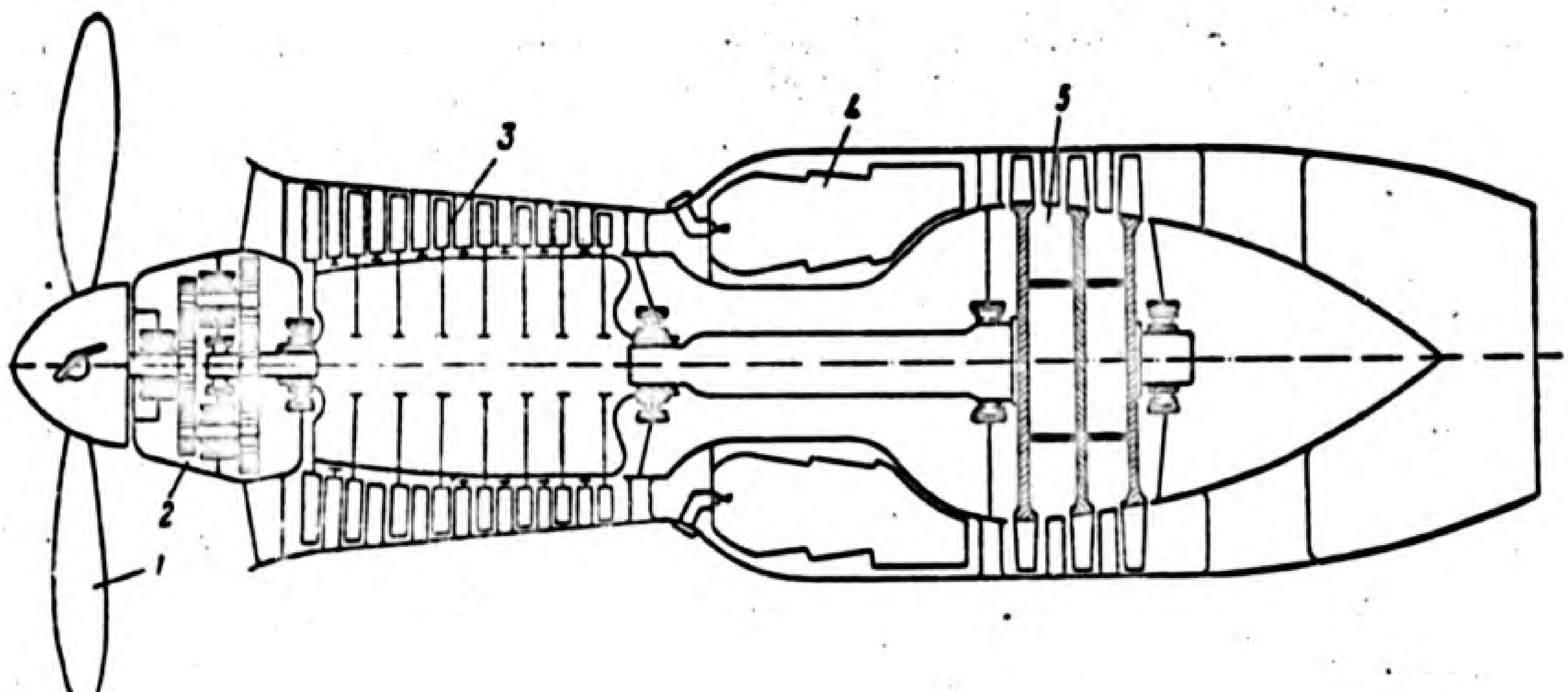

Fig. 4. Diagram of the TVD with common turbine. 1) Propeller; 2) reducing gears; 3) compressor; 4) combustion chamber; 5) common turbine.

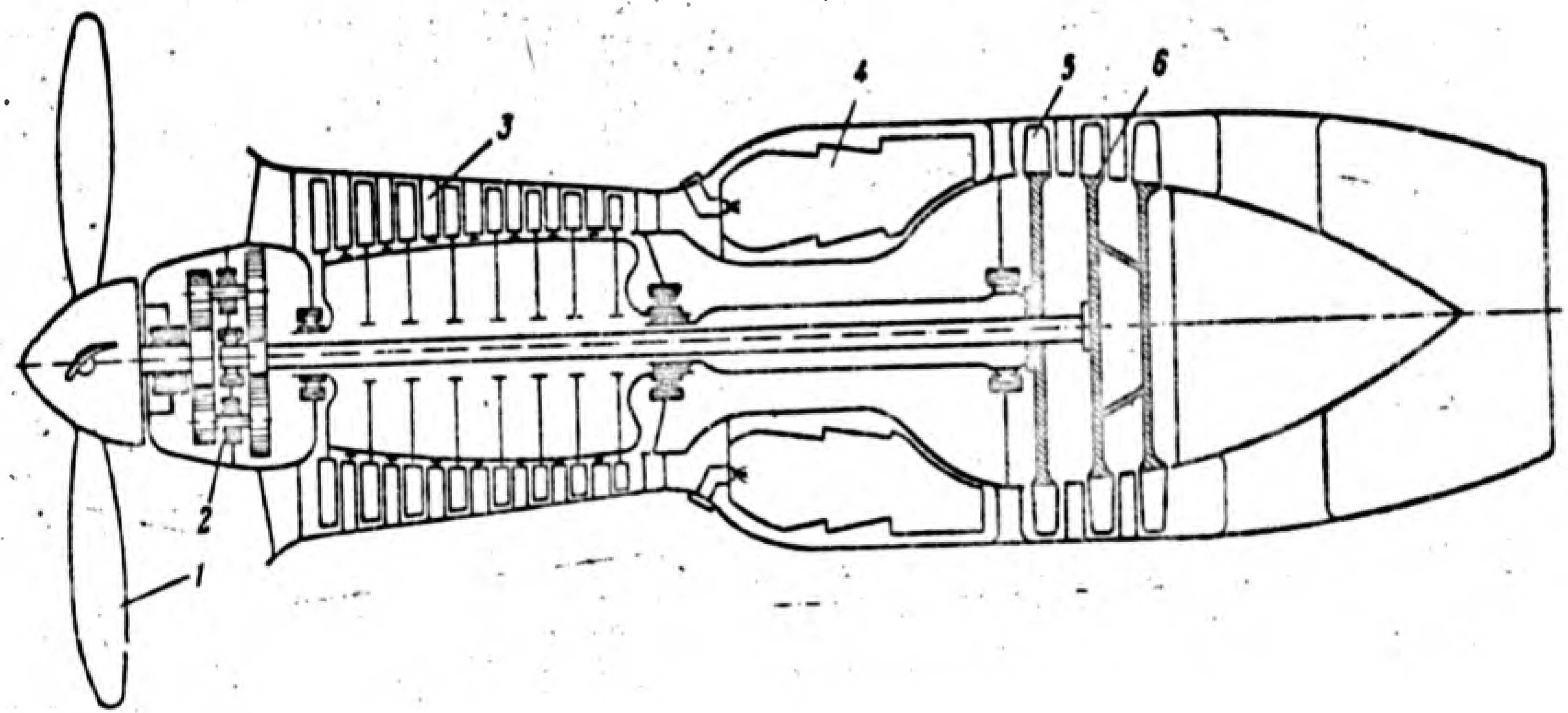

Fig. 5. Diagram of TVD with separate turbines. 1) Propeller; 2) reduction gearing; 3) compressor; 4) combustion chamber; 5) compressor turbine; 6) propeller turbine.

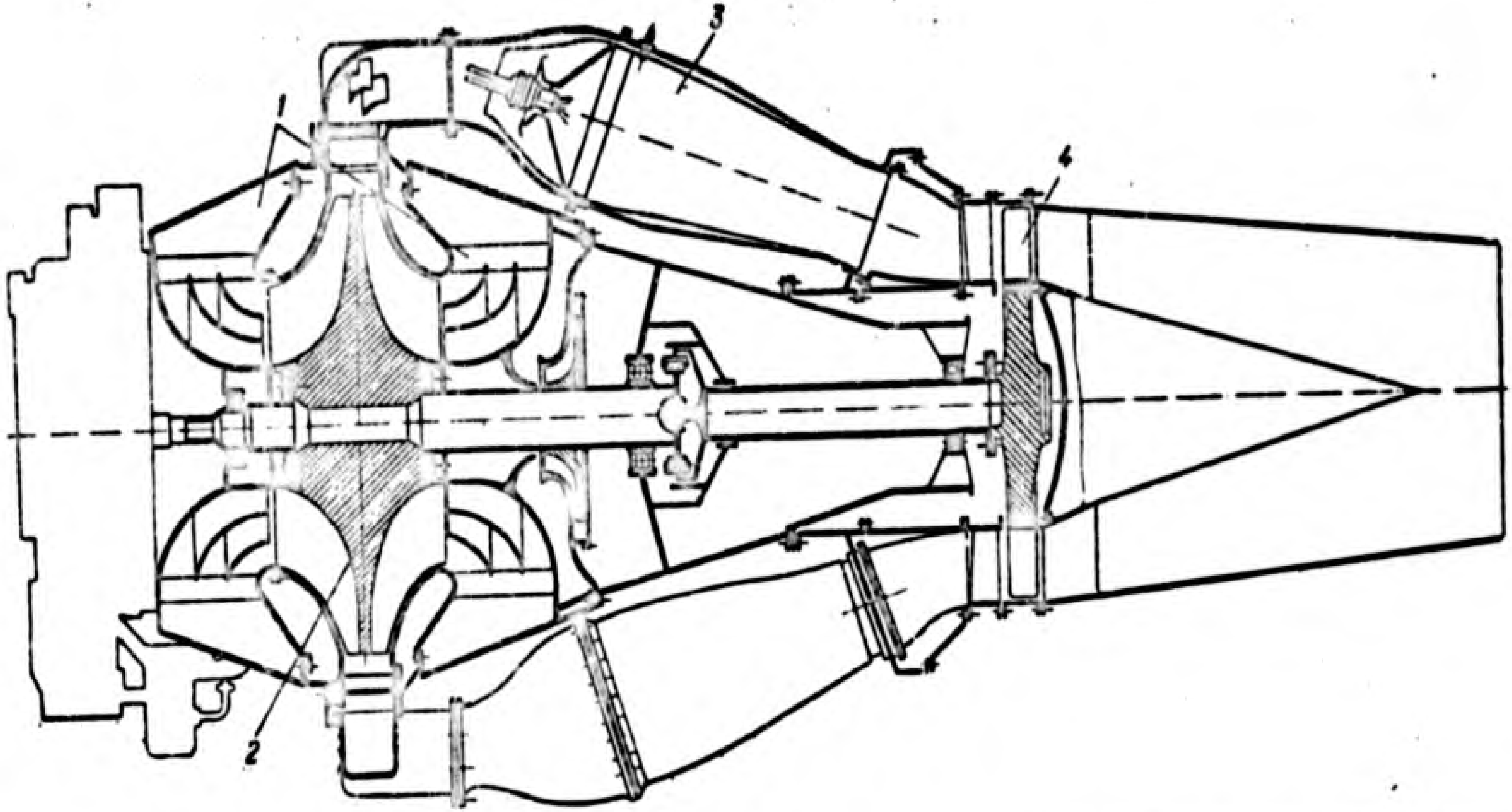

Fig. 6. Diagram of TRD with a centrifugal compressor. 1) Entrance section; 2) centrifugal compressor; 3) combustion chamber; 4) turbine.

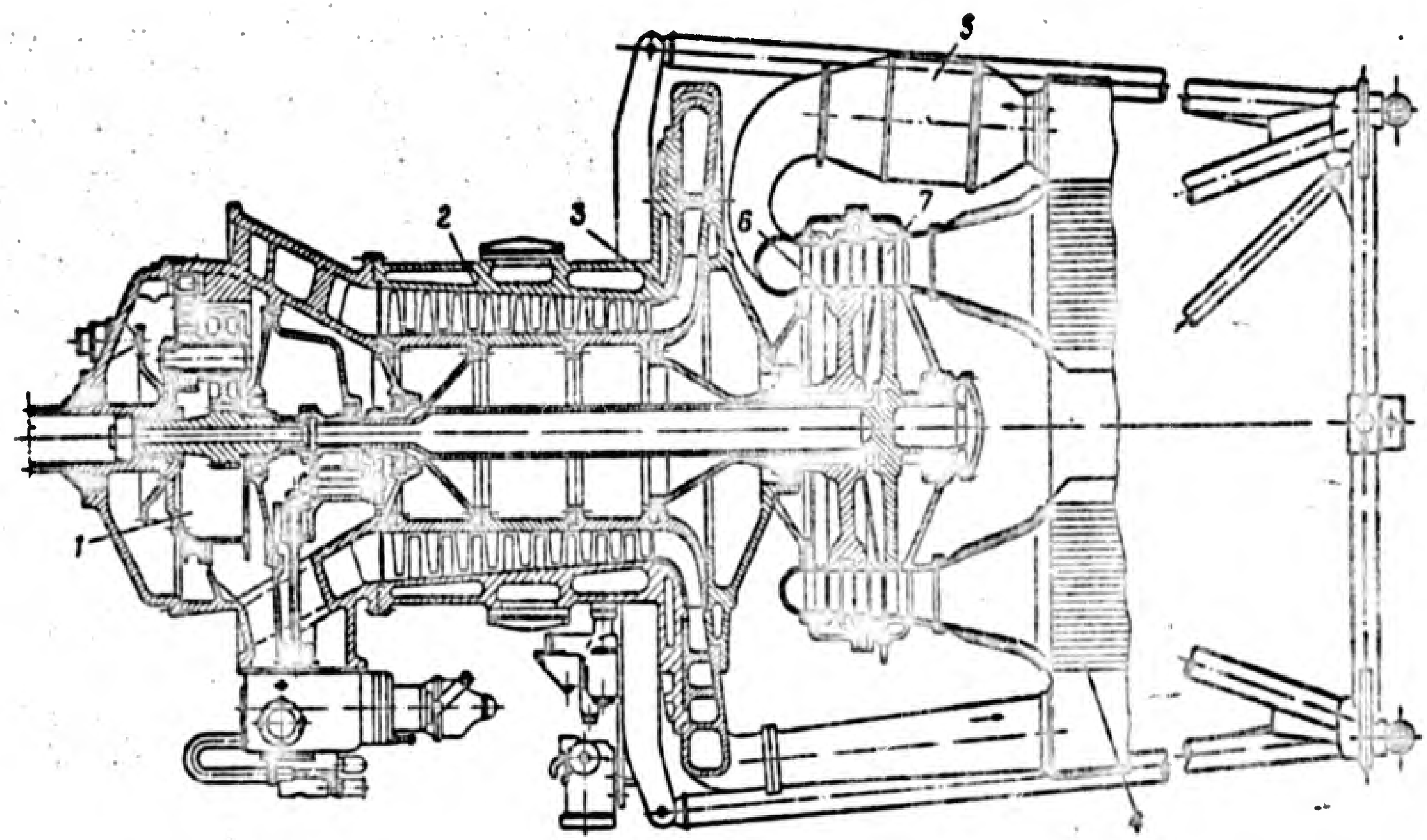

Fig. 7. Diagram of TVD with axial-centrifugal compressor. 1) Reduction gearing; 2) axial compressor; 3) centrifugal compressor; 4) heat exchanger; 5) combustion chamber; 6) compressor turbine; 7) propeller turbine.

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reduces the energy lost in the exhaust gases and increases the economy of the motor, particularly at low flight speeds. Since only part of the air goes through the entire compressor, combustion chamber and turbine, those parts can be reduced in size and weight. If necessary, the thrust can be increased by use of a burner in the outer duct.

In a turboprop engine, the thrust is generated not only by the reaction of gases flowing through the nozzle, but also by the reaction of the mass of air accelerated by the propeller. At subsonic flight speeds, the utilization of the energy of the fuel fed to the engine for generating thrust is more efficient in the TVD than in the TRD, since the efficiency η_v of the propeller, because of the larger mass of air handled by it, is much higher than the so-called thrust efficiency of the TRD. Because the rotational speed of the propeller is substantially less than the rotational speed of the turbocompressor, the TVD is equipped with reduction gearing.

Figure 4 presents a diagram of a TVD in which a single turbine is used for turning both the compressor and the propeller (through a reduction gear). Since the turbine of a TVD should generate correspondingly greater power, at low flight speeds the turbine expands the gases nearly back to atmospheric pressure.

In a TVD with separate turbines (Fig. 5), there is, in addition to the compressor turbine, a propeller turbine that is kinematically independent of the compressor turbine. Such a TVD design is used for driving the propeller of a VTOL-type aircraft, where independent operation of the propeller guarantees a number of design and operational advantages [27].

In addition to the classification of gas turbine engines on the basis of thrust generation principle, they can also be classified on the basis of various design criteria. Thus, for example, according to the type of compressor, gas-turbine engines may be divided into those with axial compressors (see Figs. 1, 2, 3, 4 and 5), those with centrifugal compressors (Fig. 6), and those with combined or axial-centrifugal compressors (Fig. 7). Gas-turbine engines are also classified according to the type of combustion chamber (TRD's with separate combustion chambers, TRD's with annular chambers, etc.), according to the direction of flow of air and gas (TRD's with direct flow ducts, TRD's with flow ducts having a reverse loop) and according to the type of turbine, as well as a number of other bases for differentiation. TRD's are now beginning to be classified as subsonic or supersonic and high-altitude or low-altitude, according to the region of application and the design features thereby required.

3. BASIC PARAMETERS OF GAS-TURBINE ENGINES AND THEIR DERIVATION

Evaluation of the quality of any type of engine and the determination of the region of reasonable application of the type are based on a whole complex of absolute and relative parameters.

One of the most important parameters of a TRD is the total thrust, denoted by P. The absolute value of thrust depends on the altitude and speed of flight, and also on the program of regulation adopted. Ordinarily (in the absence of some special limitation) the term thrust is applied to the maximum static thrust on the ground, under normal atmospheric conditions.

The first production TRD's (1945) had thrusts of the order of 1000 kg. The subsequent development of the TRD was characterized by both reduced and increased thrust. At present, TRD's have gone into service with thrusts from a few hundred kilograms up to 10,000-12,000 kg and more [27]. Such a broad range of thrusts accounts for the variety of types and designations of aircraft in which the TRD's are installed. The absolute magnitude of the thrust determines the size and weight of the engine. On its magnitude also depend the choice of a reasonable design layout for the engine, its operational properties, and its "specific" parameters.

The requirement of large thrusts for bombers and transport aircraft may be satisfied either by a small number of engines with large thrust, or by a large number of engines with small thrust. In each specific case an optimum thrust is chosen for the single engine, taking into account its weight, the technology and cost of production, reliability and convenience in operation, convenience of mounting in the aircraft, etc.

Turboprop engines are not evaluated according to thrust, but according to the propeller-shaft power N_v, the power from the jet thrust N_r, and the total power N_e, equal to the sum of the two. Here, just as in the case of TRD's, a tendency toward expansion of the range of power is observed. Along with TVD's of large power (up to 6000 hp and more) there are TVD's of small power (200-300 hp).

Another important characteristic of gas-turbine engines is their weight G_{dv}. Since the absolute magnitude of the weight of a TRD depends on the magnitude of the thrust P, the relative weight evaluation of engines with different thrusts is carried out by use of the specific weight

$$\gamma = G_{dv}/P \text{ [kg of weight/kg of thrust]} \qquad (6)$$

In the development of all aviation engines, including turbojet engines, there has been a characteristic effort to decrease specific weight.

Thanks to the efforts of designers, production engineers and metallurgists, the specific weight has been forced down (on the average) from $\gamma = 0.8\text{-}1.0$ (1945) to $\gamma = 0.2\text{-}0.3$ (1959); that is, approximately by a factor of 4.

The quality of a TVD in relation to its weight is evaluated in terms of the specific weight, determined by dividing the weight of the engine by the total power:

$$\gamma = G_{dv}/N_e \text{ [kg of weight/horsepower]}. \qquad (7)$$

The specific weights of TVD's lie within the limits 0.15 to 0.30 kg//hp.

No less important for an aviation engine is its size, that is, the cross-sectional area of the midsection (frontal area) and length of the engine. Since the frontal area F_{lob} depends on the magnitude of the thrust, the relative evaluation of a TRD is made according to the specific frontal-area thrust P_{lob}:

$$P_{lob} = P/F_{lob} \text{ [kg/m}^2\text{]}. \qquad (8)$$

Over a relatively short space of time, the magnitudes of frontal-area thrusts have increased substantially. For engines with axial compressors, the specific frontal-area thrust increased from 2000-2500 kg/m^2 (1945) to 8000-10,000 kg/m^2 (1959), that is, by a factor of approximately 4. Due to the peculiarities of its design, the specific frontal-area thrust of a TRD with centrifugal compressor is substantially smaller, amounting at present to about 3000-3200 kg/m^2.

The economy of a TRD is evaluated by use of the specific propellant consumption C_p, which is equal to the ratio of the hourly rate of fuel flow rate G_t to the thrust P generated by the engine:

$$C_p = G_t/P \text{ [kg of fuel/(kg of thrust/hour)]}. \qquad (9)$$

While in 1945 the specific fuel consumption was 1.3-1.5 kg of fuel/kg of thrust/hour, at present it has fallen to 0.75 to 0.9 [9; 29], that is, a decrease by a factor of nearly 2.

The economy of a TVD may be evaluated according to the specific power fuel consumption C_e, which is equal to the ratio of hourly fuel flow rate to the power:

$$C_e = G_t/N_e \text{ [kg fuel/hp-hour]}. \tag{10}$$

Under normal static-test conditions, the specific fuel consumption of modern TVD's lies in the range 0.22-0.3 kg/hp-hour.

The values of C_p and C_e are interrelated. Comparing the expressions for C_p and C_e, it is possible to write that

$$C_p = \frac{N_e}{P} C_e \tag{11}$$

In static operation of a TVD, the ratio of total power to total thrust is approximately equal to 0.9. Therefore, under ground test conditions,

$$C_p = 0.9 C_e \tag{12}$$

Setting into this expression the previously mentioned limits of C_e for TVD's, we obtain

$$C_p = 0.2 - 0.27 \text{ [kg of fuel/kg of thrust]}.$$

Thus, under static test conditions the TVD is about 3 to 4 times more economical than the TRD. In flight, the ratio of power to thrust changes as a function of speed.

$$\frac{N_e}{P} = \frac{PC_H}{75P\eta_v} = \frac{C_H}{75\eta_v}, \tag{13}$$

where C_H is the speed of the aircraft in meters/sec and η_v is the propeller efficiency.

Setting the value N_e/P into Expression (11), we obtain

$$C_p = \frac{C_H}{75\eta_v} C_e \tag{14}$$

With increasing flight speed, C_p increases, and the advantages of TVD's over TRD's decrease. At flight speeds of about 400 m/sec, a powerplant with TVD loses its advantage in relation to economy in comparison with the powerplant with TRD.

Very important characteristics of gas-turbine engines are their service life and durability.

The service life of an engine is characterized by its time of operation between overhauls. The term "overhaul" means complete dismantling of the machine, inspection and reconditioning of parts, replacement of defective ones, and rebuilding of the engine.

Over the time of the development of the TRD, thanks to improvements in design and technology, careful adjustments and accumulation of experience in use, the service lives have substantially increased. While in 1945 the time between overhauls of a TRD was 20-25 hours, the time has increased to several hundreds of hours at the present.

The time between overhauls for an engine is fixed not only by technical factors, but by actual requirements depending on the designation of the aircraft. It is obvious that it is unnecessary to specify a time between overhauls for a motor, when this time is considerably greater than the service life of the aircraft itself. For TRD's in aircraft intended for use just once, the "service time" may be reduced to a few hours, since it is necessary that this time exceed the length of the single flight by only a short period. Therefore TRD's with short time between overhaul may be made simpler and cheaper by not using critical materials; or the engines may be used with considerably higher thrust augmentation than would be permitted with TRD's with long times between overhauls.

The durability of an engine is characterized by the total time of its service; that is, it is determined by the number of overhauls and the period between overhauls specified for it.

4. BASIC REQUIREMENTS FOR THE DESIGN OF ENGINES AND THEIR REALIZATION

The following basic requirements exist for the design of aviation gas-turbine engines.

1. The magnitude of the thrust (for TRD's or power (for TVD's) generated by the engine should correspond to the thrust necessary for

meeting the flight-engineering specifications imposed on the aircraft.

2. The design of the engine should certainly take into account the peculiarities of the airframe into which it is to be installed.

3. The engine should have the lowest possible specific weight and the largest possible frontal-area thrust.

4. The specific fuel consumption in the range of operational modes of the engine should be as small as possible.

5. The design of the engine should be simple and adaptable to production and should require as little use of scarce materials as possible.

6. In operation, the engine should be simple and reliable, have a long period between overhauls, not require time-consuming frequent routine servicing, and, on overhaul, permit rapid disassembly and reassembly without need for complicated fixtures.

7. The engine should be balanced, and should not cause harmful vibration of the engine mounts.

8. Control of the engine should be simple, with good pickup mandatory.

The requirements listed are in many respects mutually contradictory. Therefore in laying out and fabricating a specific type of engine, the designer often makes engineering compromises, or tries to satisfy the most important requirements, which depend on the service required of the aircraft for which the engine is to be built.

Engines intended for long-range bombers or transport aircraft should be economical and reliable, even at the expense of a somewhat increased specific weight. For fighters, on the other hand, small weight is very important for the engine, whereas the economy plays a secondary role.

Engines intended for fighters should be built so as to take into account the greater maneuverability of the aircraft, and the consequently high g-forces encountered in actual operation. The design of engines for aircraft with high supersonic flight speed (M_H equal to 2.0-3.5) should guarantee operation at high temperatures of the air entering the engine, and should also support operation in a wide range of the temperature variation and dynamic pressure of the air at the intake of the motor.

Engines designed for operation at high altitudes are designed so as to take into account the low density and low pressure of the air. In design of engines intended for aircraft with high speeds at low altitudes, the high absolute pressures of the air in the air ducts of the engine are taken into account, as is the increased aerodynamic loading on the elements of the rotor, etc. [27].

If the engine is intended for vertical takeoff aircraft, a special importance is attached to reduction of specific weight, the requirement of long-term continuous operation in a vertical position, excellent pickup, and absolute reliability, etc. [27].

Small specific weight and large frontal-area thrust are enhanced by certain measures, the most important of which is to increase the specific thrust P_{ud}, that is, the thrust generated by each kilogram of air handled by the engine per second.

The specific thrust is increased by reduction of the hydraulic losses in the compressor and turbine, that is, by increasing the shaft efficiency of the compressor and turbine. The specific thrust depends on the effective ratio of the air in the compressor, and for a certain optimum value approaches a maximum. In design, therefore, one ordinarily attempts to ensure that the compressor has an effective pressure ratio π_k^* close to the optimum. While in 1945 the value of π_k^* varied within the limits 2.8-4.0, most production TRD's now have π_k^* =

= 6-12.

The most important factor increasing P_{ud} and decreasing the weight and size of the engine is the temperature T_3^*. Especially important is the increase of temperature T_3^* in engines intended for high flight speeds and having substantial temperature rise in the air because of dynamic pressure. The higher the temperature T_3^*, other things being equal, the more efficient will the TRD be at higher flight speeds.

A possibility of increasing the temperature T_3^* is offered by the use of more highly refractory materials for fabricating the turbine and use of a more effective cooling system. During the time of development of the TRD's, the temperature T_3^* has been increased by 200-230° (that is, from 1000-1030 to 1200-1250° abs). Further achievements in the increase of the temperature T_3^* will depend, as before, on the development by metallurgists of refractory alloys, and on development of more effective cooling methods. Measures to increase the shaft efficiency of the compressor and turbine, increasing π_k^* and temperature T_3^*, have resulted in the situation that the specific thrust under normal test-stand conditions has increased from 40-50 kg sec/kg to 70 kg sec/kg.

The heating of gases in the afterburner up to 1600-2000° abs [9; 29] permits, under test-stand conditions, an increase in the specific thrust P_{ud} by 30-50% at the expense of a comparatively small increase in weight of the engine.

A very important parameter of the TRD exerting an influence on its weight and size is the specific air consumption $G_{v\ ud}$, which is equal to the ratio of the per-second air flow rate G_v to the frontal area of the engine F_{lob}:

$$G_{v\ ud} = G_v/F_{lob}\ [\text{kg of air/m}^2\text{-sec}]. \qquad (15)$$

The larger the specific air flow rate, the lower the weight and size of the engine under otherwise similar conditions. In engines with centrifugal compressors, the frontal area is determined by the size of the compressor, the diametric size of which considerably exceeds the diameter of the turbine. The first production TRD's with centrifugal compressors having a double intake (see Fig. 6) had $G_{v\ ud}$ equal to 35-40 kg/m^2-sec. Subsequently, by increasing the air speed over the impeller and varying the relationship between its inlet and outlet with a simultaneous increase of circumferential speed, brought about an increase in the specific air flow rate up to 60 kg/m^2-sec, which may be considered a practical limit for a TRD with a centrifugal compressor.

Axial compressors have diameters that ordinarily do not exceed those of the combustion chamber and turbine. In view of this circumstance, even the first production TRD's with axial compressors had $G_{v\ ud}$ equal to 75-90 kg/m^2-sec, which exceeded the specific air flow rate of a TRD with centrifugal compressor of the same period by a factor of more than 2. Subsequently, as a consequence of an increase in the axial speed of air at the inlet of the compressor, and decrease in the diameter of its shaft, the specific air flow rate was increased to 200 kg/m^2-sec.

For given values of the specific thrust and specific air flow rate, small specific weight is ensured by lightening the engine parts. Here an important part is played by the experience and skill of the designers, high quality of the technology, and high quality of the materials.

Excellent balance is accomplished by careful balancing of the rotor of the TRD and taking measures to ensure freedom from distortion of the rotor shape during operation.

Excellent controllability of the engine, that is, the capability of rapid change from one operational regime to another without complex manipulations by the pilot, and the ability to hold automatically to a preset regime of operation are achieved by means of making sure that the characteristics of the compressor and turbine are suitable and by installation of an automatic control system on the engine.

5. BASIC INFORMATION ON THE DESIGN OF GTD [27]

Aviation gas-turbine engines are built so as to take into account the specified flight and technical requirements dictated by the anticipated application of the engine, and on the basis of a broad analysis of the current and prospective state of the art in engine development.

The design of the engine begins with a choice of general geometry and basic thermal and gasdynamic parameters. Thereafter, calculations are carried out for its thermo- and gas-dynamics, and to determine the geometric dimensions of the flow passage of the engine: intake section, compressor, combustion chamber, turbine, and exhaust section.

Since the thermo- and gas-dynamic design requires knowledge of experimental quantities and the interrelation of the parameters of the engine, it is usual during the design of the engine to set up and test models of the separate parts: compressor, combustion chamber, and turbine.

After refinement of the design and the resulting changes in the intended shape of the engine, its characteristics are calculated and the plans for the engine are sketched out, usually in several variations for choice of the best design solutions. The preliminary sketch is used as a basis for carrying out the strength calculations for the engine, after which necessary changes are made in the design. Then detailed consideration is given to the matters of lubrication, cooling, arrangements of auxiliary systems and their drives, etc. The auxiliary systems for the motor being designed are either chosen from among the systems already available or else they are developed specifically for the engine.

After the rough design, a detailed part-by-part design is carried through. From the working drawings prepared, an initial check set-up of the various parts is first carried through, followed by a check set-up of the entire engine. The checked drawings are used to work out the technology and the components are fabricated, after which the parts and the entire engine are assembled. The lines of the lubricating and fuel-supply systems are adjusted to the engine model.

Tuneup tests constitute one of the important and time-consuming stages in the development of a new engine. In the process of tuning the engine, design changes are introduced with the objective of removing defects. Whenever necessary, supplementary component tests are carried through and their characteristics recorded. Design changes must without fail be accompanied by checking of gasdynamic and strength calculations.

Subsequent testing permits ascertaining the operational properties of the engine, as well as removing defects, working up the instructions for operation and preparation of the engine for mass production. Accumulation of test data from a large number of applications and refinement of design and technology, contribute to increased reliability and lengthening of the time between overhauls of the engine.

The eventual fate of any engine that is built depends primarily on how advanced it is. Therefore in designing an engine, one always strives to use the latest possible principles, elements of inventiveness and originality, chosen so that the design will reflect all the most recent achievements of aviation engine building and so that the engine to be designed will be significantly distinguished by superior capabilities from those already in general use. Otherwise there is no adequate justification for the production of a new engine in quantity if it has become obsolete before it reaches the stage of quantity production.

The construction details should as a rule take account of all the available possibilities of technology and metallurgy, so as to exploit them to the full, but at the same time, they should not be overestimated, since this might lead to delay or even complete impractibility in actually implementing the design.

In order to accelerate the completion of an aviation engine, it is very important to be able to carry out experimental investigations, since the creation of a new engine design involves solving complex problems in which theory is not always able to yield the requisite accuracy in prediction. This justifies the fact that large experimental installations are being set up to permit carrying out engine tests and tests of their components, not only for normal sea-level conditions, but also for high speed and altitude flight conditions.

The history of the development of aviation-engine design shows that in the creation of a new engine, along with new and original design solutions, old and well recognized ones are also used. In occasional cases, engine builders may also turn to old design forms the application of which under new conditions appears to be convenient. Therefore it is very important for designers to know the history of engine design. Such knowledge, furthermore, permits the designer to build new engines without duplicating errors made by previous designers. The so-called "intuitive" solution of design problems is always based to one degree or another on considerations of the historical experience of engine designers.

In the design of engines it is very important to know how to use such designs, particularly in those cases when it is required to develop a new engine with thrust greater or less than that of an engine already in existence (prototype). Using the theory of similarity, it is possible to evaluate the specific parameters of the new design of engine, its strength and other parameters quite easily on the basis of a given prototype engine.

In projecting an engine design, the engineer should choose, from among all the various possibilities in the design solutions for the problem placed before him, the one that is optimum – a choice which is not, as a rule, at all obvious.

The correct solution of the problem placed before the designer is always characterized by elegance in the design forms and by reciprocal harmony. The finally adopted solution of the design problem is influenced by a number of factors: not only the requirement of minimum weight with adequate reliability, but also requirements of simple shapes for fabrication, conformity of each section with the others, assurance of the possibility of assembly and disassembly of the engine, technological simplicity in the fabrication of the parts, etc. All of this, will unquestionably depend on the skill of the designer, on his experience, his knowledge of production engineering, metallurgy, etc.

In the design of the engine one must take care that the design itself resists improper assembly and damage during the assembly process, and at the same time ensure normal operation of the engine even by ill-qualified personnel.

A lightweight and reliable design should have equal strength of parts. Ensuring their equality is the main problem to be resolved by the designer in the construction of the engine. Inadequate strength of any element in the construction under any operational condition of the engine may cause engine failure, even when it is generally overdesigned.

In order to guarantee equal strength of parts of the engine, it is necessary first that the designer have a clear conception of the way in which the parts of the engine are loaded, the distribution of the forces and moments in the various most characteristic steady-state and transitional regimes, and second, that the designer be able to evaluate, from the loading and strength data of the materials utilized, the strength of the engine parts already designed (checking calculation for strength), and specify the dimensions of the parts subject to design (design calculation for strength).

The determination of the actual static and particularly dynamic loading on the elements in the construction of the engine, like the determination of the actual stresses, is a problem that is very difficult, and often quite insoluble. At this point the designer makes use of experimental research and the so-called comparative strength calculations.

In comparative design of a newly-designed engine, the stresses are determined and then compared with stresses calculated by the same methods in the design of elements used in current similar engines which have been found satisfactory in actual operation.

Of great importance for design is the accumulation of statistical material on stresses and margins of safety of engines in production. In many cases the decisive factor is not the magnitude of the stress, but the amount of deformation (for example, of housings, shafts, etc.). In this case, a calculation is made for the rigidity of the engine design.

Very many units and parts of an engine (housing, standard fasteners, complex castings) are not generally analyzed for strength. Their dimensions are chosen in design on the basis of technological experience and the operation of similarly built engines.

Part One

COMPRESSORS

Chapter 1

AXIAL COMPRESSORS

1. PRINCIPLES OF OPERATION- BASIC PARAMETERS AND CONSTRUCTIONAL GEOMETRY OF AXIAL COMPRESSORS.

Axial compressors (Fig. 8) are ordinarily of multistage construction. An axial compressor consists of an inlet section and several rows along the axial direction of alternate moving blades 1, set on the rotor 3, and fixed (stator) blades 2, fixed to the casing 4 of the compressor. The combination of one row of rotor (or moving) blades and the row of stator (or fixed) blades immediately following is called one compressor stage.

In an axial compressor, the direction of the air flow is primarily axial. In the flow passages formed by the rotor blades external mechanical energy supplied from the turbine is imparted to the air, as a result of which the pressure and velocity of the air are increased. In the stator vane assembly downstream of the rotor blades, the kinetic energy of the air is transformed into potential energy, i.e., by reducing the velocity of the airstream, the pressure is increased. The stator vane assembly also fixes the proper direction of air flow for its entry into the following stage.

The ratio of pressure at outlet from the stage to the pressure at inlet is called the pressure rise ratio of the stage. The pressure rise ratio of a multistage compressor is greater, the greater the pressure rise ratio of the individual stages, and the greater the number of stages.

The pressure rise ratio in a stage of an axial compressor depends basically on the average circumferential speed of the rotor blades, that is, on the circumferential speed of the blades at midlength. The greater this speed, the greater the pressure rise ratio. The mean circumferential speed of blades because of considerations of strength ordinarily does not exceed 300-450 meters per second.

The diameter of a compressor stage is determined by the required air flow rate, the air density, and its axial speed. In all the stages the axial speed of the air is maintained either constant or it decreases slightly in the later stages. Since the density of air at entry into the first stage is minimal, it has the largest flow cross section, and that cross section decreases in the later stages. The flow cross section is limited by its outside and inside diameters. For the purpose of decreasing the outside diameter of the first stage for a given flow cross section, the inside diameter is reduced; in order to provide room for the blades around the rotor, the inside diameter is no less than 0.35-0.4 of the outside diameter.

In subsequent stages the outside diameter may be retained at the same magnitude as that of the first stage (Fig. 9a); the inside diameter may also be kept the same (Fig. 9b), and the mean diameter may be retained (Fig. 9c).

In the first case, the required reduction of the flow cross section (by reason of the increase in density of the air) is accomplished by increasing the inside diameter of the flow cross section. In this case the mean circumferential speed of the stages increases, and consequently, their pressure rise ratios also increase. Along with this advantage, the compressor so constructed exhibits a shortcoming which consists in the reduced length of the blades in the later stages. The clearance between the blade tips and the casing of the compressor is relatively greater when the blades are short,

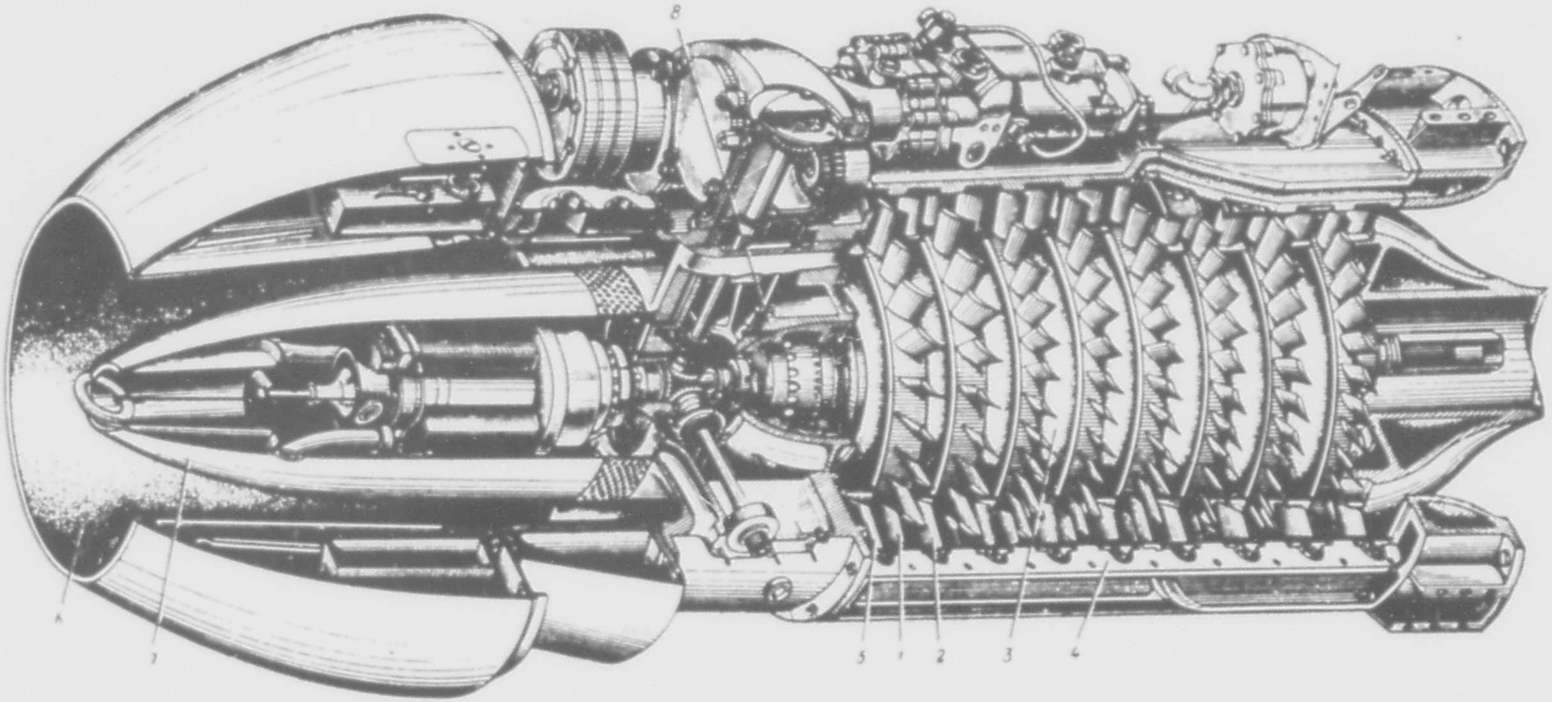

Fig. 8. Axial compressor of the RD-10 engine. 1) Rotor blades; 2) blades of the stator vane assembly; 3) rotor; 4) compressor casing; 5) guide-vane assembly; 6) housing of the entrance section; 7) inner fairing; 8) forward bearing housing.

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than when they are long. As a result, in a compressor with short blades there is an increase in the recirculation of air in the clearance space, and consequently, a decrease in the pressure rise ratio of the compressor.

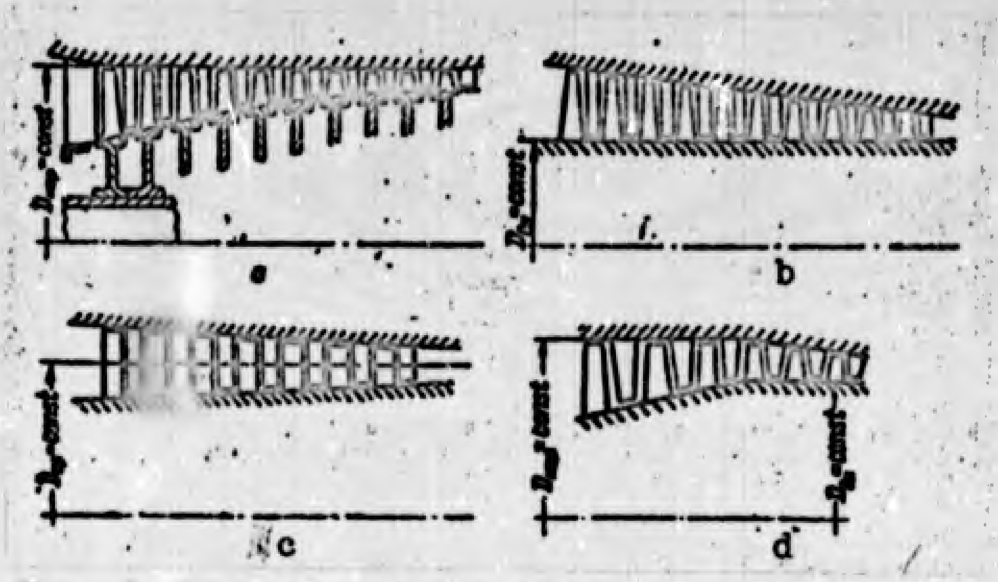

Fig. 9. Diagrams of the flow cross-section pro profiles. a) With constant outside diameter; b) with constant inside diameter; c) with constant mean diameter; d) with variable outside, inside, and mean diameters.

With constant inside or mean diameter (Fig. 9b and c), the blades of the later stages are longer than in the preceding case, and the recirculation in the clearance space is reduced. The pressure rise ratio of the stage remains constant (with constant mean diameter) or is reduced (with constant inside diameter) as a result of its dependence on the average circumferential speed.

In practice it is possible to use compressors with flow cross-sections having a combination of the variations considered here (Fig. 9d).

The axial speed at the inlet to compressors in operational designs exceeds 220-230 m/sec. Speeds higher than this do not lead to substantial decrease in the flow cross section, since the density of the air at the compressor inlet thereby is reduced substantially.

In compressors having the indicated maximum values of axial and circumferential speeds, the air enters the interblade passages of the first stage with high relative speed, which is the geometrical sum of the axial and circumferential speed (Fig. 10). In this case, the relative velocity may be larger than sonic velocity, and special profiling (shaping) of blade and interblade passage is required. Stages designed for operation at supersonic relative air velocities are called supersonic stages. In some compressor designs, all or several of the first stages may be supersonic.

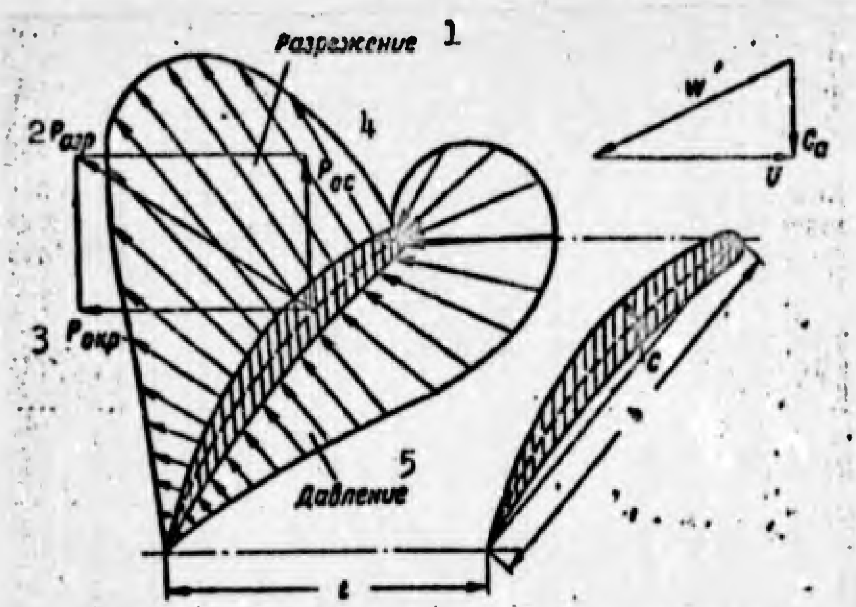

Fig. 10. Profile of the rotor blades of a subsonic stage. P_{aer}) Aerodynamic force; P_{os}) axial component; P_{okr}) circumferential component; U) circumferential speed; C_a) axial speed; W) relative speed; b) chord; c) thickness of profile; t) pitch of blades; 1) Rarefaction; 2) P_{aer}; 3) P_{okr}; 4) P_{os}; 5) pressure.

The profiles of rotor blades intended for subsonic and supersonic stages are shown in Figs. 10 and 11.

The axial compressors now in service have pressure rise ratios within the limits 10-12, and their specific air flow rates range from 100 to 200 kg/m^2 sec [3].

Axial compressors with large values of pressure rise ratios, and especially those composed of stages having large pressure rise ratios, have a relatively narrow range of stable (surge-free) operation, and low efficiency values under nonrated conditions. Unstable (surge) operating regimes arise by reason of deterioration of the flow around the rotor blades and the breaking away of the flow upon change in the air flow rate relative to the designed (rated) condition. The surging regime is characterized by oscillations of pressure, pulsations of flow, and by vibration shaking of the compressor and the engine as a whole.

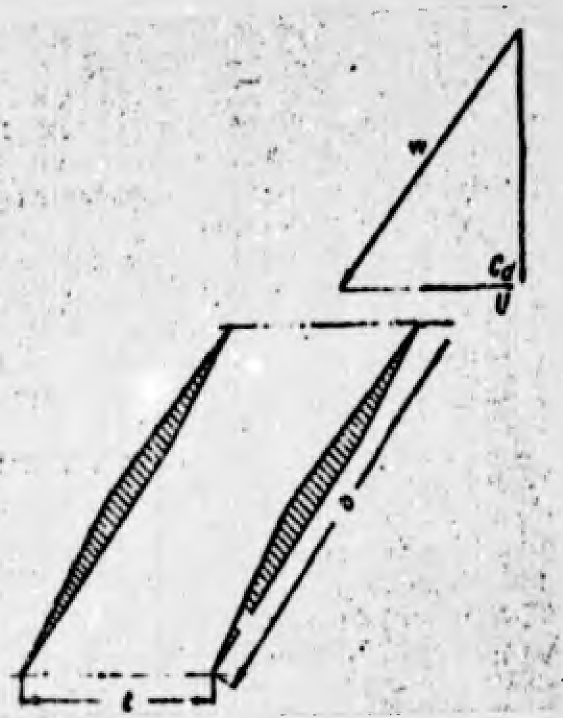

Fig. 11. Profile of the rotor blades of a supersonic stage.

In order to broaden the range of stable operation and increase the efficiency of operation under nonrated conditions, two-speed axial compressors are used (Fig. 12). In a two-speed compressor, two rotors are placed in tandem – the low- and high-pressure rotors – independently driven by separate turbines.

The advantages of the two-spool compressor are as follows. When the compressor is separated into two parts, each of them has a relatively low rated pressure rise ratio, and consequently, a reduced tendency to surge. At the same time, under nonrated conditions, because of the peculiarities of the distribution of the pressure differences between the turbines and the loading of the compressors, changes in rotational shaft speed result; here the departure from the design condition of flow across the blading of both compressors is reduced to a minimum.

In single-rotor axial compressors, in order to expand the range of stable operation, special antisurging devices are often used,

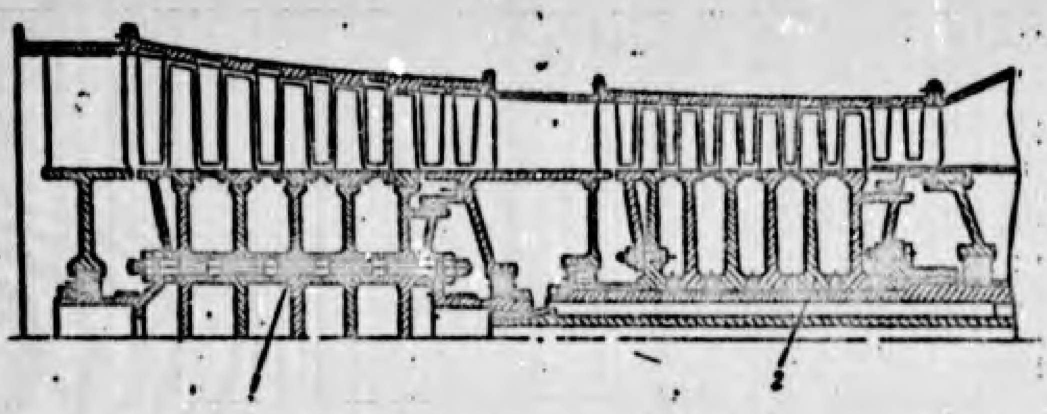

Fig. 12. Diagram of a two-spool axial compressor. 1) Low-pressure rotor; 2) high-pressure rotor.

i.e., rotating guide vanes, and bleed off valves.

At high pressure rise ratios, the heating of the air from the compression process in an axial compressor amounts to 250-300° C. Since in flight at high supersonic speeds heating of the air also occurs as it is decelerated in the entrance section, the temperature of the air at the exit from the axial compressor may amount to 500°C and above (for Mach numbers of 2.5 to 3).

2. SECTIONS OF AXIAL COMPRESSORS

The inlet section serves to introduce air into the compressor at a definite [controlled] speed and to convert the dynamic pressure of the approaching air into static pressure. The losses in the inlet section should be minimal. For this purpose the inlet section is so shaped that the compression of the air by dynamic pressure will take place entirely, or for the most part, ahead of the engine; the air must then flow through a converging duct to even out the flow, and to guide the air into the compressor rotor.

The inlet section of an engine intended for supersonic flight speeds (Fig. 13) has a special shape. The cone of the entrance section is designed for generating a system of oblique shocks, terminated by a normal shock. Such a means of transition from supersonic

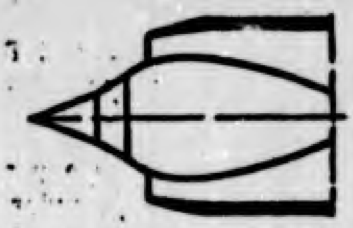

Fig. 13. Diagram of the inlet section of an engine intended for flight at supersonic speeds.

to subsonic speed reduces losses to a minimum in the conversion of the dynamic pressure head into static pressure. For varying flight speeds, it is necessary to regulate the inlet section, shifting the cone, or changing the flow cross-section. This circumstance complicates the design of the inlet section.

For subsonic or transonic flight speeds an inlet section of constant shape is used, similar to that shown in Fig. 8.

The inlet section is formed by an outer shaped shell 6 (in its absence, the skin or covering of the compartment in the aircraft containing the engine), an inner fairing 7, and channels in the casing 8 of the front bearing. On the inside, the walls of the inlet section are subject to a pressure greater than the ambient. Radial and axial forces are transmitted as they arise in the front bearing through special struts and in a turboprop engine, these struts also transmit the forces due to the thrust of the propeller and the torque of the reduction gear.

To protect the compressor blades from harmful solid particles or chunks of ice, the inlet section is sometimes fitted with protective screens 2 (Fig. 14). Such screens cause additional resistance in the engine inlet, and decrease thrust and economy. In some design versions, screens are used that are automatically retracted after takeoff.

At air temperatures within limits of $\pm 5^{\circ}$ C, and humidity of more than 2 g/m^3, the inlet section ices up. For prevention of icing, an anti-icing liquid is sprayed into the inlet section, or the parts are heated to a temperature of +20 to +40° C. Figure 14 shows an anti-icing system in which the liquid is fed into the inlet section by a rotating spray injector 1.

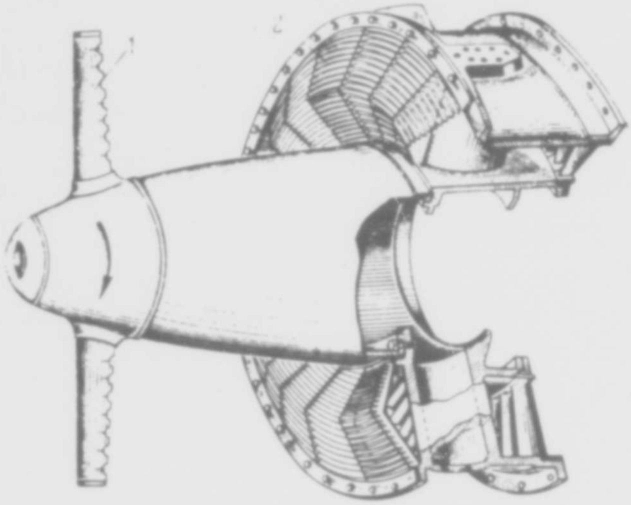

Fig. 14. Inlet section with protective screen. 1) Injector of anti-icing system; 2) protective screen.

The heating of the protective screen, nose fairings, struts in the inlet section, and the blading of the guide-vane assembly is accomplished by use of an electric current, warm air, or gas.

For electric heating, wire spirals are put inside the elements that are to be heated, and current is passed through the spiral from a special generator. The protective screen may be made up directly of conducting spirals. With this method of heating, the expenditure of a large amount of electrical energy is required.

In the second method of heating, hot air from the compressor, or hot gas from the exhaust system, is ejected through the hollow parts of the inlet-section components. In the schematic diagram (Fig. 15) of the warm-air heating system the method of supply and removal, as well as the path of the heated air, is shown. The amount of air introduced is regulated by use of a calibrated orifice 7. The heating system is connected upon the opening of the gate valve 8.

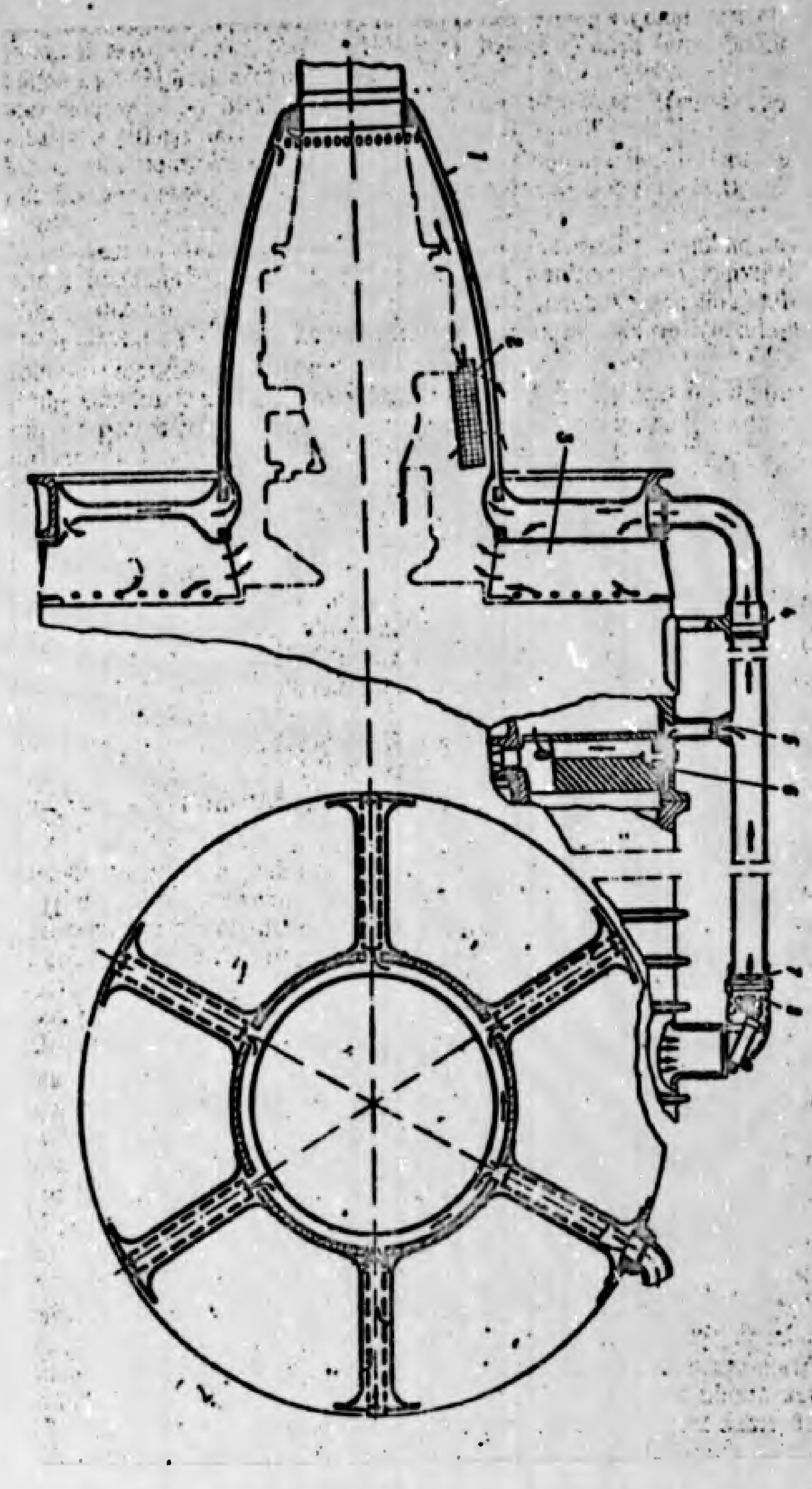

Fig. 15. Diagram of inlet-section heating. 1) Nose fairing; 2) aperture for outflow of heating air; 3) island fairing; 4, 5, and 7) calibrated orifices; 6) rotating guide vane; 8) gate valve.

If the air flow rate is not large, the heating system may be arranged to operate at all times, never being disconnected.

Heating by use of hot gas is applied relatively infrequently, because the parts flushed by the hot gas are subject to corrosion.

For fabrication of inlet-section parts operating at air-temperature range of 150-200° C at the compressor inlet, aluminum alloys are used; for lower temperatures, magnesium alloys are used. For much higher temperatures, heat-resistant aluminum alloys are used; welded parts are ordinarily made of steel.

3. DESIGN OF ROTOR BLADES

A rotor blade has a shaped part (the fin) 1 (Fig. 16), and the keying part (the shank) 2.

The fins of the blade in subsonic stages have profiles similar to the profiles of propellers or aircraft wings, whereas the profiles of blades for supersonic stages are wedge-shaped. This latter circumstance is explained by the fact that when a supersonic stream flows over the leading edge of a wedge-shaped blade, compression shocks of low intensity are formed, and the losses upon transition to subsonic speed in the channel between the blades will be less than on the blunt leading edge of the ordinary profiles, characterized by very intense compression shocks.

The profile thickness of rotor blades amounts ordinarily to 2.5-8% of the chord length. Thinner profiles, having smaller drag, are not used because of the difficulty of fabrication and insufficient strenth.

In conformity with the various conditions of flow incidence at various radii of the rotor blades, their profiles are set at different angles in relation to the axial direction of the airflow. As a consequence, the fin of a blade has a twist (see Fig. 16a).

The key has the function of transmitting force from the blade to the rotor, locating it precisely, and is designed to insure easy assembly of the rotor, or replacement of blades in case of damage.

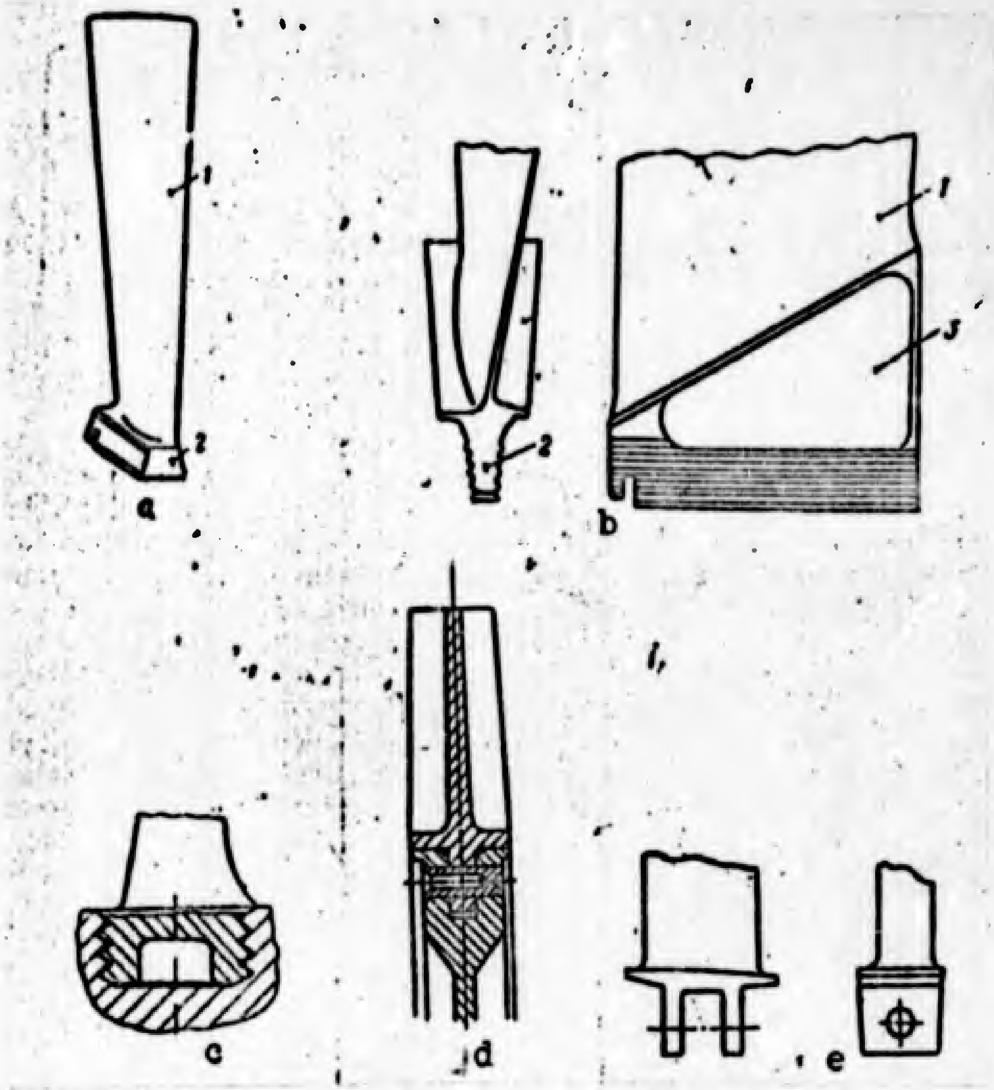

Fig. 16. Rotor blades. a) With trapezoidal key slot; b) with pine- or fir-tree key slot; c) with toothed key slot; d) with triangular ridges; e) with doweled key slot; 1) Fin or scoop; 2) root; 3) transitional part of the blade.

Keys of blades are differentiated into trapezoidal, ("dovetail"), fir tree, toothed, and doweled types.

The trapezoidal key (see Fig. 16,a) has the shape of a symmetrical trapezium. Forces are transmitted through the side walls to the disk from the shank of the blade. The root of the blade is set into a slot with zero clearance. The keys are kept from displacement along the groove by locks, for which purpose screws, axial and radial dowels, plate locks and slotted rings are used (Fig. 17).

In case of attachment by a screw 1 (Fig. 17,a) the hole beneath

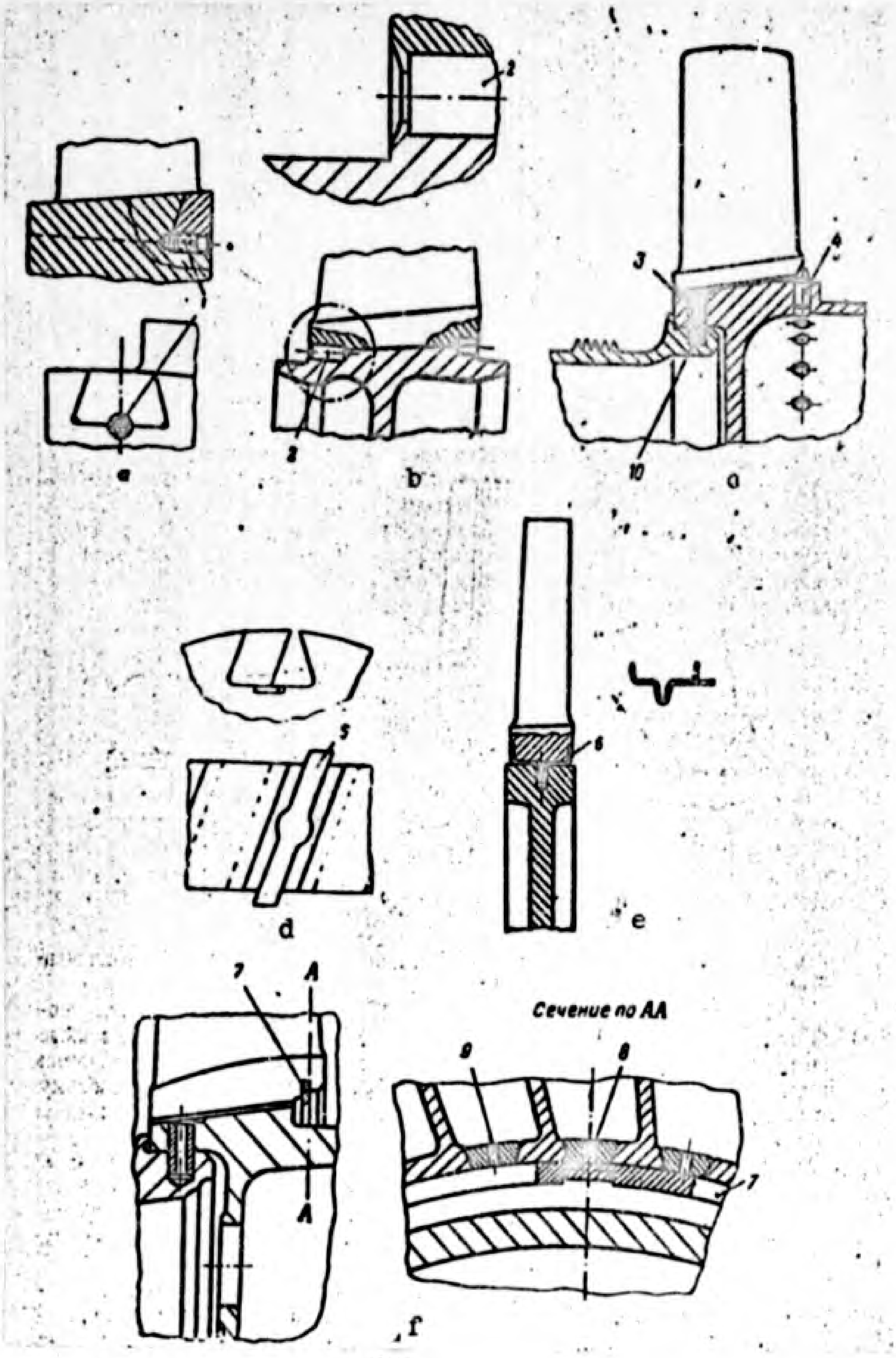

Fig. 17. Methods of attachment of blades to disks. a) Attachment by a screw; b) attachment by an axial dowel; c) attachment by a radial dowel and plate lock; d) attachment by a plate lock with an enlargement; e) attachment by a wire lock; f) simultaneous attachment of all blades by a slotted spring ring; 1) screw; 2) axial dowel; 3) horn of plate lock; 4) radial dowel;

5) plate lock; 6) wire lock; 7) slotted collar; 8) wire ring lock; 9) opening for introduction of removable pin; 10) radial dowel; X) cross-section along AA.

the screw is drilled and threaded from the front face of the rim so that part of the screw enters the body of the key, and part, the body of the disk. After the screw is screwed in it is punched. This method of attachment makes disassembly of the attachment difficult, and is possible only if the material of disk and blade exhibit the same hardness.

An analogous method of making the attachment is by use of an axial dowel 2 (Fig. 17,b). The hole, after setting in the pin, is caulked.

In the design shown in Fig. 17,c, the blade is secured on one side by a radial dowel 4, and on the other, by bending over a horn on the plate lock 3.

Attachment of the blade by use of the plate lock 5, which has an enlargement in the face of the groove, is shown in Fig. 17,d, and a version of the same principle of attachment by a wire lock 6, in Fig. 17,e.

Another method of attachment is the simultaneous attachment of all the blades to the disk by means of a notched spring ring 7 (Fig. 17,f), the junction of which is secured by the lock 8. For removal of the ring, holes 9 are drilled into the rim of the disk, and removable pins are introduced here.

The fir-tree key (see Fig. 16,b) is used relatively rarely in compressors; it is analogous in design to the fir-tree key used in turbines.

The toothed keys with triangular ridges (see Fig. 16,c) are used in drum rotors with annular grooves in view of the ease of fabricating the teeth in the grooves of the drum by use of a shaper tool.

The doweled keys (see Fig. 16,d and e) are used in disk rotors. One or several annular grooves are made in the disk, and the lugs of the blade key are positioned in these grooves. These lugs are joined to the disk by axial dowels pressed into holes, drilled and reamed together for this purpose, in the disk and the keying part of the blade. The dowels absorb forces transmitted from the blades, and work in shear. In order to increase the surface in shear, keys with several dowels and grooves are used. The dowels are held fast axially by being expanded at the side surface of the disk, or by being punched. Such a method of locking makes disassembly difficult.

Blades of stages operating at air temperatures below 210° C are made of aluminium alloys. For higher temperatures, steel is used.

Blades are made by milling, forging, and stamping. The least economical method is milling of blade from bar. In this case, when forging is used, some allowance is made for the final machining to give the blade its final shape. The most up-to-date method is forging with subsequent coining - pressing the forging in a special precision die. After machining or coining, the fins of the steel blades are ground, while aluminium blades are only polished.

The trapezoidal blade keys are fabricated by milling, whereas fir-tree type keys are broached.

4. FORCES ACTING ON BLADES. STRESSES IN BLADES

The contoured part of the blade is subject to aerodynamic and centrifugal forces.

Aerodynamic forces on a blade arise from the flow of air across it, as a result of the pressure difference between the concave side (with a trough shape) and the convex side (on the back) of the profile (see Fig. 10). The resultant of the aerodynamic forces, which we shall refer to, in brief, as the aerodynamic force P_{aer} on the blade (Fig. 18,a), is applied approximately at the midpoint of the blade and it is directed toward the back of the blade, at some angle with respect to the axial direction of motion of the air. The aerodynamic force may be resolved into two components, i.e., the axial component P_{os} and the circumferential component P_{okr}, directed, respectively, along the axis of the rotor and perpendicular to it.

The aerodynamic force causes a bending of the blade. Under this circumstance, just as in the bending of a flat plate (Fig. 18,b), some of the fibers are under compression, some under tension, and some neutral (the length being unchanged). The maximum compression and tensile stresses exist in the fibers furthest from the neutral fibers. In the rotor blade, the compressive streses occur on the back of the blade and the tensile stresses on the concave side.

In each cross section of the blade (Fig. 18,c) the most heavily loaded fibers are on the vertex of the back and near the leading and trailing edges, since those furthest from the neutral line in the cross section are drawn through the neutral fiber. Especially heavily loaded is the root section of the blade, located near the key, since for the root section the arm of the aerodynamic forces is the greatest and, consequently, the bending moment has its maximum value.

The magnitude of the aerodynamic force, and the stresses therefrom, depend on the air flow rate through the engine. The air flow rate through the engine depends on the rotor rpm, and also on the speed and altitude of flight. At maximum flight speed at sea level, under conditions of minimum air temperature, the air flow rate is a maximum. Under these conditions, the stress due to bending reaches maximum values of 2000-2500 kg/cm^2. With minimum air flow rates in

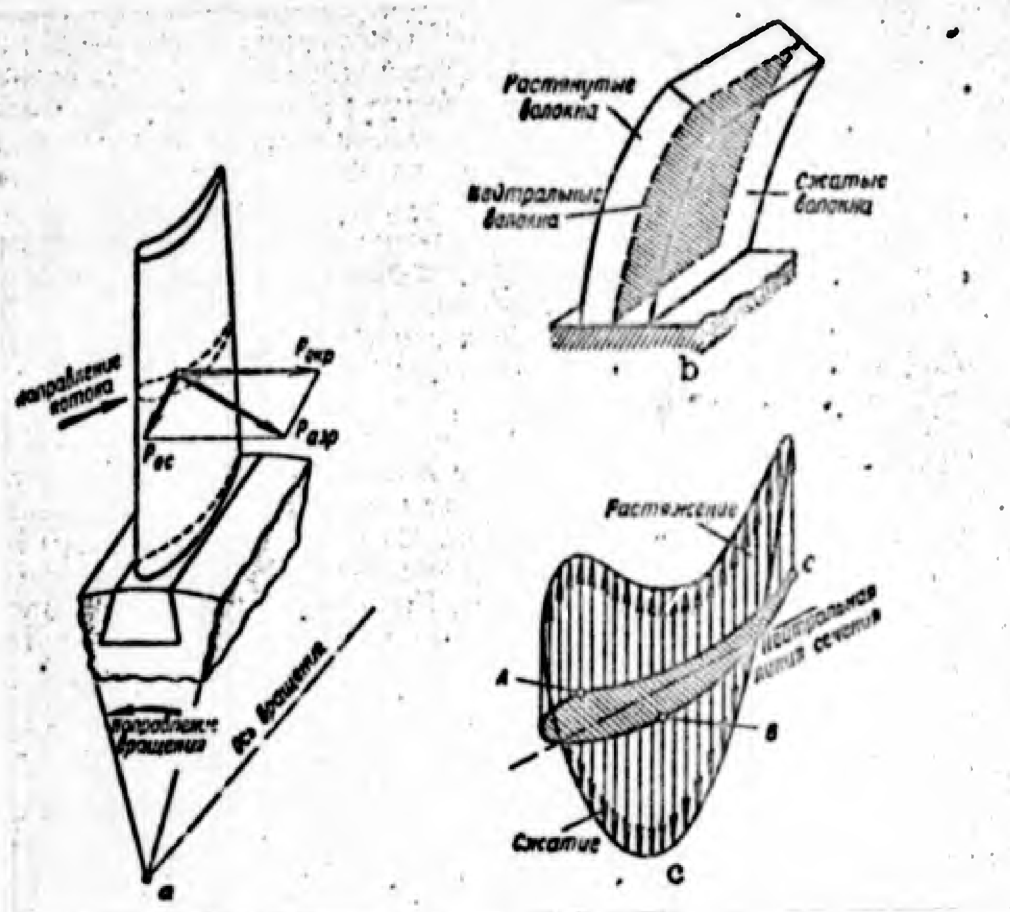

Fig. 18. Illustrations bearing on blade bending because of aerodynamic force. a) Diagram of action of aerodynamic force; b) diagram of bending of a flat plate; c) stresses acting on the blade cross section; P_{aer}) aerodynamic force; P_{okr} and P_{os}) circumferential and axial components of aerodynamic force, respectively; 1) Flow direction; 2) direction of rotation; 3) rotational axis; 4) fiber under tension; 5) neutral fiber; 6) fiber under compression; 7) tension; 8) neutral line of cross section; 9) compression.

the case of flight at the altitude ceiling and at minimum speed the bending stresses amount to 200-300 kg/cm^2.

The action of centrifugal forces reduces to the following. Turning the compressor rotor produces centrifugal forces in the contoured part of the blade, the magnitude of which depends on the mass of the blade, the radius at which its center of gravity is located, and the rotor rpm (Fig. 19). When these parameters are increased in magnitude, the centrifugal forces are increased.

Due to the action of centrifugal forces, the uppermost parts of a blade at each section exert tensile stress which for a constant blade cross section would naturally increase from the periphery toward the root of the blade where their values reach a maximum (Fig. 19,b).

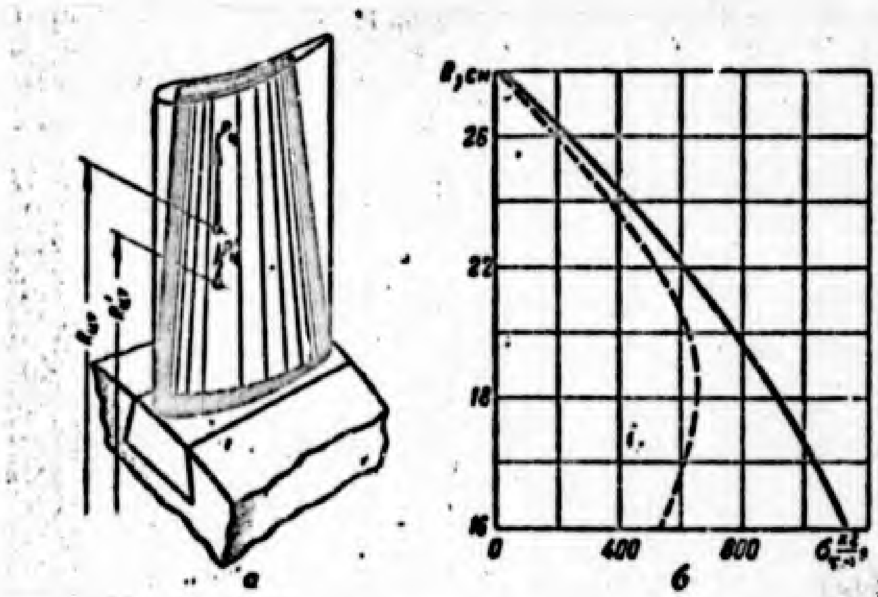

Fig. 19. Loading of blade by centrifugal force. a) Loading diagram of blades of constant and varying cross section in elevation; b) variation of stress because of centrifugal force along the radius; P_{ts} and P'_{ts}) centrifugal forces of blades with constant and variable cross section, respectively; R_{tst} and R'_{tst}) radii of the centers of mass of blades with constant and variable cross section; σ) tensile stress

From a comparison of a blade of constant cross-sectional area along its length with a blade of decreasing area toward the periphery (Fig. 19,a) it is evident that the latter has less mass, and its center of gravity is located at a smaller radius. As a result, the centrifugal force of such a blade is also less than that of a blade with a constant cross-sectional area, as is indicated in the dashed curve of Fig. 19,b. This circumstance is an explanation for the fact that rotor blades ordinarily have cross-sectional areas that decrease toward the periphery. This decrease is achieved in practice by profiles thinner near the tips than at the roots of the blades, and sometimes also by decrease in blade chord.

Centrifugal forces on the contoured part of the blades of the first stages may amount to several tens of tons. As a consequence, it is especially necessary to insure that the rotor is balanced, and therefore it is necessary for the centrifugal forces on the blades in any one stage to be as closely the same as possible. For this reason blades are matched to within tolerances of 5-6 grams.

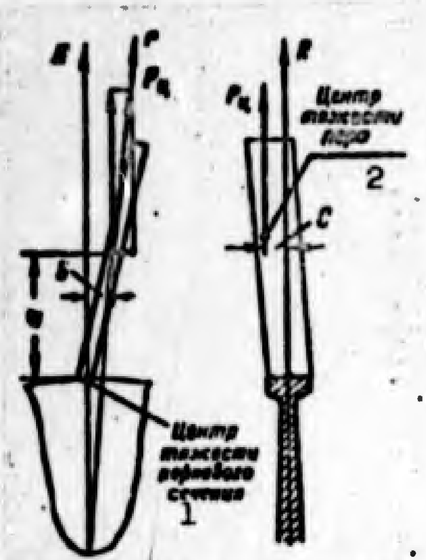

Fig. 20. Compensation of the bending of a blade by use of the moment due to centrifugal forces. 1) Center of gravity of root section; 2) center of gravity of the fin.

Tensile stresses due to centrifugal forces at the most highly loaded cross sections reach 1000-1500 kg/cm^2 in blades made of aluminium alloys, and 3000-3500 kg/cm^2 in those made of steel.

Tensile stresses due to centrifugal and aerodynamic forces are additive, and therefore cracks may develop at the leading and trailing edges of the blade, if the total stresses should exceed the permissible design strength of the material. In order to decrease the stresses in the blade, compensation is applied so that the bending due to aerodynamic forces is countered by bending due to centrifugal forces.

For this purpose the center of gravity of the contoured part is located at some radius r which does not coincide with the radius R through the center of gravity of the root section (Fig. 20). Thus the centrifugal component of the forces induces a bending of the blade at the arms a, b, and c. The arms are selected in such a fashion that the moments due to the centrifugal forces are equal and opposite to the moments due to the axial and circumferential components of the aerodynamic force. With this type of arrangement, bending forces do not develop at the root section of the blade.

Because of the dependence of the bending moment of the aerodynamic forces on the flight regime, complete compensation can be realized only for a particular flight regime; for any change in regime, the compensation is upset.

5. BLADE VIBRATIONS

The rotor blades of a compressor, are subject to periodically changing forces, arising from nonuniformities in the flow of air through the cross section of the air-flow passage of the engine. The action of these forces leads to the generation of forced vibrations of the blades, which under resonance conditions may become dangerous and cause rupture of the blades because of metal fatigue.

In addition to forced vibrations, sometimes self-excited vibration of the blades arise during engine operation such as, for example, vibrations of the bending-torsional type, or destructive flutter, which may set in even in case of uniform airflow.

Fatigue cracks in rotor blades of an axial compressor arise most frequently at the root section, and less frequently in the middle and outer parts of the blade (Fig. 21). Fatigue cracks and rupture caused by harmful vibrations are also generated in the guide- and stator-vane assemblies.

Blade vibrations may be bending, twisting, and bending-twisting (complex).

Let us consider vibrations of a compressor rotor blade rigidly attached to the rim of the disk. If the blade is displaced from its position of equilibrium and then released, because of the action of the elastic forces of the material, the blade will begin to move toward the position of equilibrium. The speed the blade increases. At the equilibrium position, the vibrating blade has its maximum speed, and the elastic forces of the material are equal to zero. After reaching the equilibrium position, the blade is displaced partly by inertia, whereupon the forces of elasticity increase, and the speed decreases. After a short time, the blade comes to rest for an instant, and then under the action of the elastic force, it begins to move in the opposite direction.

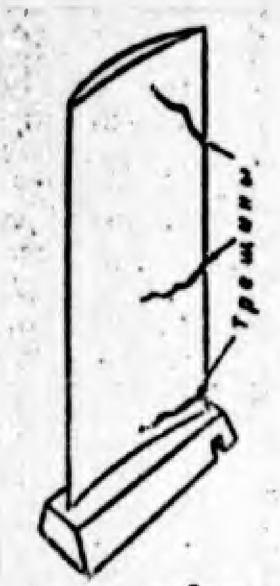

Fig. 21. Location of fatigue cracks in a compressor blade. a) Cracks.

If, in the vibration about the equilibrium position, no external forces act, the vibrations are called free or natural vibrations.

Vibrations of a blade which take place under the continuous action of a periodically changing external exciting force are called forced vibrations.

The vibrations of a blade are characterized by two basic magnitudes, i.e., amplitude and frequency. The amplitude of an oscillation is the largest displacement of a given point on the blade from its position of equilibrium. The frequency of the oscillation is the number of complete blade oscillations that take place per unit time. Ordinarily the frequency of an oscillation is denoted by the letter f, and is measured in cycles per second. If the blade describes forced vibrations, the frequency of the exciting force is defined as the number of complete changes of that force per second. In forced vibrations of a blade, the frequency of such vibration is equal to the frequency of the exciting force.

Blade vibrations are also characterized by the relative oscillation amplitudes of its different parts, called the vibration mode.

The mode, frequency, and amplitude of the free vibrations of a blade are fixed by the conditions of blade distortion from equilibrium, by the geometrical characteristics of the blade, and by the elastic properties of its material. In the free vibrations of a blade, each point describes a motion composed in general of a sum of simple sinusoidal (harmonic) vibrations. The modes corresponding to these harmonic vibrations are called normal modes. The blades of a compressor have an infinite number of normal modes of vibration, but practical interest is limited to those modes which have a connection with failures caused by harmful resonance vibrations in the operational range of engine speed. Ordinarily such modes are the low-frequency first and second bending modes, the first torsional mode, and certain high-frequency complex modes of vibration.

In free vibrations of the blade in one of the normal modes, all of the points vibrate with the same frequency, but with different amplitudes. In such case, certain points of the blade, for example, at the root section, remain motionless. These points are called nodal points, or nodes of the vibration. The geometrical loci of the points (nodes) remaining motionless in a given vibration mode are called nodal lines.

Vibrations of a blade characterized by one nodal line, are called single nodal modes (or vibrations of the first mode); when there are two nodal lines, it is a two-node mode (vibration of the second mode) and so forth.

Figure 22, a, b and c show the three first modes of the free bending vibrations of a rotor blade. In the first bending mode (see Fig. 22,a) all the sections of the blade vibrate, i.e., move back and forth together. The nodal line of the vibration is located in this case immediately at the fixed end of the blade. In oscillations of a blade in the second, third, and higher bending modes (Fig. 22,b and c) the motion of parts of the blade lying on different sides of the nodal lines takes place in opposite directions. The frequency of the second mode of the free bending vibration of a rotor blade is ordinarily 3-5 times that of the first mode, and the third mode has a frequency higher than that of the first, by a factor of 8-10.

In these examples of the bending vibrations of blades, all the lateral cross-sections of the blade move in planes parallel to each other without distortion (deformation) of the shape of the sections. However, there are also forms of blade bending vibrations in which the lateral cross sections are bent (see Fig. 22,d). In such vibrations the blade describes so-called plate bending vibrations. The frequency of such a free plate vibration mode is ordinarily substantially higher in the case of a blade than the frequency of the first bending mode a factor of 12-18).

Figure 23 shows the two lowest modes of free torsional blade vibration (the first and second) in which the blade cross sections oscillate about their equilibrium position. In the first torsional mode (see Fig. 23,a) all the blade cross sections rotate simultaneously in the same direction. The nodal line of the vibration is in this case situated along the blade, approximately in the middle. In blade vibrations of the second (Fig. 23,b) and higher torsional modes the cross-sections in neighboring parts of the blade rotate in opposite directions.

In addition to the pure bending or torsional vibrations, compressor blades may exhibit complex bending-torsional vibrations. The location of the nodal lines for one of the complex-vibration modes of a rotor blade is shown in Fig. 24.

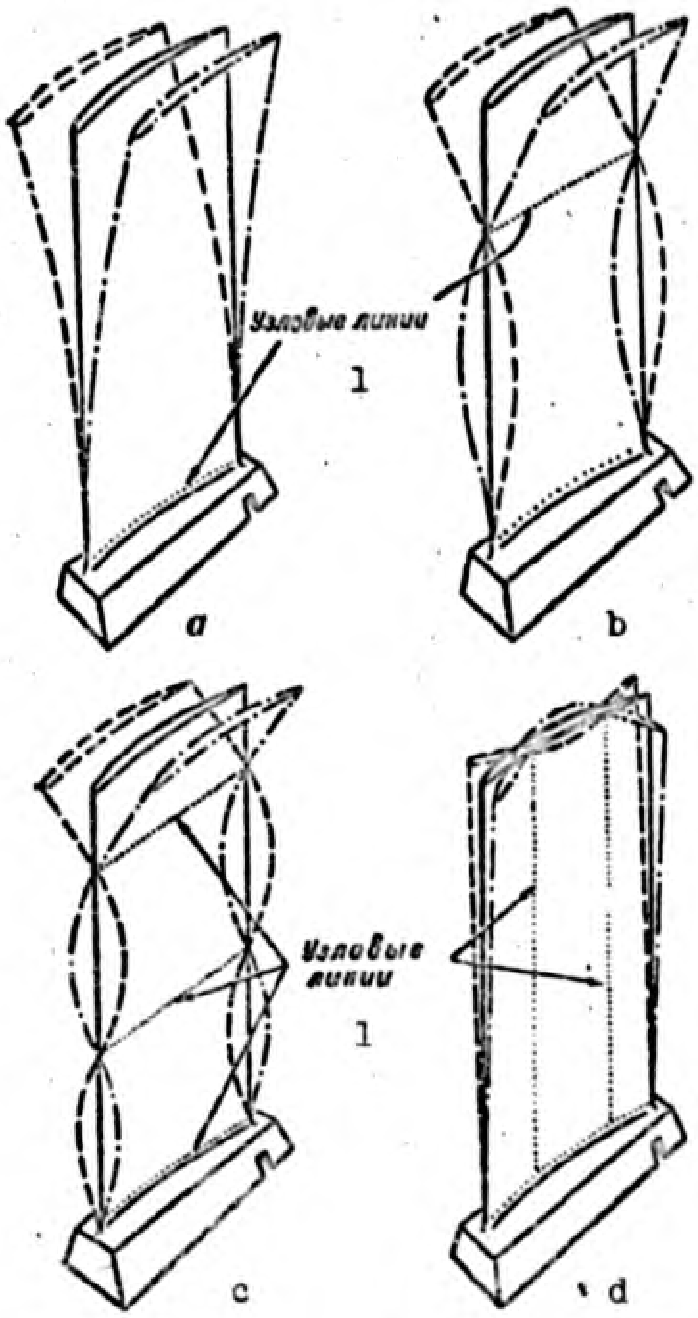

Fig. 22. Modes of blade bending vibration. a) First; b) second; and c) third bar modes; d) plate mode. 1) Nodal lines.

The frequency of the free blade vibrations at elevated temperature is decreased because of the decrease in the modulus of elasticity of the material. The influence of temperature on the frequency of the free vibrations is noted principally on those blades of the later stages of an axial compressor that are fabricated of aluminium alloys. The influence of temperature on steel blades is negligible.

The frequency of free oscillations of blades turning with the disk is increased upon increase in the rotational speed of the disk, because not only the forces of elasticity, but also the centrifugal forces tend to return an oscillating blade to the position of equilibrium, as shown in Fig. 25.

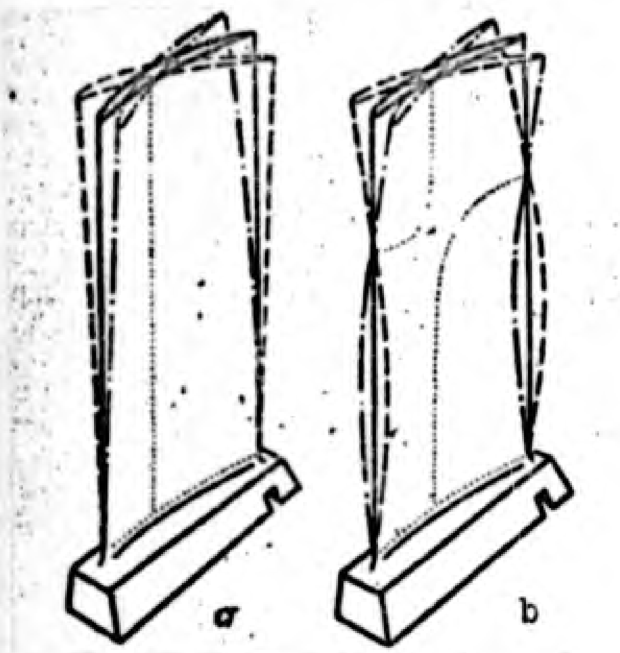

Fig. 23. Modes of torsional blade vibration. a) First; and b) second.

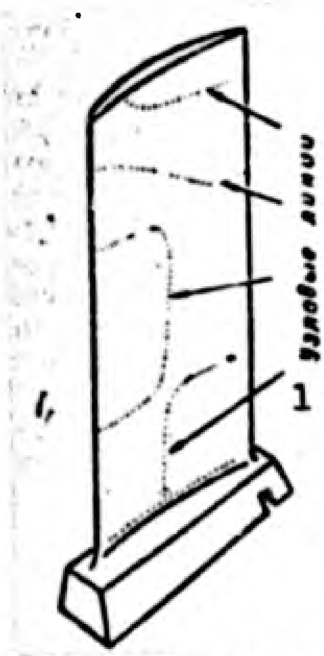

Fig. 24. Complex bending-torsional vibration of a blade. 1) nodal lines.

An increase in the frequency of the first vibration bending mode of rotor blades of axial compressors, due to rotation, may be very considerable (for nominal rotational speeds the frequencies are increased by factors of 1.5-2).

In an axial compressor, nonuniformity of airflow, i.e., nonuniformity of the pressure field and in speed of airflow along the circumference of the air flow passages in the engine, are caused by the parts located in the path of the airflow (ribs and casing struts, the blades of the stator vane assembly, etc.), as well as by pronounced distortion of the flow because of the release of air from antisurge fittings, or by the removal of the air used in cooling the hot parts of the engine. Nonuniformity of flow may be caused by the airflow breaking away from rotor or stator vane assembly when angles of attack are critical or nearly so. In this case, the stall [breakaway] zone ordinarily rotates with the rotor. Disrupted airflow may lead to strong fluctuations in pressure in all parts of the engine flow passages, and to surging which, in turn, may serve as a source of harmful forced blade vibrations.

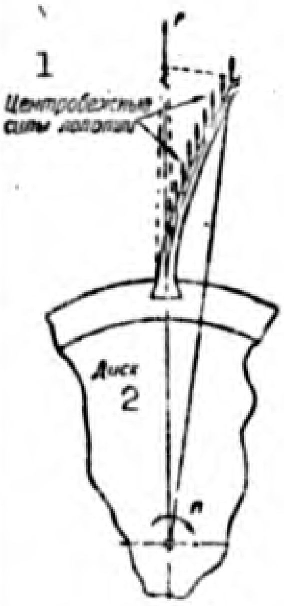

Fig. 25. Diagram of the restoring action of centrifugal forces on a blade attached to a rotating disk. 1) Centrifugal forces on the blade; 2) disk.

With forced vibrations on a rotating blade, there is a periodic force which acts, varying along the circumference of the airflow passage of the engine according to the changes in the pressure and velocity fields. If in a single revolution of the rotor the pressure field of the airflow changes K times, for example, due to K parts interrupting the airflow, the frequency of the exciting force f_{vozb} will be equal to the product of the rotational speed n_{sek} of the rotor in revolutions per second, and the number K:

$$f_{возб} = Kn_{сек} = \frac{Kn}{60}\ \frac{колебаний}{сек}, \qquad (16)$$

where n is the engine rotor rpm.

The frequency of the forced oscillations of the blade, equal to the frequency of the exciting force, increases with increase in the rotational speed of the engine, and at some rpm, may become equal to one of the frequencies of the free vibrations of the blades. At this point resonance vibrations set in, and these may lead to rupture of some blades because of the substantial increase in the amplitude of blade vibration and their stresses.

Blades set in the same compressor rotor always exhibit some difference in geometrical dimension (for example, a difference in the thickness of the profile) within the limits of the technological tolerances placed on their fabrication. Therefore it is possible to encounter a significant (up to 15%) scatter in frequency in the same modes of free vibrations. Because of this frequency scatter, resonance blade vibrations do not set in at a rigorously determined rotational speed, but in certain regions of the operational range of speeds, which are referred to as resonance regimes of engine operation.

Vibrational stresses arising in a blade in any particular vibrational mode are proportional to the amplitude of the vibration. The amplitudes of resonance vibrations, in turn, are functions of the magnitudes of the exciting forces and of the damping of the vibrations. Even when there are strong disturbances in the airflow, the resonance vibrations of a blade may be harmless, if their damping is sufficiently large. Resonance vibrations that are harmful to the integrity of the blading are called critical. Critical resonance vibrations should not be encountered in the range of operational engine speeds, but should lie at least 15-20% above or below the operational engine regimes.

Damping of vibrations in rotor blades takes place in the material of the blades themselves and of the attachment parts (mechanical damping) and in the airflow (aerodynamic damping).

Damping of vibrations in the material of the blade takes place by reason of the dissipation of energy in overcoming intramolecular forces in blade deformation, and damping in the attachment parts takes place by dissipation of energy in overcoming the force of friction in the key when the elements of the blade attachment to the disk rub against each other.

Aerodynamic damping is due to change in the angle of attack and relative speed of flow over the vibrating blades. When the blade moves toward the back edge of the profile (Fig. 26,b) the angle of attack and the relative speed of flow are decreased, and the aerodynamic force on the profile is reduced by minus ΔP. In the motion of the blade in the reverse direction (Fig. 26,c) the angle of attack and relative speed of flow increase, and the aerodynamic force increases by plus ΔP. Because of the change in relative flow speed and primarily because of the change in the angle of attack on the blade, in its vibration a variable force arises to act on the blade in the plane of its vibration and in a direction opposed to its motion. The greater the amplitude of the blade vibration, the greater the change in the angle of attack. Therefore on long and flexible blades, the aerodynamic damping is usually greater than on short and stiff ones. For this same reason, aerodynamic damping predominates in blade vibrations in the first bending mode, where the amplitude of the blade is relatively large. At rated and maximum rotational speeds of the engine, the aerodynamic damping exceeds the mechanical damping (by a factor of 10-15). The maximum vibrational stresses in a blade ordinarily occur in those sections of the blade, excluding the tip, where the amplitudes of the vibration are the greatest, i.e., at the locations of the loops between the nodes. As an example, Fig. 27 shows three first bending mode vibrations of a blade, and the corresponding distribution (curve) of the magnitude of vibration bending stresses along the blade. As is apparent from the figure, the maximum bending stress in vibrations of the blade in the first bending mode, arise at the root section. Fatigue failure of the blades takes place, as a rule, on the side of the thin exit edge of the fin. For thin blades of an axial compressor, there are risks also of high frequency modes of vibration, in which the maximum stresses arise mainly in the upper part of the blade fin.

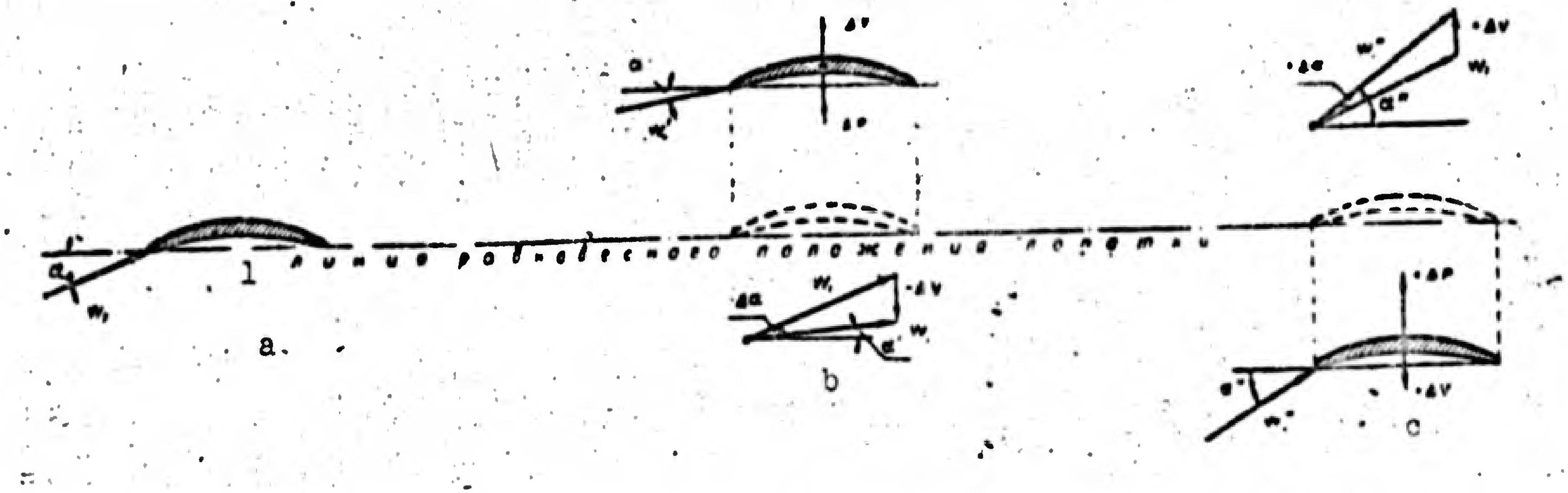

Fig. 26. Diagram explaining the origin of aerodynamic damping. a) Blade that is not vibrating; b) blade deflected toward the back side; c) blade deflected toward the concave side. 1) Blade-balance line.

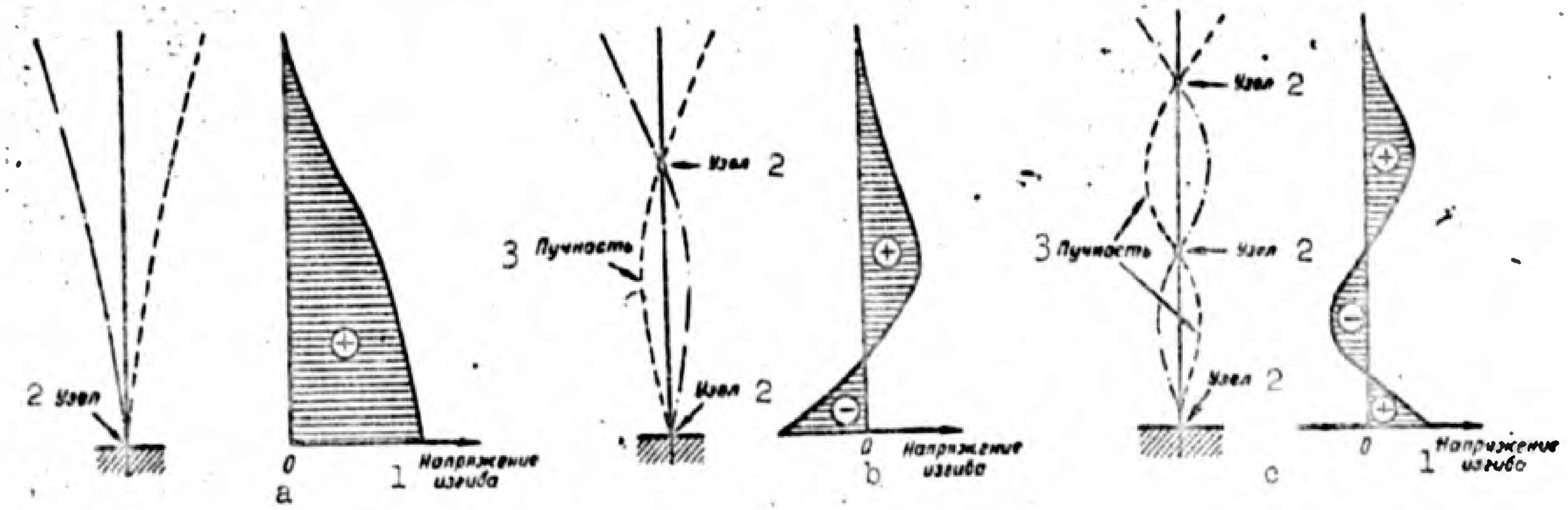

Fig. 27. Modes of bending vibration in a blade, and the corresponding vibrational stress curves. a) First; b) second, and c) third modes. 1) Bending stress; 2) node; 3) loop.

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The magnitude of permissible variable stresses in blades depends on the locations of resonance vibra'ions with respect to the rotational speeds of the engine, on the duration of operation of the engine at such regimes, and on the magnitude of the tensile stresses in the blade due to centrifugal force. If the vibrational stresses in the blades of the engine exceed the permissible limit, necessary changes in design are made, when the engine is improved, for moving harmful resonances from the operational range of speeds, or for decreasing the vibration amplitudes of the blades. Displacement of the harmful resonance rpm from the limits of the operational engine regime is effected either by change in the frequncy of the free blade vibrations, or by change in the frequency of the exciting forces that are of magnitude such as to cause harmful resonance vibrations.

The frequency of the free vibrations of a blade may be changed by appropriate alteration in the dimensions and configurations of the blade; the frequency of the exciting forces are changed by alteration in the configuration of the air passages. For this purpose, it is ordinarily considred desirable to reduce distrubances in the airflow to a minimum for example, by change in the number or position of parts projecting into the flow, and also by increasing the axial clearance between the blades of the rotor and the stator vane assembly. The amplitudes of resonance vibrations in blades very often turn out to be reduced by increasing the mechanical or aerodynamic damping. For example, friction can often be increased in the keying attachment by means of which the blade is mounted on the disk by a convenient choice of type of key or of clearance. The margin of fatigue strength of the blades may be increased by decreasing the stress concentration, for example by providing a more gradual transition from the fin to the shank of the blade, as well as by use of stronger materials or by improving their heat treatment.

6. COMPRESSOR ROTORS

In addition to the rotor blaaes on the compressor rotor, there are also other parts turning with the rotor, which are designed to hold the rotor blades last and to transmit to them the power from the turbine, required for air compression. The blades are attached to a drum 2 (Fig. 28) or to a disk 2 (Fig. 29 and 30).

Rotors in which the blades are set on a drum are called rotors of the drum type. The drum 2 (Fig. 28) has end walls 1 and 3, and journals 6, turning on bearings.

Disks of a rotor may be set on a common shaft 1 (Fig. 29), turning on bearings, or they may be connected with one another by annular sections 4 and 5 (Fig. 30) turning on bearings on a divided shaft, attached to the outermost disks of the rotor. In the first case, the rotor is of disk construction, and in the second, it is a mixed (drum-disk) design, since it includes both disk and annular parts, making up a structure similar to a drum.

Basic loads acting on the elements of a rotor include the centrifugal forces of the rotor mass and the blade masses, the weight of the rotor itself, the forces of inertia due to the maneuvers of the aircraft, twisting (torsional) and bending moments, and axial forces. A diagram of the loading of the compressor rotor is shown in Fig. 31.

The forces of intrinsic weight and of inertia, due to aircraft maneuvers, act in a plane passing through the axis of the engine, and cause bending of the rotor. Bending moments are taken up in the shaft of disk rotors, and in the rotors of the drum and mixed-construction types, by the drum and ring parts, respectively.

Since the resultant force due to intrinsic weight is applied at

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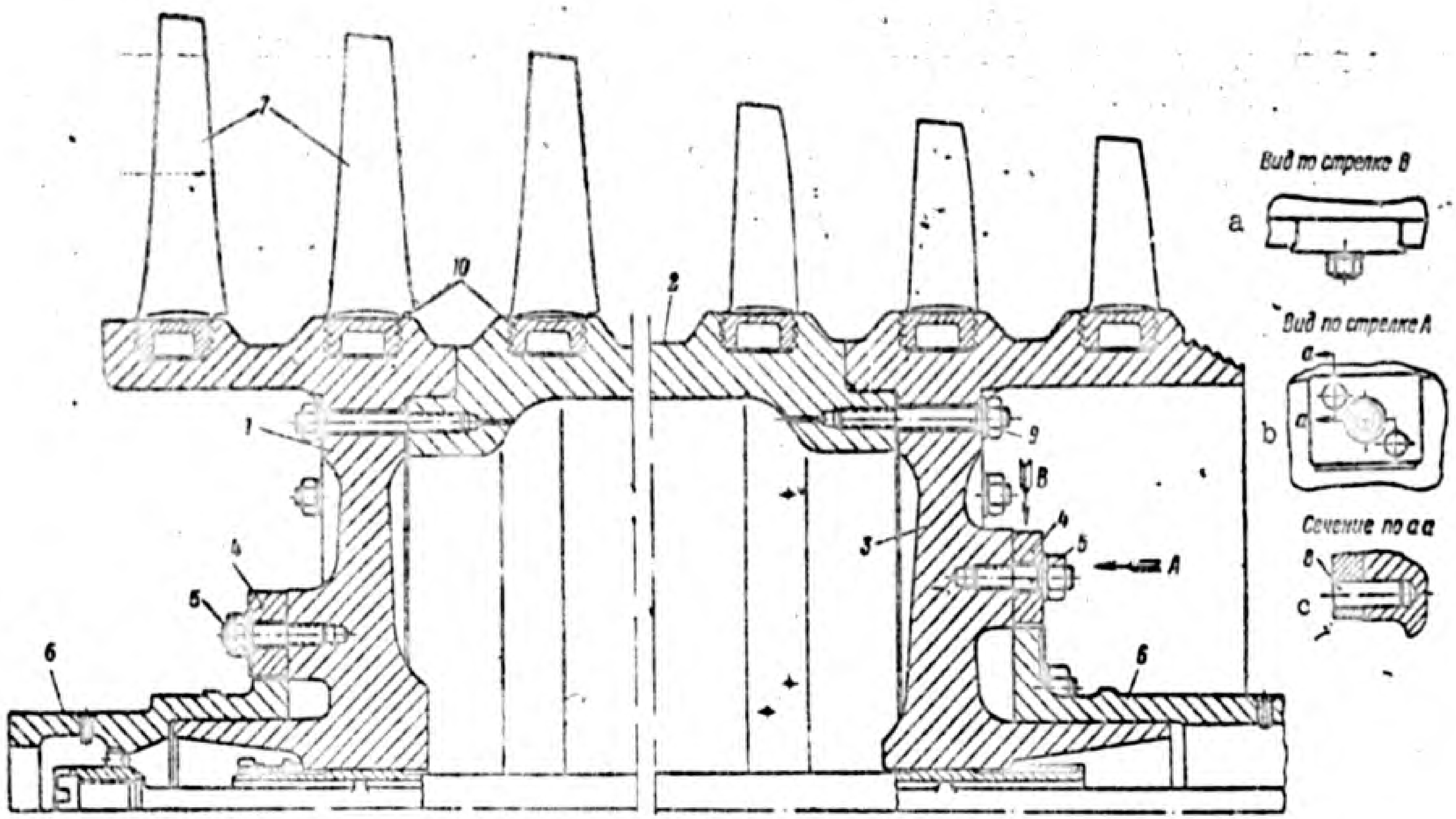

Fig. 28. Drum-type rotor. 1 and 3) End walls of the rotor; 2) drum; 4) biscuits [sukhari]; 5 and 9) studs; 6) journals; 7) blades; 8) dowel; 10) blade shank. a) View along arrow B; b) view along arrow A; c) cross section through a-a.

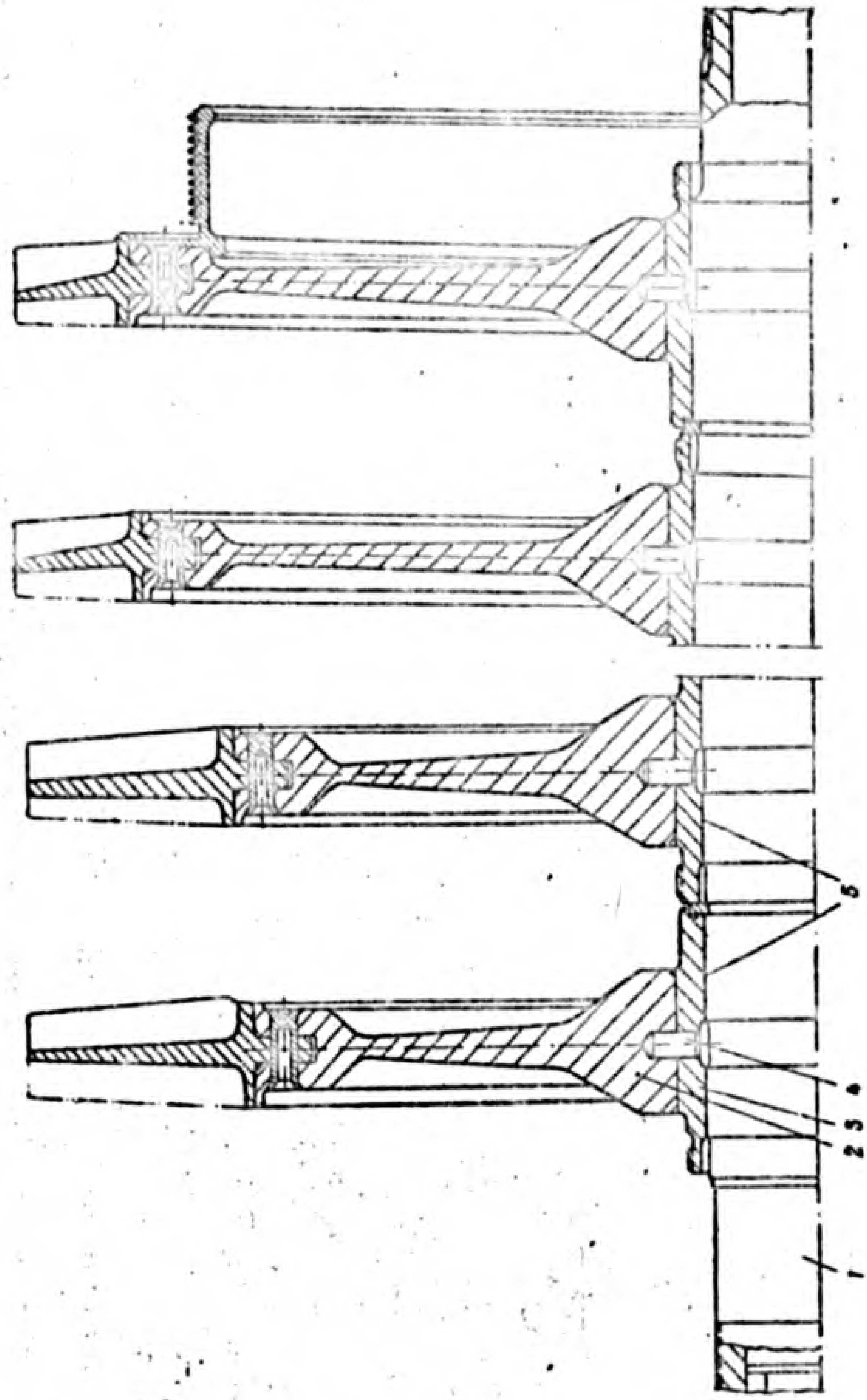

Fig. 29. Disk-type rotor. 1) Shaft; 2) disk; 3) pressed-on sleeve; 4) dowel; 5) conic collars.

the center of rotor gravity, the bending moment reaches its maximum value near the center of the span between the bearings. The

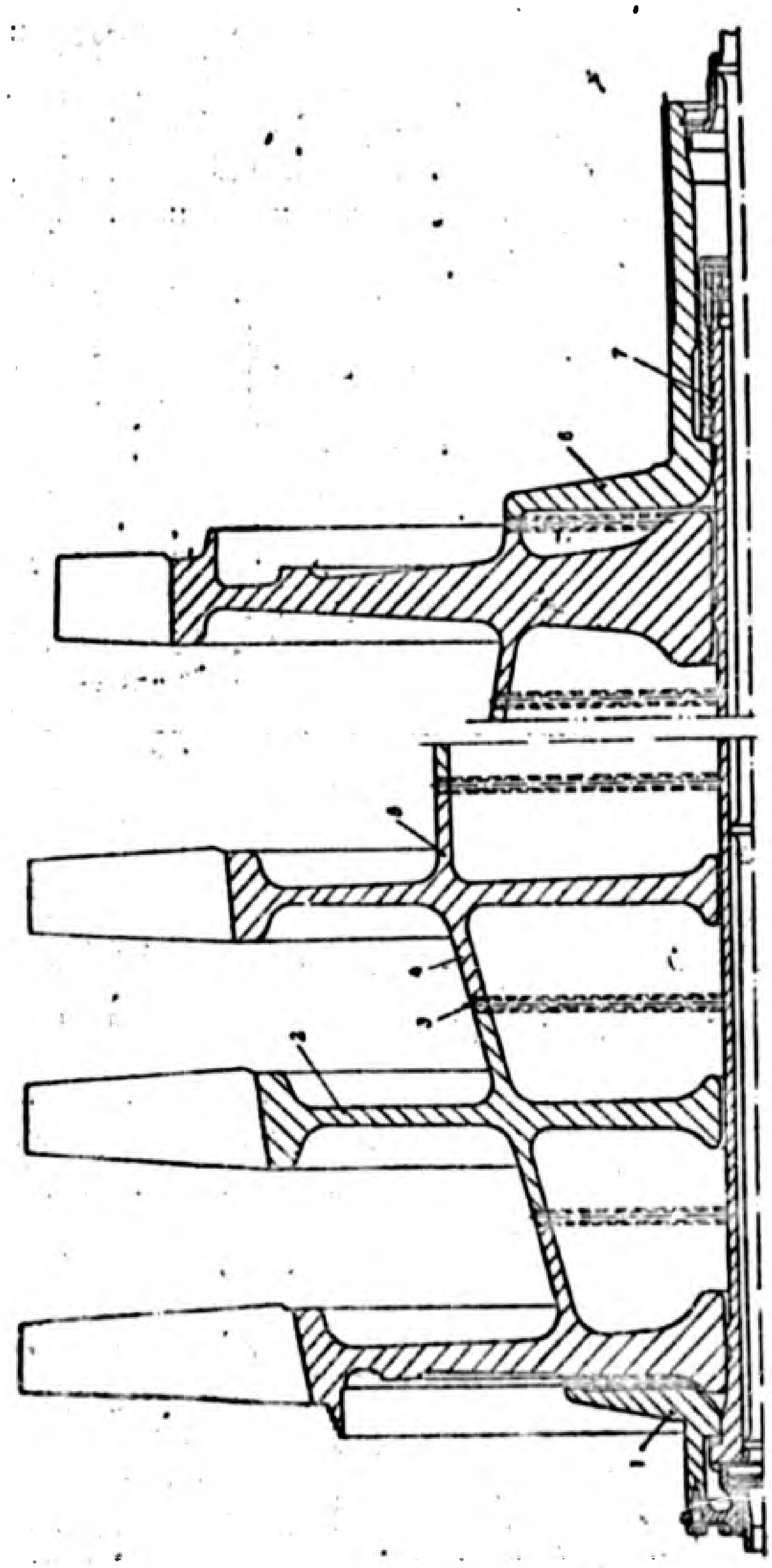

Fig. 30. Drum-disk (mixed) rotor design. 1 and 6) Shaped flanges; 2) disk; 3) end grooves; 4 and 5) annular sections; 7) tie bolt.

bending moment is decreased with shortening of the span between bearings. For this purpose, for example, in a rotor of the drum type (see Fig. 28) the first and last stages of the compressor are displaced over the end-braced parts of the drum.

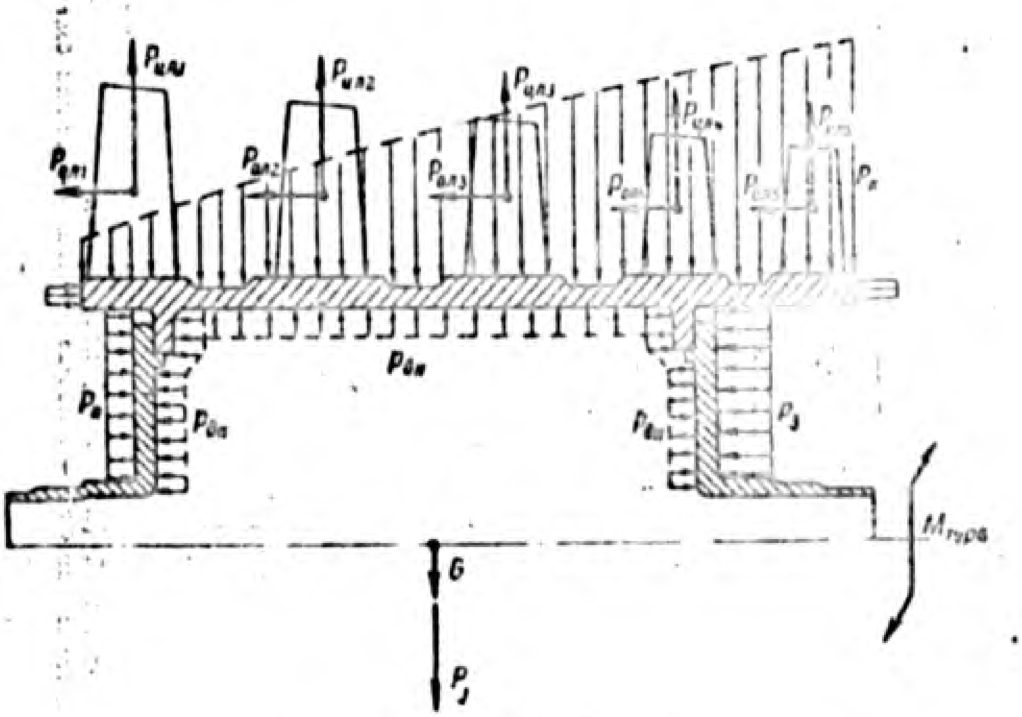

Fig. 31. Loading diagram of the compressor rotor. P_{tsl}) Centrifugal blade forces; P_{ol}) axial forces on the rotor blades; p_k) air pressure in the flow passage; p_{vn}) air pressure inside the rotor; p_p) air pressure on the front end wall; p_z) air pressure on the back end wall; G) gravity force; P_j) force of inertia; M_{turb}) turbine moment.

The resistance to deformation in bending, that is, the bending rigidity of a part, is proportinal to its diameter and the wall thickness. Therefore such rigidity in a rotor of disk construction (Figs. 29 and 32), ordinarily with an outside shaft diameter within the limits 0.1-0.3 of the disk diameter, is always less than the rigidity in bending of the rotors of drum design, and is also less than in the rotors of mixed construction with annular parts spread toward the periphery of the disks. It should be kept in mind, in this connection, that the increase in bending rigidity of annular parts distributed in this manner is accompanied by an increase in the weight of the rotor. In the rotor of mixed construction shown in

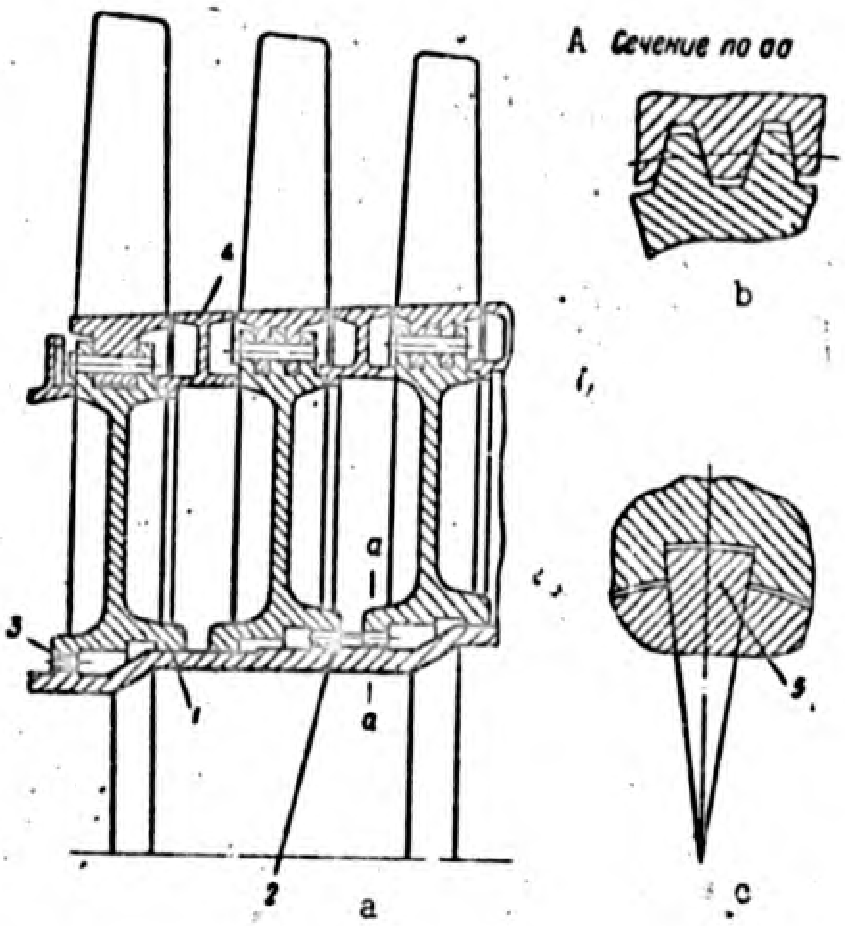

Fig. 32. Slotted joints. a) Elements of the disk rotor; b) profile of the involute slots; c) profilès of trapezoidal slots; 1) Central collars of the disks; 2 and 3) evolvent (involute) slots; 4) space; 5) trapezoidal slots. A) Cross section through a-a.

Fig. 30, the diameters of the annular parts are a maximum at the location of the maximum bending moment, and diminish at the bearings. Because of this, adequate rigidity of the rotor is attained as well as some advantage in lower weight as compared with a rotor having a uniform diameter for its annular sections (Fig. 33).

The components of the aerodynamic blade forces P_{okr} (see Fig. 18) in each stage create a torsional moment about the rotor axis, opposing the direction of rotation. This moment of resistance is overcome by a torque transmitted to the compressor from the turbine (see Fig. 31). The torque of the turbine acts in the direction of compressor-rotor rotation and is transmitted through its parts to the rotor blades. The largest torques are transmitted to the compressor-rotor parts directly connected to the turbine shaft. Since in each compressor stage a part of the moment is transmitted through the rotor blades to the air, the moment transmitted by the working parts of the rotor decrease in the direction from the last stage of the compressor toward the first.

In the transmission of torques to elements connecting the parts, circumferential forces are generated. The magnitude of a circumferential force is proportional to the moment transmitted, and inversely proportional to the number of connecting elements and their distance from the rotational axis.

In the rotor of drum construction (see Fig. 28) the torque from the rear journal 6 through the "biscuits" 4 and dowels 8, working in shear, is transmitted to the rear end wall 3, and from it to the drum by the frictional forces over the annular surfaces arising from the tightening down of the butt joint by the studs 9. The shoulders of the "biscuits" transmit the pressure from the bolts on the studs 5 to hold the journal to the back-end wall. The side surfaces of the "biscuits" can be slid along the channels of the collar of the journal, to provide free radial deformation of the back wall and flange, which are made of various materials and exhibit various thermal coefficients of expansion.

In a rotor of the disk type (see Fig. 29) the torque of shaft 1, with a conical collar 5, is transmitted by the frictional force of the pressed-on sleeve 3 against the conical inner surface, and from the sleeve to the disk through the radial dowels 4 working in shear.

The pressing of an intermediate sleeve onto the shaft, instead of pressing a disk directly onto the shaft, is a procedure used for the following reasons. In operation, when heated, and under the action

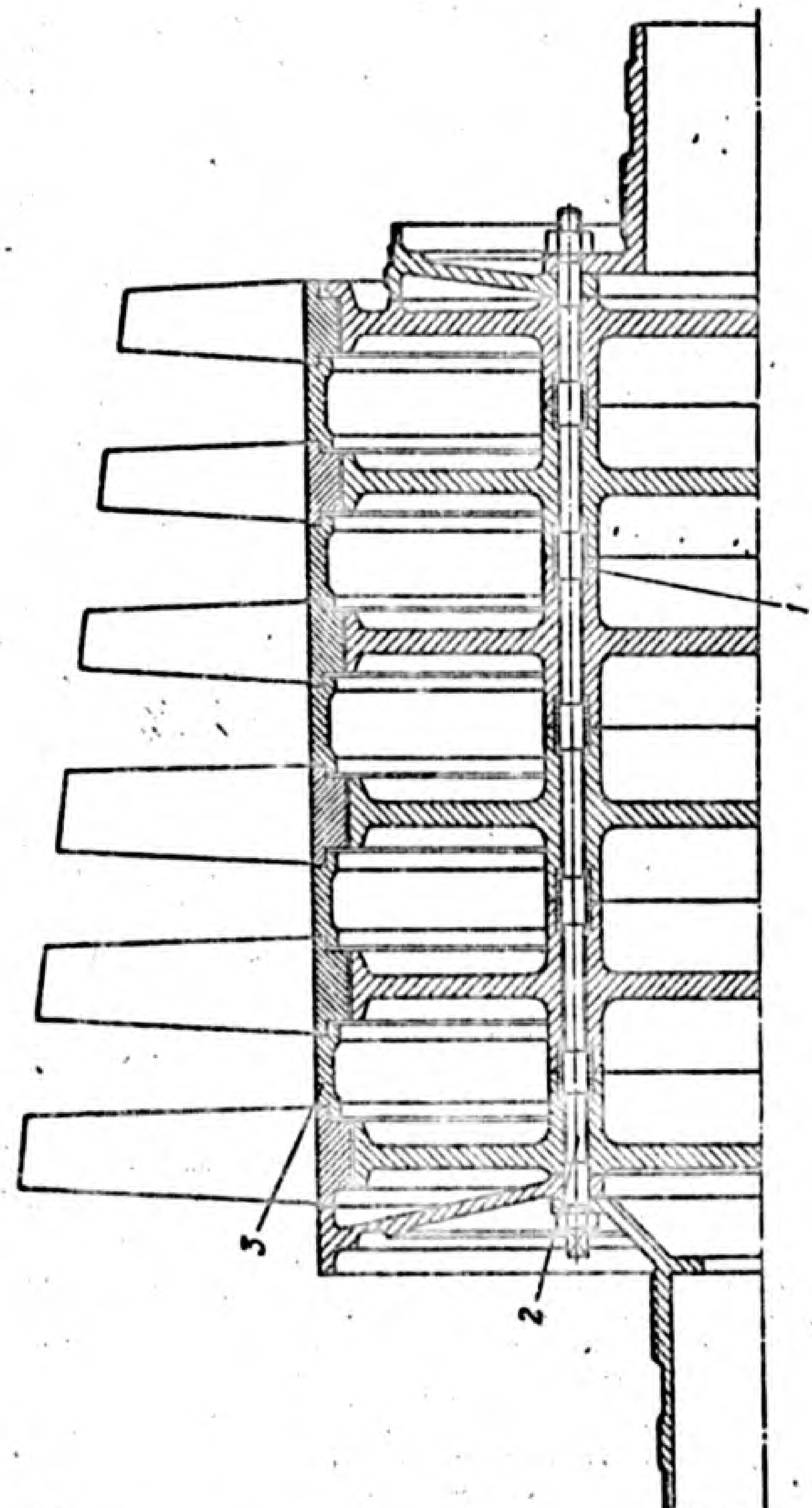

Fig. 33. Drum-disk rotor with transmission of torque through sleeves. 1) Axial dowels (sleeves); 2) tie bolt; 3) spacing rings.

of centrifugal forces, the deformation in expansion of the disk is greater than that of the shaft. Therefore, in an operating engine in

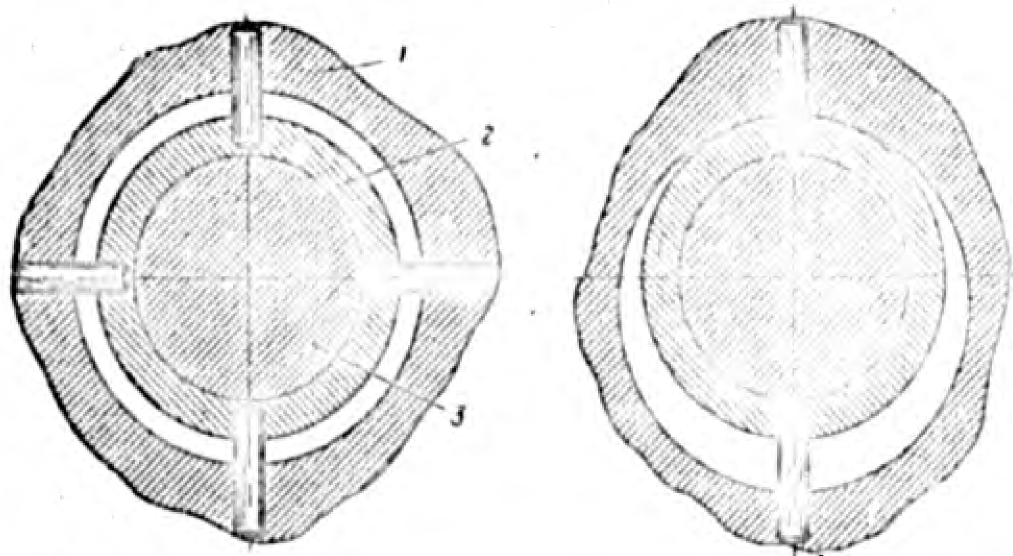

Fig. 34. Centering of disks by radial dowels. a) Four dowels; b) two dowels; 1) Disk; 2) sleeve; 3) shaft.

which the disk was pressed directly onto the shaft, there would be a decrease in the force holding the two parts together, causing a decrease in the frictional force, as a result of which the turning forces due to friction might be insufficient to transmit the torque. The sleeve 3 is stressed to an insignificant degree by the centrifugal forces of its own mass; in the operational condition the tightness on the shaft is practically unchanged. The torque from the sleeve to the disk is transmitted by radial dowels regardless of the fact that there may be a clearance between the sleeve and the disk (Fig. 34,a). The presence of radial dowels, if there are three or more of them, insure centering of the disk relative to the axis of rotation, whereas in the absence of dowels, or if there are only two of them (Fig. 34,b), if there be a clearance, the disk might be located eccentrically with respect to the rotational axis.

The action of frictional force due to the press fit permits the transmission of torque within the limit on tightness set by the strength of the sleeve.

A substantial amount of torque can be transmitted by the sides of the evolutes (see Fig. 32,a and b), the rectangular, and the trapezoidal (Fig. 32,c) slots or cogs.

The centering collars 1 of the disks (Fig. 32,a) which are located on the shaft by pressing, promote the maintenance of the centering of the disks under operational conditions. If trapezoidal slots 5 are used with stress on the side radial surfaces, then under operational conditions with thermal deformations of the disk and deformations caused by the action of centrifugal forces, the angular dimensions of the slots in the disk do not change. In this case the central position is maintained, just as in the case of the radial dowels, which may be thought of as analogous to the side walls of the trapezoidal slots.

In rotors of mixed construction, the torque from disk to disk can be transmitted by action of frictional forces, by the radial slots on the ends, by tight-fitting [template] bolts, by radial or axial dowels (sleeves) and through weld seams.

The origin of frictional forces on the end surfaces of the annular parts 2 and 3 (Fig. 35) is the tension of a central tie bolt 4. The tie bolt should be tightened enough so that the action of the bending moments in the longitudinal plane on the side of the fibers in tension should maintain pressure sufficient for torque transmission by frictional forces. Consequently there should always be some margin in the tightness of the bolt in comparison with that force just sufficient for transmission only of the torque, without taking into account the possible bending of the rotor.

The tension forces of the bolt will remain unchanged if the thermal deformation of the rotor and the bolt are the same, that is, if the heating or cooling (in comparison with the thermal conditions of the assembly) of the rotor and bolt, respectively, shorten or lengthen the same amount. In practice under operational conditions of the engine the thermal deformation of the bolt and the rotor are not the same.

Often all of the rotor, or at least part of it, is fabricated of aluminium alloy, and the bolt is made of steel. As a consequence of the difference in the coefficient of linear expansion of the rotor and bolt material, and their temperature, the lengthening of the bolt under operational conditions may be more or less than the extension of the rotor, and consequently, the force of tension may change in comparison with the force set in during assembly.

For example, when the engine is shut down in flight by burner blowout, the rotor is cooled faster than the tie bolt from which the heat flows only very slowly, so that in this case bolt tension is decreased in comparison with the tension set in during assembly. Taking the above into account, it is necessary during assembly to provide for a margin of tension force for compensation of any possible weakening.

In the operational regime it is also possible for the tension to increase. This always occurs immediately on starting the engine.. The rotor, upon starting, is heated considerably faster than the tie bolt, and therefore expands more than the latter. But the tie bolt prevents the free expansion of the rotor. Therefore, the difference in the thermal deformations of the rotor and the bolt are compensated by two simultaneous effects: some compression of the rotor, and some extension of the bolt. The resistance of the rotor to compression is several times larger than the resistance of the bolt to tension, because the cross-sectional area of the annular parts is larger than the cross section of the bolt.

Consequently, the bolt, primarily, is stretched, which may lead to its failure. The tension in the bolt may be decreased if some spring member is inserted between it and the rotor, for example, springs 1 and 5 (see Fig. 35), whose stiffness is less than the stiffness of the rotor.

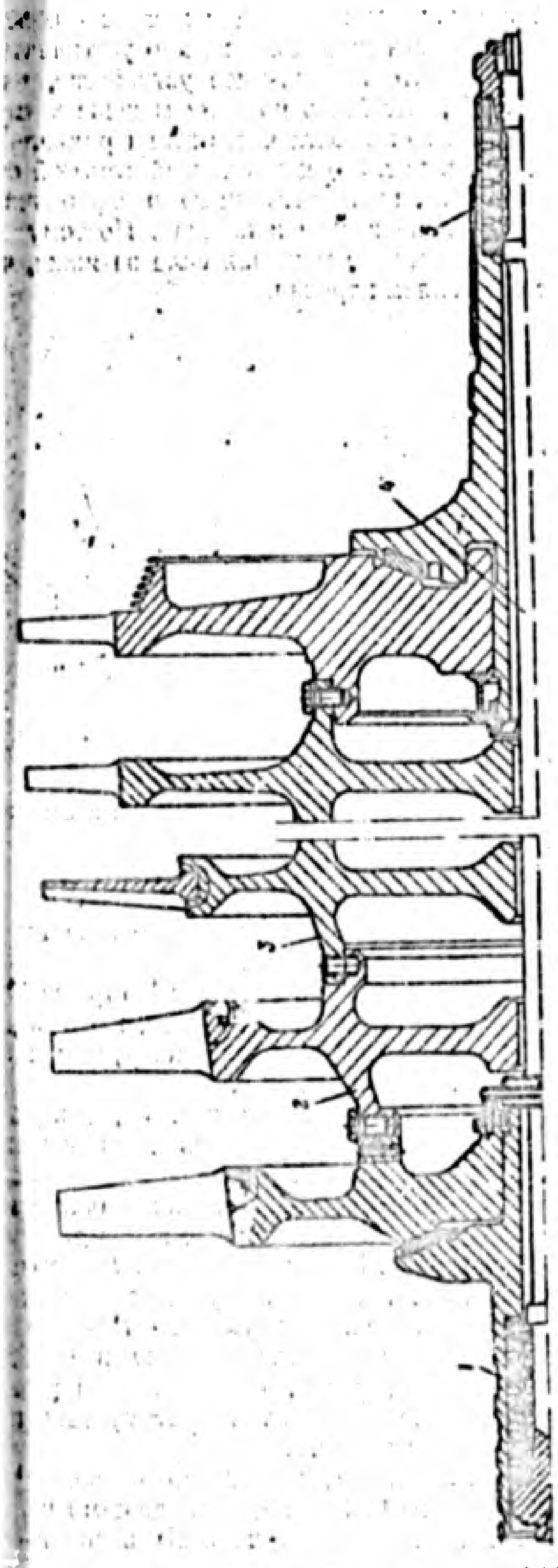

Fig. 35. Drum disk rotor with torque transmission by frictional forces. 1 and 5) Springs; 2 and 3) annular parts; 4) tie bolt.

In transmission of torque through the sides of the radial triangular end slots 3 (see Fig. 30) the central tie bolt 7 is used for the creation of compression preventing separation of the junction upon bending of the rotor and the development of clearance in the grooves. Therefore, in such designs, relatively less tension force is required than when the torque is transmitted by friction.

In case of unequal radial deformation of attached parts, their centering is retained, since the radial grooves in this connection are similar to radial dowels.

As spring elastic elements, shaped flanges 1 and 6 are used. The stiffness of these flanges is less than the stiffness of the bolt, and part of the difference in the thermal expansion of the rotor and bolt is compensated by bending the central part of the flanges in the direction of the rotor.

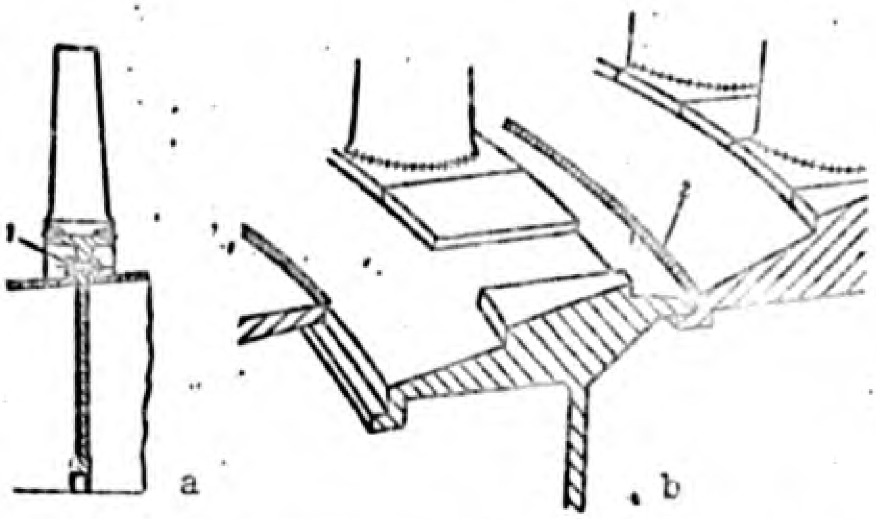

Fig. 36. Attachment of disks by template bolts and welding. a) Attachment by template bolts; b) welded rotor. 1) Bolt; 2) weld seam.

In construction of the rotor shown in Fig. 36,a, holes are drilled and reamed in the flanges and the annular parts together, and they are held tightly by bolts 1. Bolts set in such an arrangement are called tight-fitting or template bolts. Template bolts under the influence of torques from the shaft, act in shear, and under the action of bending moments, they act in tension. Little compression of the flanges by the template bolts is required, since even in the case of separation of the joint upon bending of the rotor, transmission of torque is still possible. Attachment by use of template bolts makes assembly and disassembly of the rotor easier. The relative position of parts joined by use of template bolts does not change during operation of the engine, so that no auxiliary equipment is necessary for centering.

The axial sleeves 1 (see Fig. 33) are analogous to the template bolts, working in shear. The disks are held together by bolts 2, passing through the holes in the sleeves. Separating rings 3 are located between the disk rims. The outer surface of a ring forms the inner wall of the air passage of the compressor, between disks.

The radial dowels 10 (see Fig. 17,c) act in shear when the rotor is subject to rotational or bending moments. The dowels are placed tightly in holes, drilled and reamed together, on the rims of the disks and in the annular parts, in grooves under the blade keys. Upon being thrown outward under the action of centrifugal forces the dowels are retained by the blade keys. A disadvantage of attachment by radial dowels is complexity in assembly and disassembly of the rotor.

Welded one-piece rotors of mixed construction (Fig. 36,b) may be resorted to in those cases in which the parts of the rotor are fabricated of steel, and the annular parts are located at the periphery of the disks, where less strength is required of the weld seam for transmission of torques and bending moments.

Fabrication of drums and disks, is done by machining of forge blanks. The circular grooves under the keys of the blades are drilled. The grooves under the fir-tree keys are shaped by broaching. Longitudinal grooves of trapezoidal shape are made by the same method or by slotting.

For disks and drums at air temperatures in the flow passages below 150-200° C aluminum forgings made of heat-resistant alloys are used. Steel is used for fabrication of disks of the first stages loaded with large blade centrifugal forces, and for disks of the later stages, operating at air temperatures above 200-250° C.

7. STRESSES IN DRUMS AND DISKS

Drums and disks are loaded primarily by centrifugal forces due to the intrinsic mass, and the mass of the blades. Figure 37 shows elements of the drum with blades set in circumferential and longitudinal grooves. Under the action of the centrifugal forces of the intrinsic mass and the mass of the blades, the drum, like a ring, tends to rupture at points where tensile stresses are generated.

Since the longitudinally placed grooves are cross-cut in the drum, such a drum is without question weaker than one with circumferential grooves. Therefore in designs of compressors, rotors of the drum type are generally used with circumferential grooves.

Stresses in the drum are greater, the greater the centrifugal forces of the blades, and the greater the circumferential speed of the drum. Even in the absence of blades, i.e., when loaded by the centrifugal forces of the intrinsic mass alone, the maximum circumferential speed permitted by the strength limitation of the drum material amounts to less than 200-250 m/sec, which is insufficient for modern compressors.

Disk rotors have greater strength than drum rotors. This can be shown in the following manner.

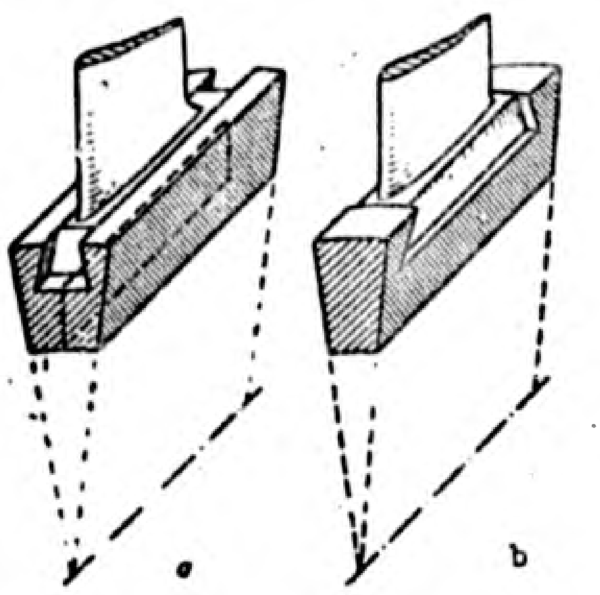

Fig. 37. Elements of a drum. a) With longitudinal grooves; b) with a transverse groove (dashed line shows the minimum cross section of the drum).

Let us imagine a disk consisting of separate unconnected rings, all turning with the same angular speed (Fig. 38,a). As the outer ring we take the rim 1 of the disk, that is, its thickened part, in which the keys of the blades are set. In this case the rim is like a drum, loaded by the centrifugal forces of the blades and the intrinsic mass, and having sufficient strength for some limiting circumferential speed. Each of the rings 2, 3, 4, etc., taken separately, also may be considered as similar to a drum, loaded only with the centrifugal force acting on its intrinsic mass. The rings lying nearest the center have less circumferential speed, and their loading is less

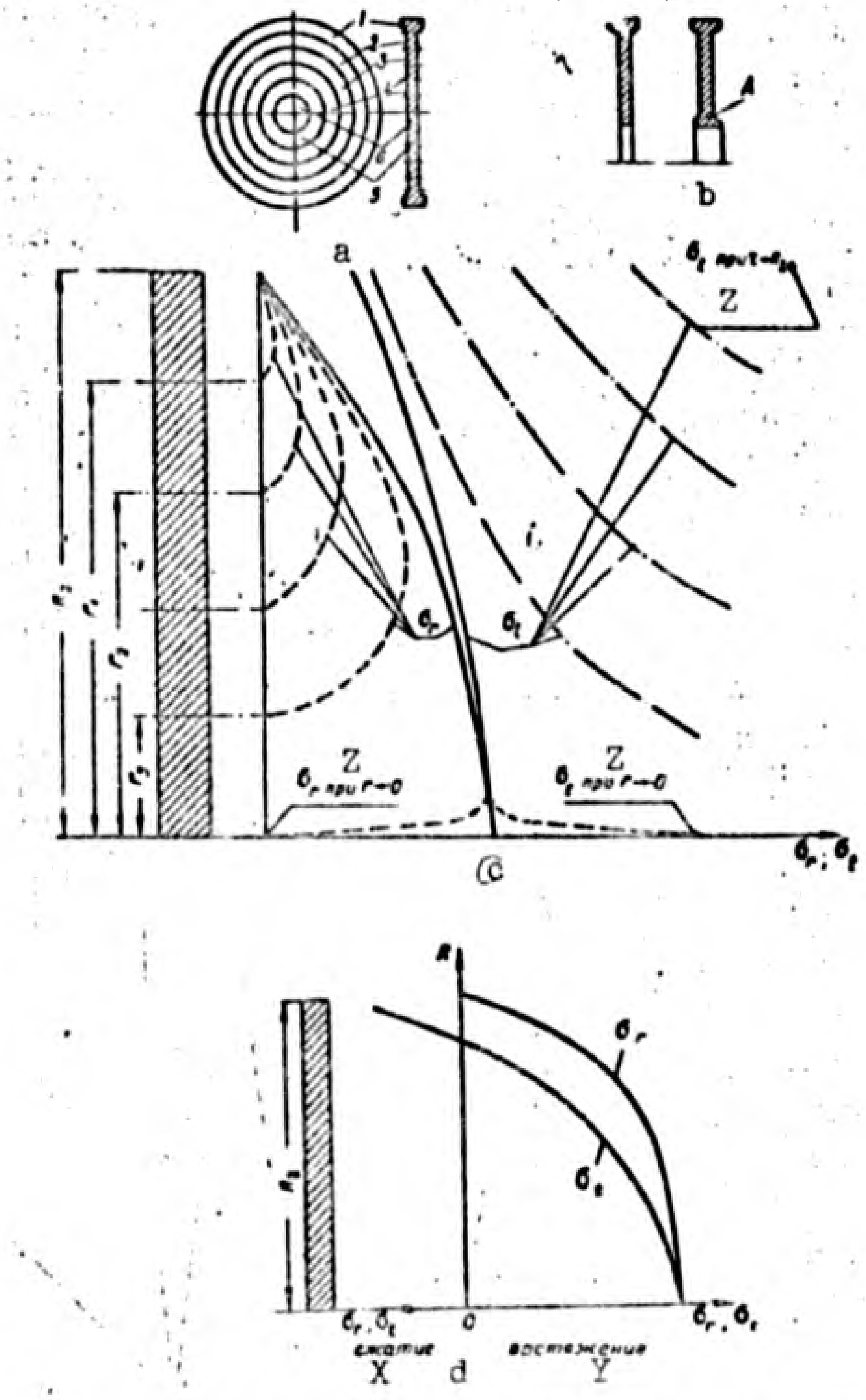

Fig. 38. On the problem of the loading of disks. a) Solid disk; b) disk with central hole; c) stresses acting in a flat rotating disk without blades; d) stresses acting in a flat rotating nonuniformly heated disk without blades (solid lines show stresses in a disk without a hole, and the dashed lines show the disk with a hole). A) Hub with collar; R_2) external radius of the disks; r) radius of hold; σ_t) circumferential stresses; σ_r) radial stresses; 1, 2, 3, 4, 5, and 6) rings. Z) for; X) compression; Y) tension.

than that at the periphery. Therefore they have a margin of strength and may assume an additional radial load, for example, transmitted from the rings of larger diameter.

At the rim and at that part of the disk located next to it, given a circumferential speed greater than that permissible for the strength of a drum with the same loading, in a real disk the loading of the elements lying nearer the center would be increased by a radial loading due to the parts near the periphery, and the disk would not fail. Therefore it follows that in addition to the circumferential tensile stresses in a disk, tensile stresses also exist in the radial direction (Fig. 38,d).

A disk with a central hole under conditions otherwise the same, will be weaker than a disk that is continuous (Fig. 38, b and c). For compensation of the weakening of the disk due to the presence of the central hole, in such disks there is ordinarily a thickened hub A around the hole.

The disk may be heated nonuniformly in the radial direction. Near the rim of the disk, the temperature is always higher than in the center, because the rim is heated by the air flow in the flow section; over the remaining surfaces of the disk, some loss of heat may take place to the relatively cool air which flushes it, or to the parts of the rotor that are heated less strongly for example, the shaft or the annular parts of the rotor.

The nonuniform heating of the disk leads to the situation where its peripheral parts tend to expand more than those lying closer to the center; this is readily represented by the model of the disk separated into rings. As a result of this, in the circumferential direction at large radii, compression stresses may develop (Fig. 38,d), since the connection of these parts of the disk with the central parts impedes their free expansion. On the other hand, in the central part of the disk, under the influence of the expansion of the peripheral part of the disk, circumferential tensile stresses exist. In the radial direction, nonuniform heating leads to the appearance of tensile stresses.

In disks without holes, the maximum stresses in the center may amount to 4500-6000 kg/cm^2 for steel disks and to 1500-2500 kg/cm^2 for aluminium disks. Steel disks with holes have stresses in the limits 6000-7000 kg/cm^2, and aluminium disks of the same type, 1500-3000 kg/cm^2.

8. VIBRATIONS OF DISKS

Varying forces acting on the blades from the airflow are transmitted to the disk and induce forced vibrations. Forced vibrations of disks may also be caused by bending vibrations of the compressor rotor. In both cases, the frequency of the forced vibrations of the disks is a multiple of the engine rpm. Whenever the frequency of the forced vibration of any disk coincides with one of the frequencies of free oscillation, resonance vibrations set in, and these often serve as a source of fatigue failure in the disks.

The modes of free vibration of disks are quite diverse. From them all, it is possible to distinguish the following especially characteristic modes:

a) vibrations about one or several <u>nodal diameters</u> (Fig. 39);

b) vibrations about one or several <u>nodal circumferences</u> (Fig. 40);

c) simultaneous vibrations about both <u>diametral and circumferential nodal lines</u> (Fig. 41). This mode of vibration consists of a combination of the first two.

Vibrations about nodal circumferences are excited by variable axial forces, and in gas-turbine engines are encountered very rarely. Most commonly observed are the vibrations about nodal diameters, and combined vibrations.

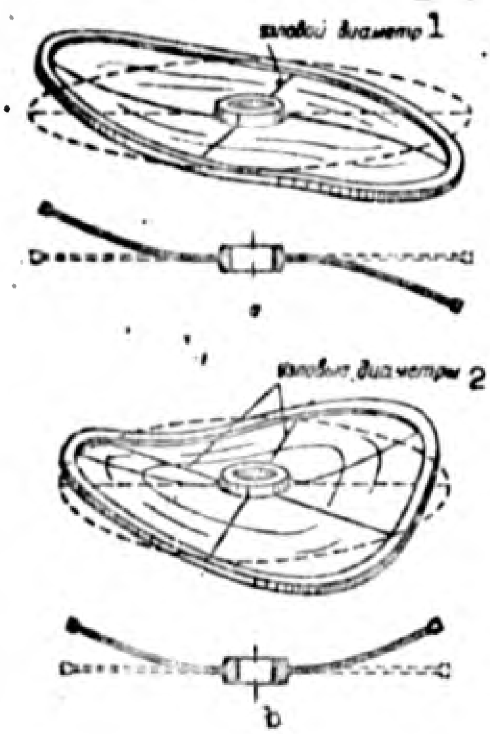

Fig. 39. Modes of vibration of a disk having a central hub, about nodal diameters. a) With one nodal diameter; b) with two nodal diameters. 1) Nodal diameter; 2) nodal diameters.

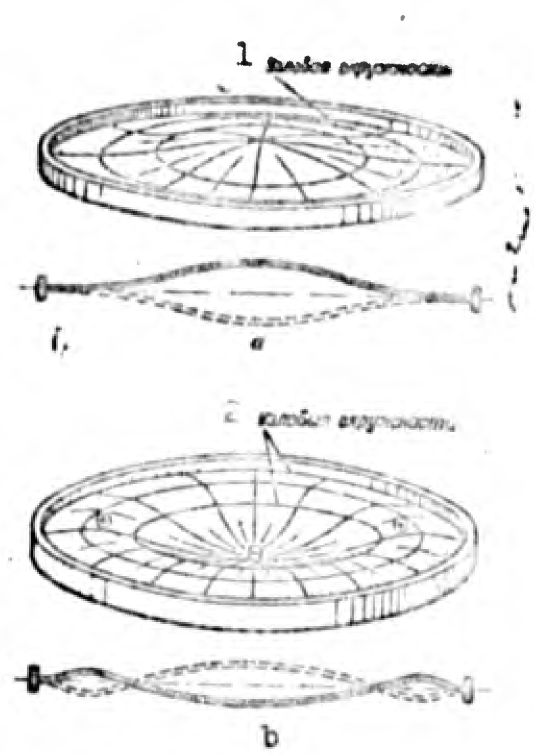

Fig. 40. Modes of vibration of a disk fixed at the rim, about nodal circumferences. a) With one nodal circumference; b) with two nodal circumferences. 1) Nodal circumference; 2) nodal circumferences.

Vibrations of a rotating disk are different from those of a stationary disk in that vibrations of a rotating disk have nodal diameters and loop diameters which are not motionless with respect to the disk, but rotate in this or that direction. This shifting of the deformation wave with respect to the disk gives us the term running waves. If the nodal diameters are turning relative to the disk at a rate equal to the rotational speed of the disk, but in the opposite direction, then the nodal diameter would be motionless in space. Conditions exciting such vibrations of the disk are forces of nonuniform airflow of which the velocity and pressure field are motionless. Resonance vibrations of disks in such a case are most harmful, since they are induced not at random, but by steady

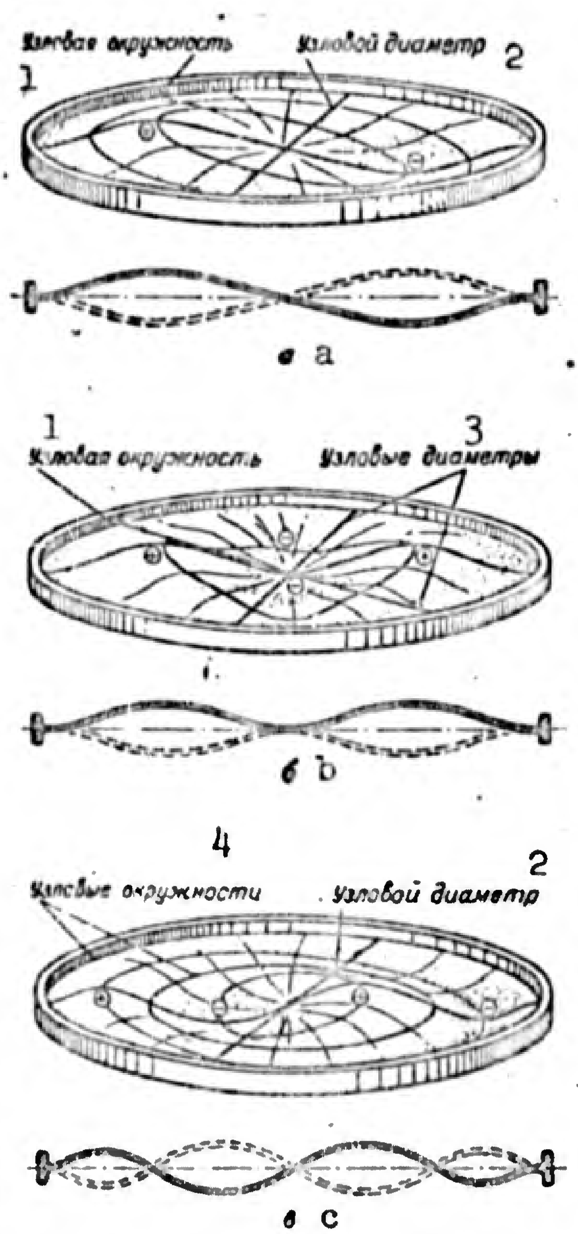

Fig. 41. Vibrations of a disk fixed at the rim, about nodal circumferences and nodal diameters. a) With one circumference and one diameter; b) with one circumference and two diameters; c) with two circumferences and one diameter. 1) Nodal circumference; 2) nodal diameter; 3) nodal diameters; 4) nodal circumferences.

disturbances, for example disturbances due to parts projecting into the airflow.

9. COMPRESSOR HOUSINGS. STATOR- AND GUIDE-VANE ASSEMBLIES

The compressor casing serves for support of the stator vane assembly and guide-vane assembly (or flies), and is one of the basic members in the stress system of the engine. The engine mounts on the aircraft, are located outside the casing as are the engine auxiliary systems, the fuel, lubrication, air and other conduits, as well as electrical wiring and other equipment.

The compressor-casing loading diagram, showing forces and moments, is presented in Fig. 42.

Aerodynamic forces are exerted on the blades of the stator vane assembly and guide-vane assembly (or flies) as is the case with rotor blades. The axial components of the aerodynamic forces are transmitted to the compressor casing, loading it in the axial direction. The circumferential components load the casing with torques.

Axial forces and torques from other parts of the stress system attached to the engine are also transmitted to the compressor casing.

Through the points at which the engine is mounted on the aircraft, located on the compressor casing the resultant of all the axial forces acting on the parts of the force system is transmitted, and this resultant is equal to the thrust of the engine.

The compressor casing is subject to bending moments from the forces of intrinsic weight, and from radial forces transmitted to the casing from the rotor bearings.

Because of the difference in air pressure between the engine flow passage and the atmosphere, the compressor casing is subject to forces normal to its surface.

The primary requirement imposed on the casing is adequate bending and torsional stiffness, with small weight.

Cast or welded casings are used; these may be split, or unsplit (Fig. 43).

Unsplit casings are used in combination with split compressor rotors; fabrication of the unsplit casing introduces difficulties of a technological nature..

Split casings are ordinarily split in one plane or in two mutually perpendicular planes. The presence of two planes of separation simplifies both the assembly and the fabrication of a cast large-diameter casing. The wall thickness of a cast casing is 6-10 mm, and the wall thickness of the rear part of the casing should be greater than that of the front, since larger pressure increments and bending moments act on the rear section. Because of the difference in temperature between the air at the front and the back parts of the casing, they may be built of different materials; for example, the front part may be made from magnesium alloy, and the rear from heat-resistant aluminum alloy. Magnesium alloys are used at air temperatures below 100-150° C in the airflow passage, and heat-resistant aluminum alloys at temperatures below 200-250° C. At higher temperatures, steel casings are used..

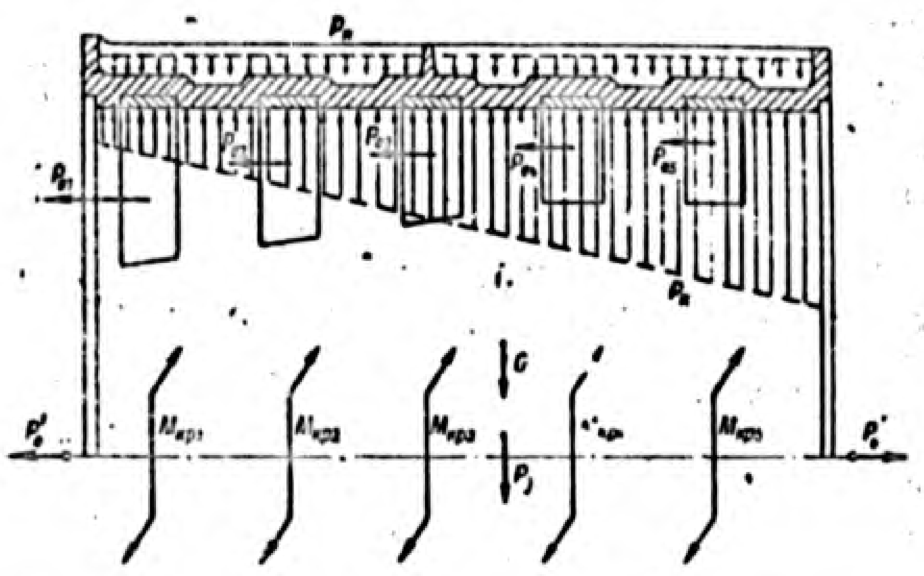

Fig. 42. Loading diagram of a compressor casing. G) Force of gravity; P_j) force of inertia; P_o') axial forces transmitted to the casing from attached parts; M_{kr}) moments originating in the blades of the stator vane assembly; P_{o1}) axial blade forces in the stator vane assembly; p_n) pressure of the outside air; p_k) pressure of air in the flow passage of the casing.

Steel welded casings are technologically simpler to produce than cast ones. The welded casing may be made of sheet materials without subsequent machining (Fig. 43,b). For the purpose of imparting great stiffness to a casing, rings with a T-profile are welded to it on the inside.

The radial dimensions of the inner surface of the casing (Fig. 44) are chosen to maintain suitable clearance between the blade tips and the casing under all conditions of operation. In casing designs the clearance amounts to 0.2-0.7% of the radius R_k of the casing, when the rotor is at rest. The relatively large clearance is explained by its reduction in operation because of the difference in radial deformation of the casing and rotor parts. The dimensions of the casing change because of heating and the force of air pressure, whereas the dimensions of the rotor change because of heating and centrifugal force. The resultant effect of these deformations may lead, under certain operational conditions of the engine, to a decrease in clearance by a factor of about two.

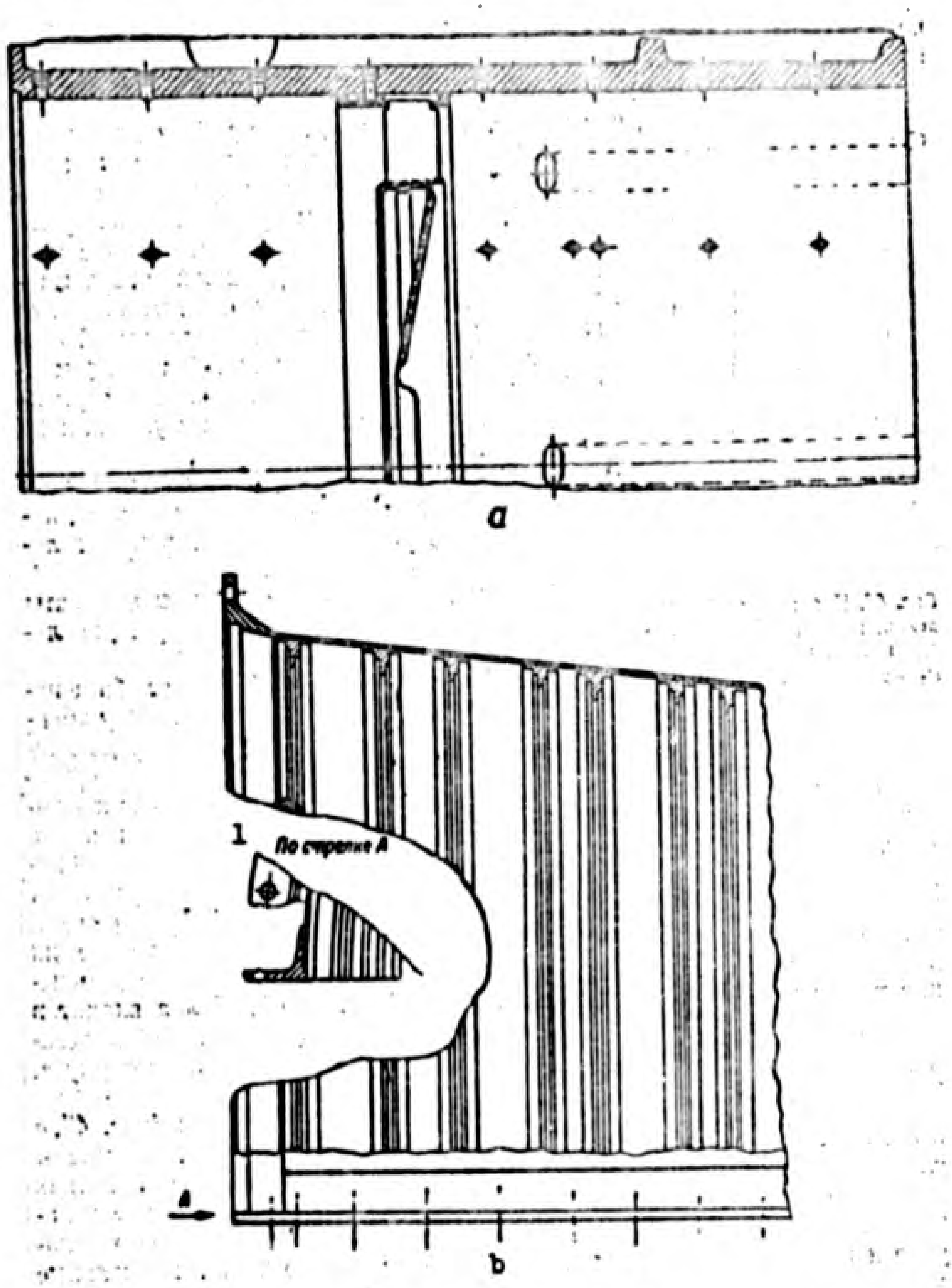

Fig. 43. Compressor casings. a) Cast unsplit casing; b) welded split casing. 1) Along arrow A.

The remainder of the radial clearance is provided to prevent contact of blades with the casing as a result of random or unpredictable radial deformations. These deformations may arise from bending of the rotor and casing, buckling of the casing, residual extension of the blades, etc.

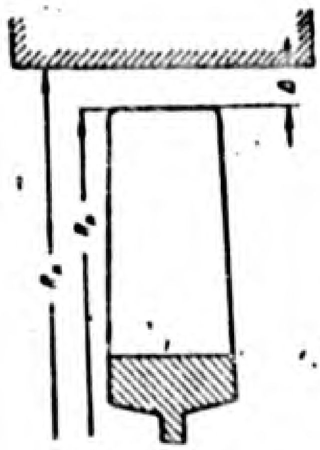

Fig. 44. On the problem of fixing the clearance between blade and casing. R_k) Radius of the casing; R_1) radius of the blade tips; Δ) radial clearance.

The presence of larger clearance than necessary impairs the operation of the engine. For the purpose of decreasing the clearance, soft coatings 12 (Fig. 45,e) are sometimes applied on the inner surface of the casing. With this arrangement, when the compressor is in operation, if there is contact between the blade tips and the casing, the soft coating wears away and thereby generates its own required clearance. A drawback of coatings is their low strength at high air temperatures. They have, therefore, limited application in the higher compressor stages.

The blades of the guide- and stator-vane assemblies are shaped much like the rotor blades.

Stator vane assembly and guide-vane assembly (or flies) may have fixed and moving blades.

The fixed blades may be fixed either directly to the casing, or to a peripheral shroud ring which is fixed in turn to the casing.

For direct attachment to the casing, the blade has a keying part. When the key is trapezoidal (see Fig. 45,a) or T-shaped (see Fig. 45,b), set into a circumferential slot in the rim, radial and axial attachment of the blades is accomplished by the walls of the slot. In the circumferential direction the keys of all, or of several blades of a row, are fixed relative to the casing by radial dowels or braces, and the key parts are in direct contact with each other. They also use a key with flange 5 and journal 4 (Fig. 45,e). The flange is placed in a circumferential slot and tightened down

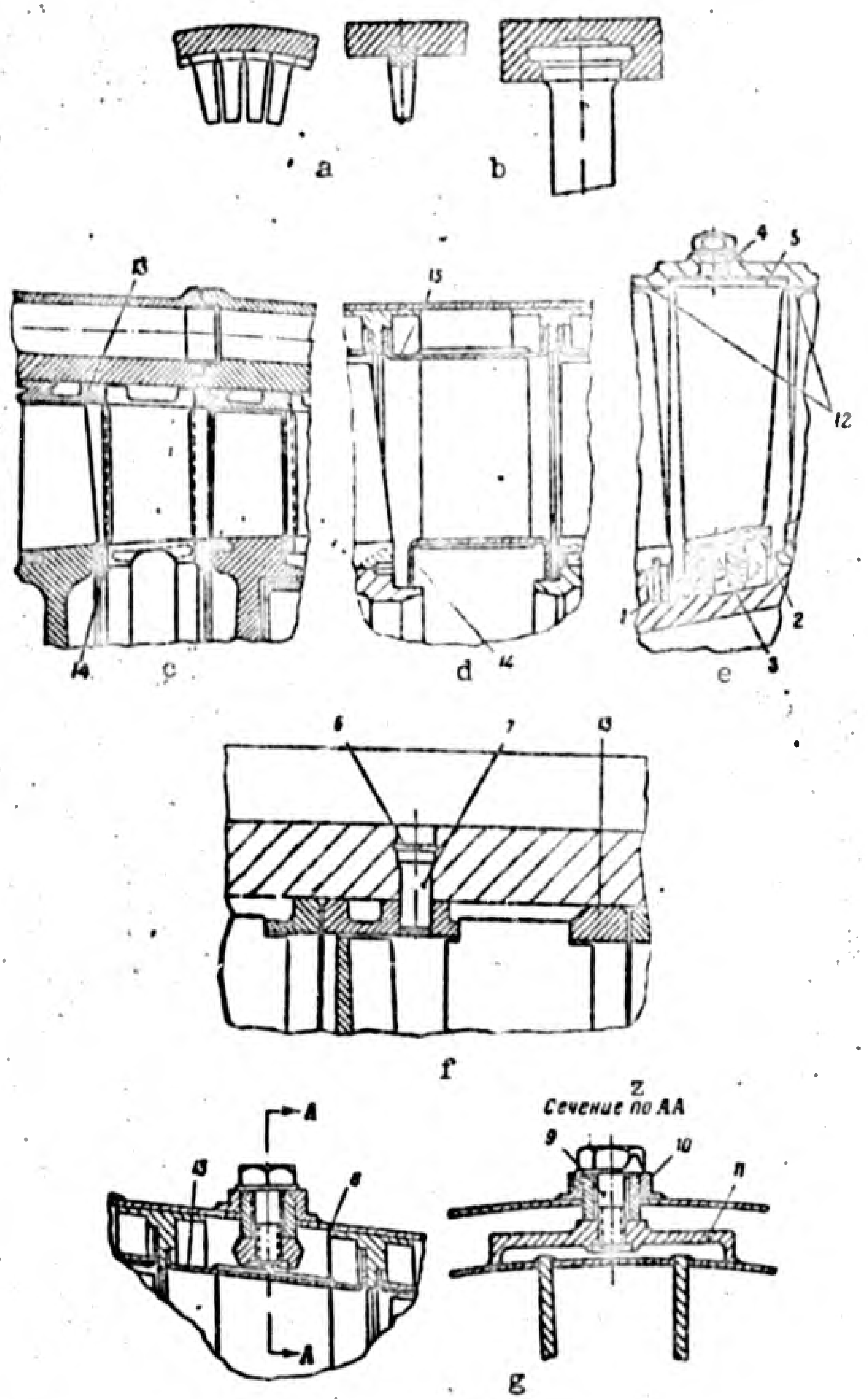

Fig. 45. Attachment of the blades and shrouds of the stator vane assembly. a) Trapezoidal key; b) T-shaped key; c and d) fixing of the blades inside shrouds; e) blade with journal and flange; f) diagram of attachment of the band to the casing by a dowel; g) diagram of shroud attachment to the casing by screw. 1 and 2) Half-rings; 3) journal; 4) threaded journal; 5) flanges; 6) spring ring; 7) dowels; 8) casing; 9) screw;

10) washer; 11) boss; 12) soft coating; 13 and 14) shroud rings. z) Cross-section along AA.

to the casing by a nut screwed to the threaded part of the journal. This latter form of key does not require reinforcement of the casing, since the thickness of the flange is relatively small.

The inner end of the blade may be either free (Fig. 45,a and b), or braced to an inner shroud (Fig. 45,c). In the first case, the blades are referred to as cantilevered. With cantilevered blades it is possible for the air to flow in a circumferential direction from the interblade channels through the clearance between the tips of the blade of the stator-vane assembly and the rotor surface, forming a flow passage. Cantilevered blades have little bending siffness, and are subject to vibration. For increase of bending stiffness and vibration resistance, and also to reduce bleedoff inside the casing, an inner shroud is mounted inside the casing.

As a rule, blades attached to the casing through external shrouds have inner shrouds as well. In case of different thermal deformation of parts of the stator vane assembly freedom of movement is provided for the blades in one of the shrouds. Thus, for example, in the design shown in Fig. 45,e, the cylindrical journal 3 of the blade can move in sockets within the shroud formed by two half-rings 1 and 2, held together by end-screws. Likewise, free deformation is provided for blades mounted in slots in the inner shroud (Fig. 45,c) when the blade is firmly fixed in the outer shroud.

When the construction material of the blade shrouds and the casing are the same, or when their thermal deformations are the same, it is possible to fix the blade tightly in both shrouds (Fig. 45,d).

The outer shrouds are joined to the casing for transmission of axial forces and torques. Radial dowels 7 (Fig. 45,f) provde simul-

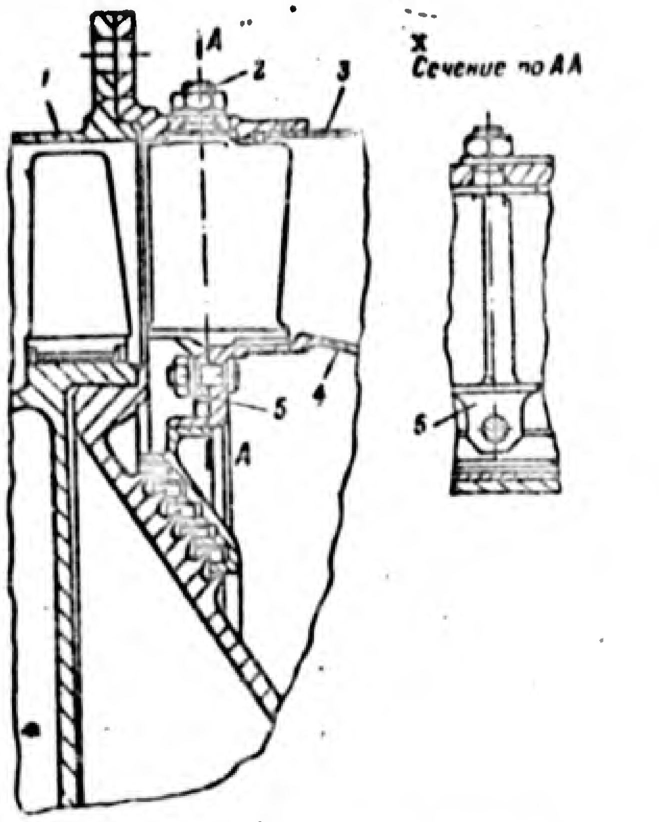

Fig. 46. Stator vane assembly of the last compressor stage. 1) Compressor casing; 2) journal; 3) outer casing of the combustion chamber; 4) inner casing of the combustion chamber; 5) lug. x) Section through A.A.

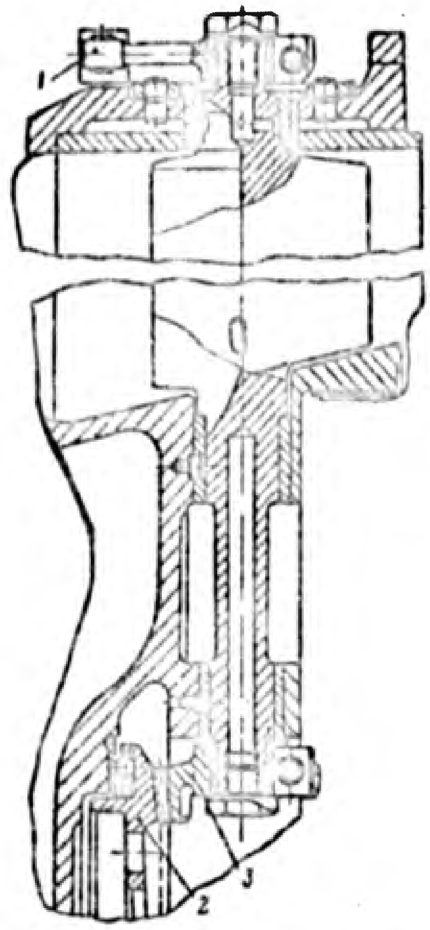

Fig. 47. Blade of rotating guide-vane assembly. 1) Driving lever; 2) gear; 3) toothed sector.

taneous axial and circumferential location of the outer shroud (such a method is used for large thicknesses of the casing and shroud). In order to prevent their falling out, the spring ring 6 holds the dowels. A thin-walled shroud with the blades welded to it may be fixed to the casing 8 (Fig. 45,g) by several bosses 11 welded to the shroud. Between the boss and the head of the screw 9 a calibrated washer 10 is set to prevent possible deformation of the shroud 8 when the screw is tightened.

The stator vane assembly may be unshrouded or shrouded. Shrouded stator vane assemblies are ordinarily used in connection with unsplit casings. In such case, the rotor of the compressor must be of split design, for example, with a tie bolt and end slots or in combination with template bolts. When the rotor is not split, a split casing and shroud rings for the stator vane assembly are used.

Blades of the stator vane assembly of the last compressor stage of some engines (Fig. 46) are used for transmission of force from the inner casing 4 to the outer casing 3 of the combustion chamber and thence to the compressor casing 1 for which are provided firm attachment of the blades to the casing, for example, by use of journals 2 and lugs 5.

Rotating blades of stator vane assembly and guide-vane assembly (or flies) are equipped with a provision for simultaneously turning all the blades through a given angle. For this purpose, the blades have pivot journals on the inner and outer ends projecting through bearings (Fig. 47).

In the design indicated, the pivot journal of the inner end of the blade has a toothed sector 3 (Fig. 47). The toothed sectors of all the blades are connected through gear 2, and the outer pivot journal of one blade (the guiding blade) has a lever 1; when this blade is turned, it causes all the blades to turn.

Turing the blades to a given angle for a desired operational condition of the engine may be accomplished by an automatic regulator connected with the lever 1.

10. EQUIPMENT FOR BLEEDING AIR FROM THE COMPRESSOR

In high-pressure compressors operating at rotational speeds below the design speed, there is a possibility of an unstable regime (surging). This is explained as follows. The cross-sectional area of the compressor is selected from among the conditions of operation, in the rated regime. Upon decrease in the rotational speed, changes are produced not only in the air flow rate, but also in its density

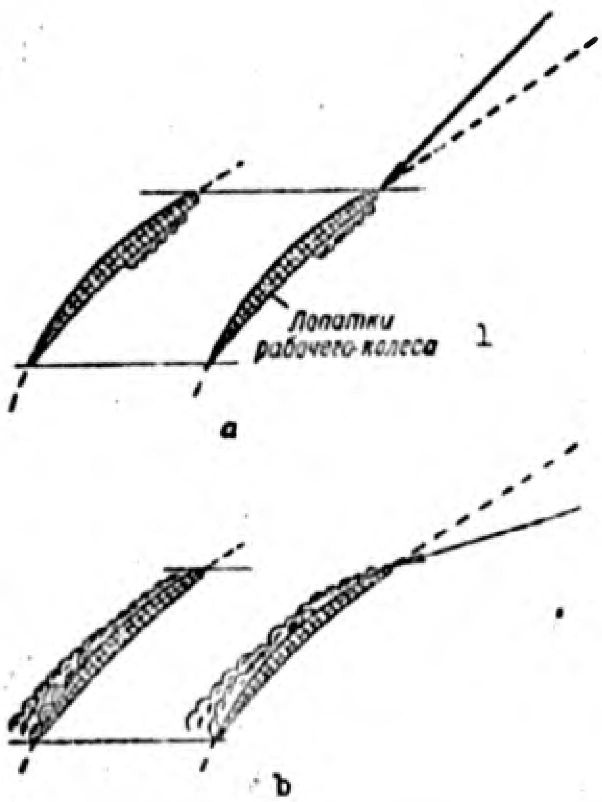

Fig. 48. Flow around the blades of compressor stages under unrated conditions. a) First stages; b) last stages (dashed line indicates the flow direction in the design [rated] regime). 1) Blade of rotor.

due to change in the pressure rise ratio. Thus in the first stage, for example, the change in air density is small, whereas in the last stage, it is considerable, because it is proportional to the change in the pressure rise ratio. As a consequence, the flow cross section of the first stages are seen to be larger, and that of the later stages smaller, than would be required in conformity with the change in air flow rate and density. As a result, the design [rated] conditions for flow around the compressor blades no longer exist (Fig. 48). In particular, in the first stages an increase in the angle of attack is produced, and the flow separates from the blades, initiating surging. In the later stages, on the other hand, the angle of attack is reduced, and the pressure rise and efficiency of the compressor are sharply reduced.

Besides the method indicated above for eliminating the unstable regimes of operation when the number of revolutions is decreased, namely, by rotating the blades of the stator vane assembly and guide-vane assembly (or flies) to an angle corresponding to the design flow conditions, a simpler method as far as structural details are concerned is also used, i.e., the bleeding of air from the compressor.

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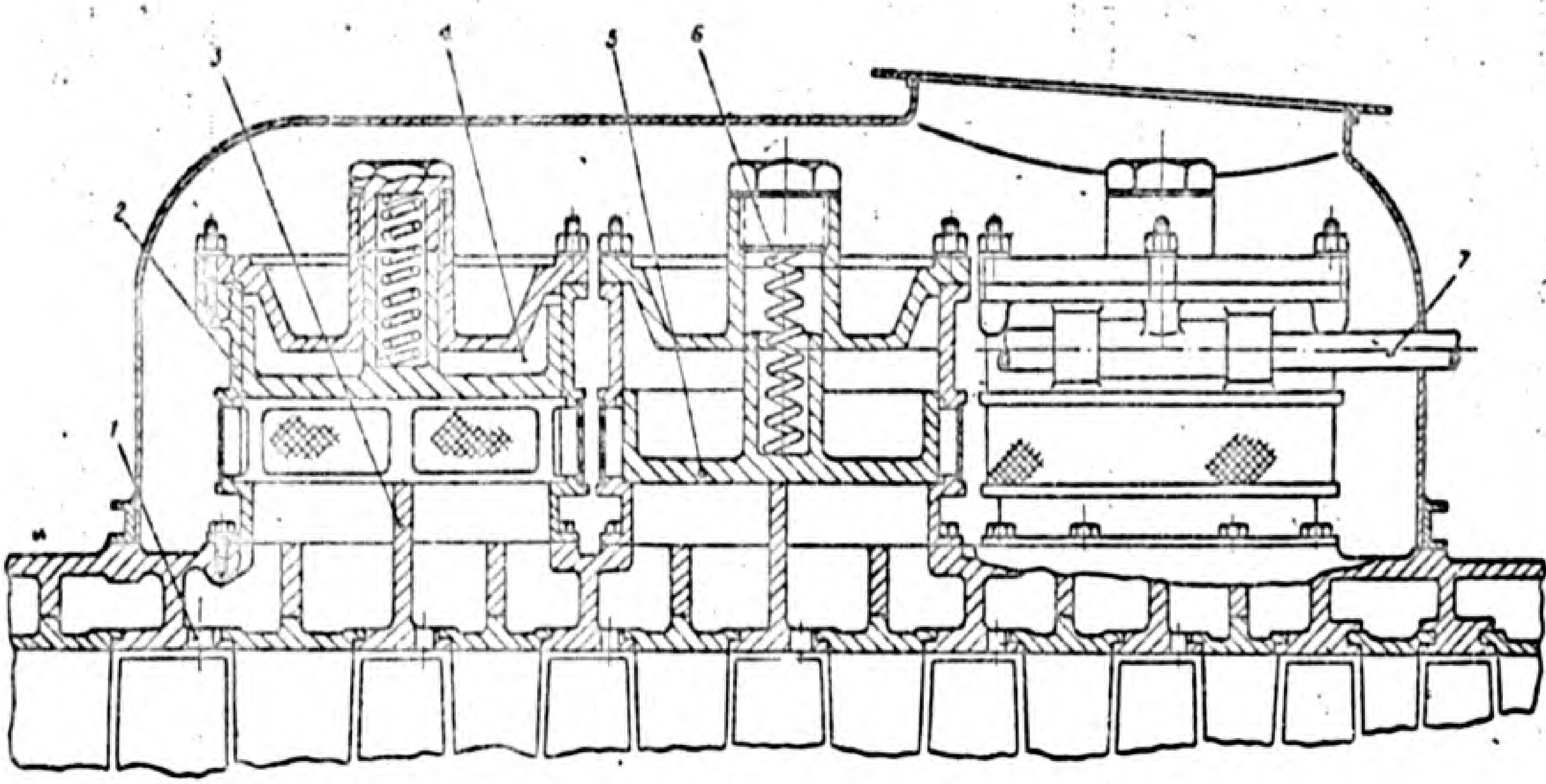

Fig. 49. Valves for air bleeding. 1) Aperture in the compressor casing; 2) valve housing; 3) fin of the compressor casing separating the bleed apertures of neighboring stages when the valve is closed; 4) recess for admission of control air; 5) valves; 6) springs for closing the valves when the engine is shut down; 7) feed line for control air.

For this purpose, apertures are put in the compressor casing in one or several rows, and these are closed by a bleed band or valves (Fig. 49); at low rotational speeds, these orifices are opened.

Bleedoff of air causes an increase in the axial speed in stages located upstream of the bleed point, and a decrease in the axial speed in the downstream stages. In this fashion the operation of the blades in all the stages of the compressor is maintained at angles of attack close to those at the design point.

Control of the aperture closure (by bleed band or valves) is automatic. When bleeding is carried out from several stages by successive closing of apertures, it is possible to regulate the quantity of air handled in conformity with the operational regimes of the compressor.

11. AIR SEALING. RELIEF OF THE COMPRESSOR ROTOR FROM AXIAL FORCES

Parts that move with respect to each other, have clearances which may separate cavities at different air pressures; if so, it is necessary to install air seals. With respect to the method of construction, seals are classified as contact seals or noncontact seals. Noncontact seals may be slit or labyrinth seals.

A contact seal (Fig. 50,a) consists of a ring holder 3, a bushing 2, and split spring rings 1, located in grooves in the ring holder. The ring holder is set into the moving part in such fashion that the centrifugal force on the rings presses them toward the bushing. The rings are lapped to the bushing, and there is practically no space between them, which accomplishes the separation of the cavities on either side of the seal.

Contact seals are used in cases where the rotational speeds do not exceed 100-120 m/sec; mostly they are used for separating cavities, in one of which oil is present. If the pressure in this case is greater in the cavity with oil than in the one with air, the seal prevents leakage of oil; conversely, when the pressure is greater in the air cavity, leakage of air into the oil cavity is prevented.

Noncontact seals are used without any limitation on the relative rotational speeds of the surfaces between which sealing occurs.

A slit seal (Fig. 50,b) is designed on the principle of creating a very narrow slit 4 between sealing surfaces. In this case no complete isolation of the separated cavities is accomplished, and some flow occurs between them, at a rate that is proportional to the pressure between them, and also proportional to the slit flow cross section. Slit seals are used, as a rule, for separating cavities in which the pressure difference between them is small, for example, in stator vane assemblies of the low pressure stages of the compressor.

Labyrinth seals are used when large differences in pressure exist between the sealed cavities. Examples of the use of labyrinth seals are shown in Fig. 45,c and 46.

A labyrinth seal (Fig. 50,c and d) consists of a series of crests 5, separated by chambers 6. Between the crests and the bushing narrow ring-shaped slits are formed. The labyrinth seal, therefore, like the slit seal, does not provide complete isolation of the separated cavities. However, in a labyrinth seal, the leakage is less than through a slit seal, a result that is accomplished by the repeated throttling of the air flowing through channels with sharply changing flow cross section. The more crests present in the seal, the smaller the leakage and the larger the chambers.

In a number of designs, the bushings of the labyrinths have soft coatings. In that case, the clearance between the crests and the bushing can be less.

Labyrinth seals are extensively used in equipment for relieving

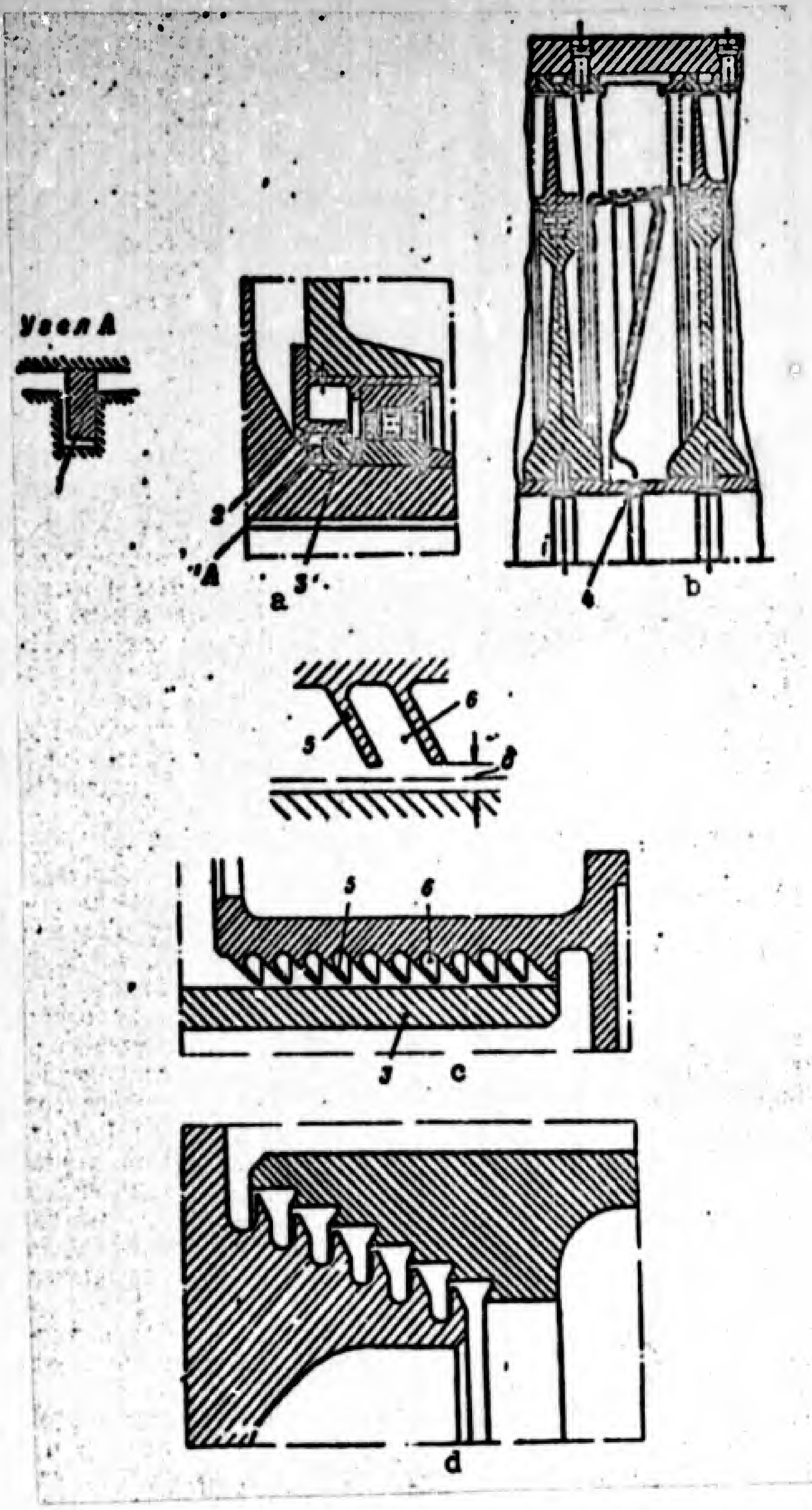

Fig. 50. Air seals. a) Contact; b) slit; c and d) labyrinth. 1) Split spring ring; 2) bushing; 3) ring holder; 4) annular slit; 5) crests; 6) chambers.

the rotor from the action of axial forces.

The compressor rotor is subject to axial forces acting on the

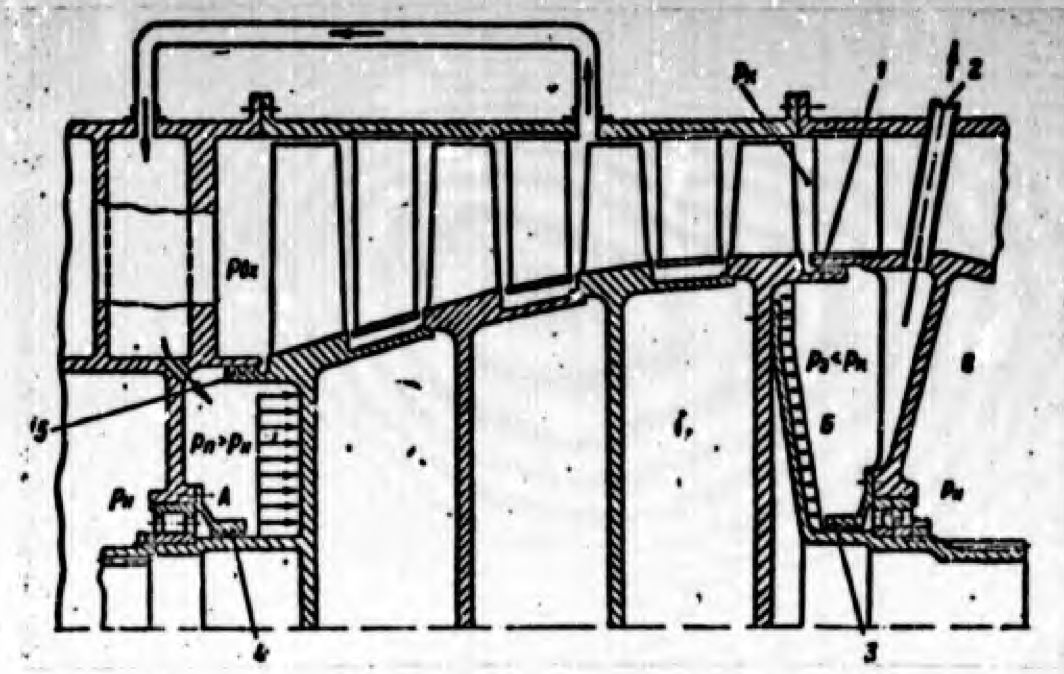

Fig. 51. For relief of the compressor rotor from axial forces. 1, 3, 4, and 5) Labyrinth seals; 2) discharge. A, B, and C) Cavities. p_{vkh}) Pressure of the entering air; p_n) atmospheric pressure; p_p) pressure of the air on the forward wall; p_z) pressure of the air on the rear wall; p_k) pressure behind the compressor.

rotor blades and to air pressure on the front wall (see Fig. 31), as a result of which a total axial force exists in a direction opposite to the airflow direction. Ordinarily the rotors of the compressor and turbine are connected so that their axial forces, oppositely directed, tend to counterbalance each other.

Since the axial force on the turbine rotor is not equal to the axial force on the compressor rotor, there remains a resultant axial force which may turn out to be greater than the maximum permitted for the thrust bearing. In such cases, the compressor rotor can be relieved from the axial forces. For this purpose, it is possible, for example, to isolate by labyrinth seals 1 and 5, the forward end cavity A (Fig. 51) and the rear end cavity B from the airflow part of the compressor and set up such pressures in those cavities as may decrease the axial force acting on the elements of the parts subject to the air flowing through the device. It is clear that one must have a pressure in the forward cavity that is greater than that in the rear cavity. Air pressure is supplied to the forward cavity from the compressor or from some intermediate stage. The rear cavity is connected to the atmosphere through a discharge 2, in which the cross section of the discharge is chosen so that the pressure in the cavity B, fed by leakage of air from the compressor through the labyrinth seal 1, and relieved by flow through the discharge, will be above atmospheric pressure by 0.6 to 0.9 kg/cm^2. This is necessary in order to prevent oil from flowing into the cavity B from the cavity C.

Chapter 2

CENTRIFUGAL COMPRESSORS

1. PRINCIPLES OF OPERATION AND BASIC PARAMETERS OF THE CENTRIFUGAL COMPRESSOR

Centrifugal compressors in current use have single inlet (Fig. 52) or double inlet (Fig. 53) impellers. The basic parts of a centrifugal compressor are: inlet section 2; impeller 7; housing 3; diffuser 5; and outlet connection 9.

Air from the entrance section enters channels formed by the impeller vanes. When the impeller turns, the air between the vanes is whirled and thrown outward toward the periphery by centrifugal force. The resulting vacuum at the entrance to the impeller draws air from the inlet section.

The vanes of the impeller, turned by the turbine, transmit energy to the air that flows in. In this manner the pressure and speed of the air flowing into the impeller channels are increased.

From the impeller, the air flows first into the diffuser slot 6 (Fig. 52), consisting of a gradually expanding flow channel, and then into the vaned diffuser in which vanes are mounted in order to make the airflow more orderly and reduce the lateral dimensions.

In the diffuser, because the air is slowed down, a portion of its kinetic energy is transformed into potential energy, i.e., the air pressure is increased.

The air is further slowed down the outlet connections of the compressor through which it enters the combustion chamber, and its pres-

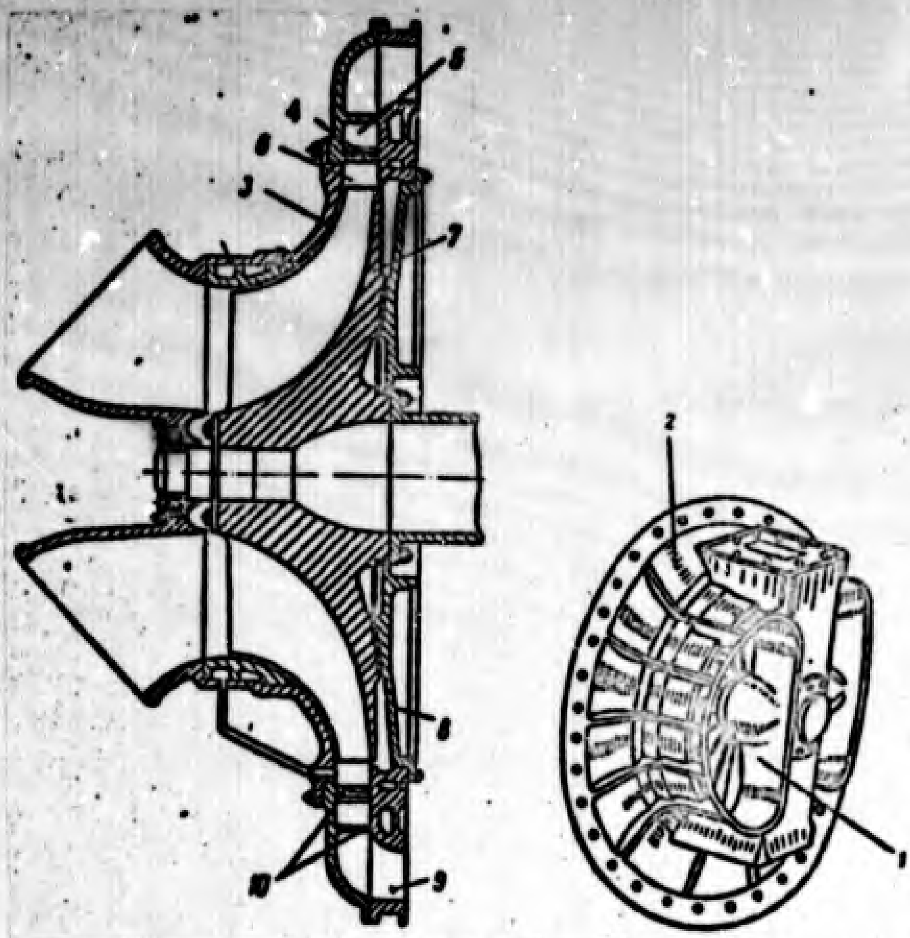

Fig. 52. Centrifugal compressor with single-side air inlet: 1) Air intake; 2) inlet section; 3) housing; 4) bolt; 5) diffuser; 6) slot diffuser; 7) impeller; 8) rear wall; 9) outlet connection; 10) diffuser vanes.

sure is thereby increased.

The pressure rise ratio of the air is the ratio of pressure at the outlet of the compressor to the pressure at the inlet; it is determined basically by the amount of energy transmitted to the air in the impeller and expended in creating a vortex within it.

Vorticity of the air in the impeller is greater in proportion to its circumferential speed. Because of inertia, the air is separated from the impeller as the latter turns. Greater vorticity may be generated than is usual in an impeller with radial vanes, for a given rotational speed of the impeller, by the so-called active impeller which has vanes curving forward in the direction of rotation.

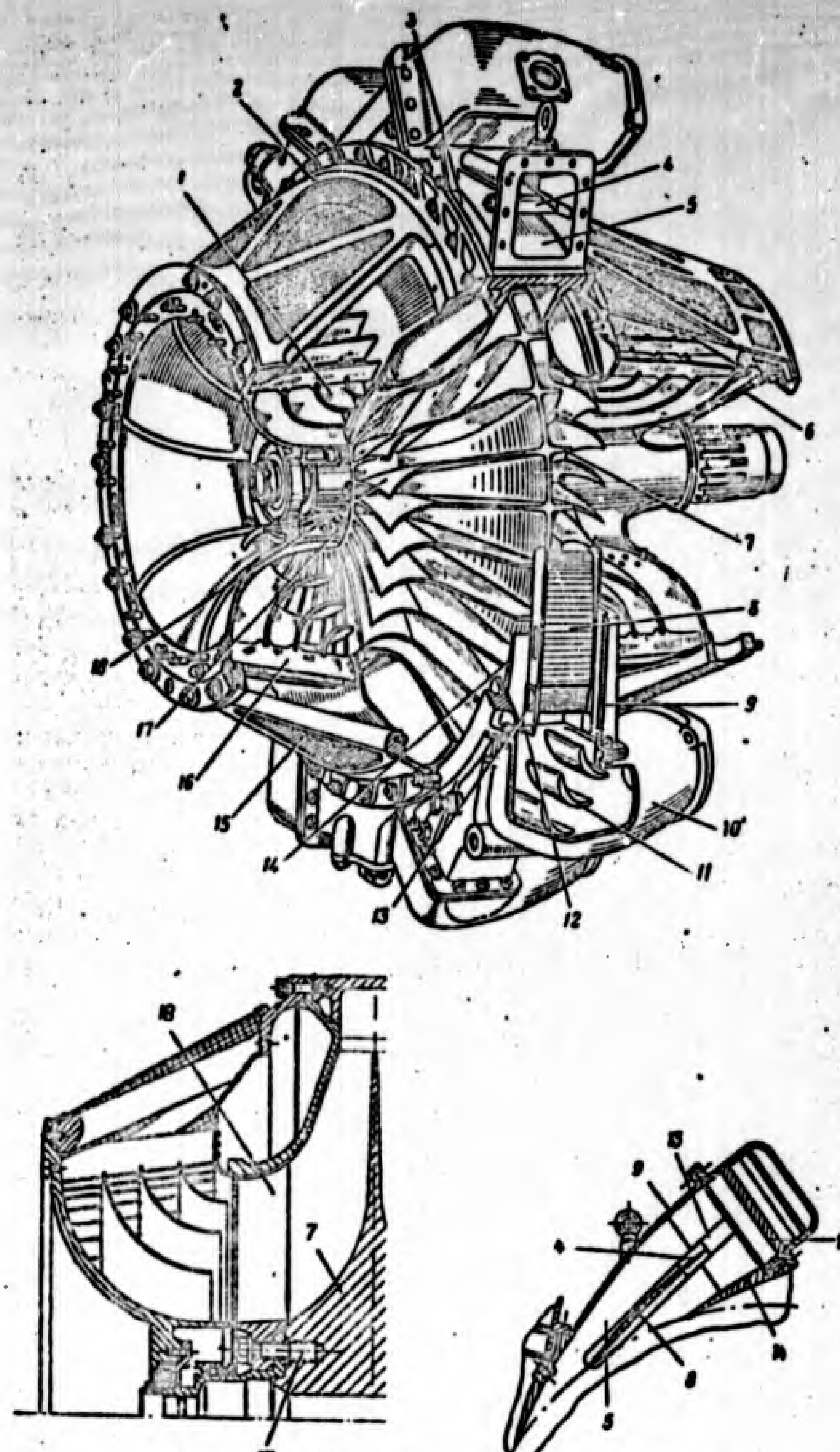

Fig. 53. Centrifugal compressor with double-side air inlet: 1) Diaphragm; 2) inlet section; 3) casing; 4) rear part of a blade; 5) diffuser; 6) shank; 7) impeller; 8) forward part of a vane; 9 and 14) wall of the compressor housing outlet; 10) outlet connection; 11) vanes; 12) screw; 13) biscuit; 15) protective screen; 16) blade of the guide-vane assembly stator; 17) pin; 18) rotating guide (VNA).

The maximum rotational speed permitted by considerations of strength of the impeller is of the order 450-500 m/sec; and the pressure ratio of a centrifugal compressor with such a speed is ordinarily about 4 to 4.5. However, when an active impeller is used with supersonic diffusers and active impellers, the pressure ratio may reach 5 to 6.

The dimensions of a centrifugal compressor are determined by the specified air flow rate and axial speed of the air at the impeller inlet. The higher the air speed at the impeller inlet, the smaller the flow cross-section at the impeller inlet, for a given specified air flow rate, and the smaller the over-all compressor diameter.

The top speed at inlet should not exceed 120-150 m?sec, since further increase in axial speed leads to increase in losses on impact of the inflowing air on the rotating impeller vanes. Under this condition in centrifugal compressor designs now in service we have a specific air flow rate, that is, the ratio of the per-second weight flow rate of air to the frontal area of the compressor, of about 15-30 kg/ /m^2 sec, for impellers with single-side inlets. Use of double inlet impellers permits increasing air flow rates by 80-85% for the same dimensions, than when there is an inlet only on one side.

The brake horse-power efficiency of the compressor, that is, the ratio of the energy expended in compressing the air to the energy transmitted from the turbine, serves as a measure of mechanical and hydraulic losses. In centrifugal compressors, the brake horse-power efficiency amounts to 0.76 to 0.80, in which the lower value applies to compressors with high pressure ratios.

The air temperature in a cpmpressure increases due to the compression of the air, and for the indicated pressure ratios, the air temperature at outlet from the centrifugal compressor may reach 200 to

250°C.

2. INLET SECTIONS AND HOUSINGS OF CENTRIFUGAL COMPRESSORS

The inlet sections of centrifugal compressors with single impeller inlets are formed by parts of the compressor housing, and are so shaped as to ensure as much as possible the axial motion of the air in the inlet section. When parts are mounted on the front surface of the engine, the inlet section ordinarily has two air intakes 1 (see Fig. 52).

In the inlet sections of compressors with double inlet impellers, radial air intakes are used (see Fig. 53). Because of the turning of the airflow, a nonuniformity in the velocity field is observed at the inlet into the impeller. The installation of a diaphragm 1 serves to equalize the velocity distribution. The diaphragms and outside walls of the inlet section form a series of annular channels, in which the speed is accelerated from 60-70 to 120-150 m/sec, with an accompanying air pressure drop. The decrease of pressure leads to a loading of the outside walls of the inlet section by pressure forces, due to the difference in pressure between the air in the inlet and the surrounding air.

To increase the rigidity of the inlet section, ribs are used in the cast walls, and they are reinforced by shanks 6, made of cast material.

Preliminary whirling of the air in the inlet section is induced by the use of vanes 16 of the guide-vane assembly.

Protective screens 15 in the inlet sections are installed to prevent induction of foreign objects into the compressor.

Inlet sections of centrifugal compressors ordinarily do not have any heating system, since they are sufficiently heated by the heat of the compressor housing.

The housing of a centrifugal compressor is a basic support member of the engine, loaded both by forces arising directly in the compressor unit and forces transmitted to it from attached parts. Attachment points for fastening the engine to the aircraft are located on the compressor .

Centrifugal compressors casings are made of cast aluminum alloys with wall thicknesses of 6-7 mm. In order to insure rigidity, the o outer surface of the casing is ribbed. Compressor casings are ordinarily of a demountable design. Parts of the casing in addition to the inlet section, serve to form the air passage, the diffusers, and the outlet connection (see Fig. 52 and 53). The channels of the vaned difusor are formed by walls of 9 and 14 (Fig. 53) of the compressor-casing outlet and the vanes.

Diffuser vanes 10 (Fig. 52) may be fabricated in one piece with one of the walls of the diffuser casing. In such cases, the walls of the diffuser casing are held together by bolts 4, passed through the vanes.

In a demountable diffuser housing (see Fig. 53) inserted vanes are used. In order to increase the vibrational rigidity of the thin trailing edges, the inserted vanes may be attached. The forward part 8 of the vane is made of aluminum alloy, and the rear part 4 of steel. Inserted vanes have flanges which fit into grooves in the diffuser casing. Attachment of the vanes in the axial direction is accomplished on one side by a stop in the end of the groove, and on the other, by a "biscuit" 13; the "biscuit" is inserted in the groove and is fixed by screw 12.

The elbow-shaped outlet connection 10 with a small turning radius is located between the outlet connections of the casing and the combustion chamber. In order to decrease losses arising from the turning of the air, blades 11 are installed in the elbow. The individually fabricated vanes are installed in the transition-member form as the latter is cast.

3. CENTRIFUGAL COMPRESSOR ROTORS

Impellers are classified as to impeller geometry into unshrouded, shrouded, and semishrouded types.

Unshrouded impellers (Fig. 54,a) have an especially simple design, but have not gained widespread acceptance because their use is accompanied by substantial losses in the flow of air between the vanes; further, they exhibit insufficient vibrational strength.

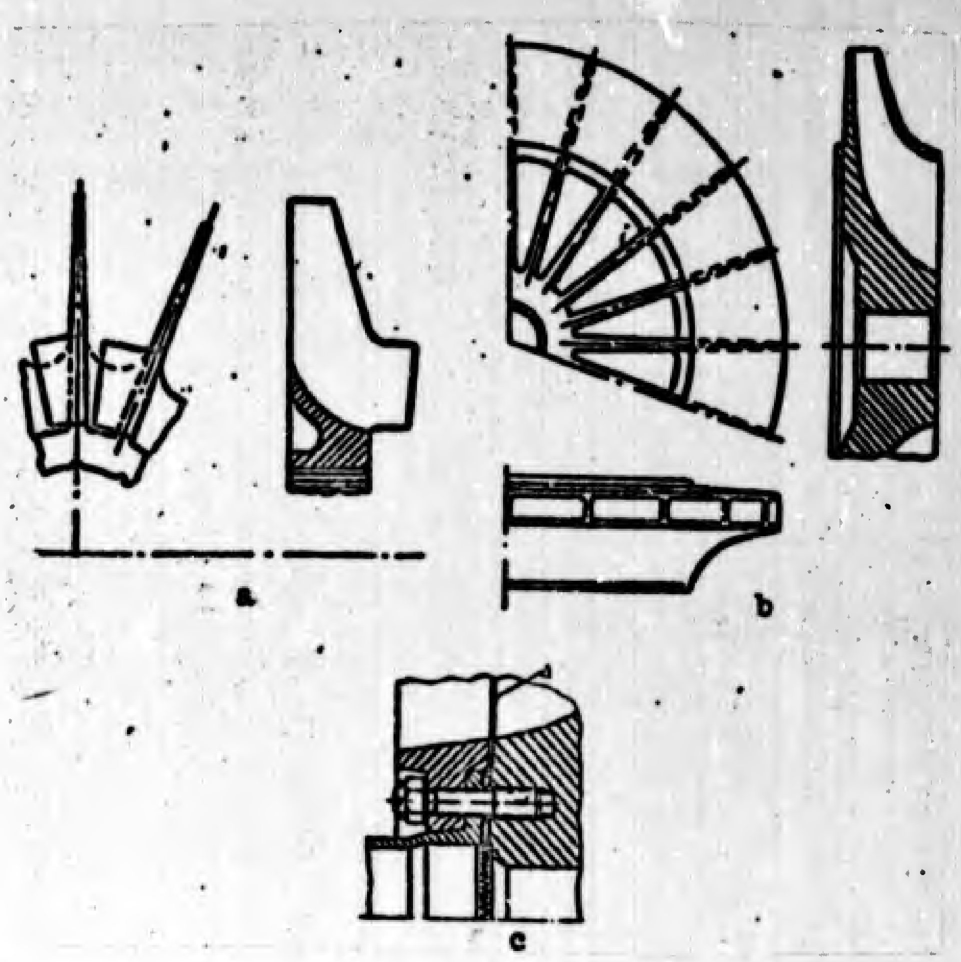

Fig. 54. Impellers. a) Unshrouded; b) shrouded; c) with rotating guide vanes.

Shrouded impellers (Fig. 54,b) are characterized by the lowest losses because in them the possibility of air escaping between the blades is completely out of the question. However, such impellers have also not come into extensive use because of difficulty in fabrication.

Semi shrouded impellers with single-side (see Fig. 52) and double-side (see Fig. 53) inlets are most widely used.

In order to provide shock-free entry of air from the inlet section into the impeller, the vanes are bent at their forward [leading] edges opposite to the direction of rotation. The bending angle changes with the radius. The bending of the edges is carried out in the heated state after machining of the impeller.

To ensure shock-free entry of air into the impeller, in addition to bending the upstream edges of the vanes, a set of turning guide vanes 18 is also installed in front of the impeller (see Fig. 53).

The torque from the shaft to the impeller and the rotating guide vanes may be transmitted by frictional forces between the ends of the flanges on the shaft and the impeller, by end splines, and in the case of small torques, for example in ventilating fans or turbostarters, by rectangular or evolute grooves.

In impellers with rotating guide vanes, the flange on the shaft and the guide-vane assembly are simultaneously tightened by dowels 17. The vane ends of the guide-vane assembly, facing the vanes of the impeller, are cut into a conical shape. In tightening down the dowels, we find that the clearance Δ (Fig. 54,c) at the hub is reduced, and that at the periphery, a close mating is effected between the vanes of the guide assembly and those of the impeller.

To prevent loosening of the dowels during operation, they are screwed into the impeller up to a stop at the end of the threaded hole. Pretensioning permits equalizing the load on the individual screw turns in tightening down the flange.

Coupling on end splines (Fig. 54,c) is resorted to when relatively large torques are transmitted, and when it is not possible to transmit them by use of frictional forces.

In double-side inlet impellers, nonuniform flow is observed at the outlet from the impeller. This is explained by the various conditions at inlet into the impeller and, consequently, by various airflows into the fore and aft sides of the impeller. In order to straighten the airflow ahead of the inlet to the diffuser, grooves are cut into the periphery of the disk.

4. LOADS ACTING ON THE IMPELLER. IMPELLER VIBRATION.

Figure 55, a shows diagram of the loading of a centrifugal compressor impeller by forces of intrinsic weight, inertial forces, torques, axial forces, and forces due to air pressure in the flow passage.

Aerodynamic forces on impeller vanes tend to bend them, and centrifugal forces due to their own mass tend to elongate them. If the impeller has active vaning, the centrifugal forces due to the intrinsic mass of the vanes cause bending.

The disk of the centrifugal compressor is loaded by centrifugal forces due to its own mass and by centrifugal forces transmitted from the vanes. Due to this loading, tensile stresses σ_r and σ_t (Fig. 55,b) arise in the impeller disk, just as in the disk of an axial compressor.

As a result of nonuniform heating of the impeller along the radius, thermal stresses also are developed in the impellers: Maximum stresses in impeller disks made of aluminum alloys reach 2000 - 3000 kg/cm^2.

Axial forces arise in impellers primarily due to differences in air pressure, and also by reason of changes in the air flow in the air passages.

In the double-side inlet impeller, axial forces arise only in case the air inlet conditions are different at the two inlets; thus the entry of air into the rear inlet might be blocked by the ends of the combustion chambers (see Fig. 6).

In the single-side inlet impeller, the pressure on its rear side is about equal to the pressure at the outlet from the impeller, that is, it is always larger than on the front. Because of this difference in pressure, an axial force in the direction toward the air inlet to the impeller is generated. In case it is necessary to reduce the axial force, in particular to unload the thrust bearing of the rotor, a labyrinth seal is mounted on the back side of the impeller.

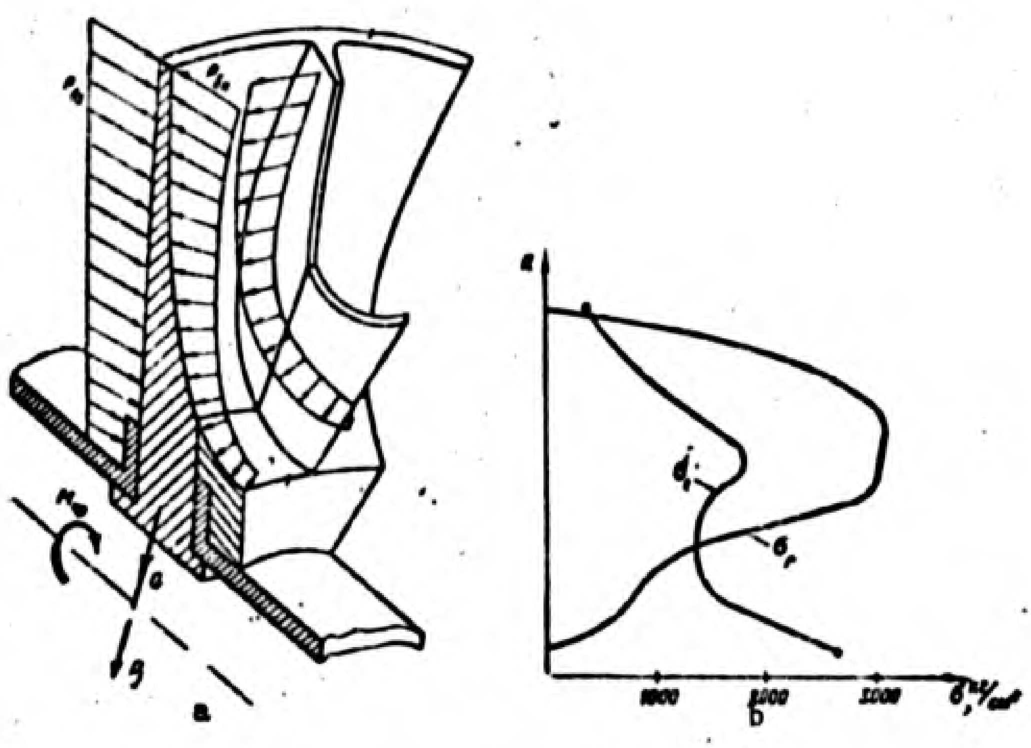

Fig. 55. Loads on the Compressor Impeller. a) Pattern of the action of forces and moments on the impeller; b) radial distribution of stresses in the impeller; p_{v_p} and p_{v_z}) air pressure on the front and rear sides of the impeller; M_{kr}) torque of the turbine; g) force of gravity; p_j) inertial force; R) radius; σ_r) radial stresses; σ_t) circumferential stresses.

Nonuniformity of the airflow into a centrifugal compressor can cause harmful resonance vibrations in the vanes of the impeller wheel

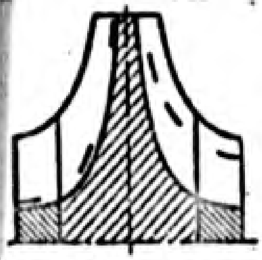

Fig. 56. Incidence of fatigue cracks in the vanes of a centrifugal compressor.

[rotor], the guide-vane assembly, or the diffuser. Extended operation of the engine under conditions inducing resonance fibrations can lead to fatigue failure of the vanes. Figure 56 shows the places (dashed lines) where fatigue cracks appear most frequently in the vanes.

The vanes of the impeller wheel of the centrifugal compressor are joined to the disk over a considerable part of their contour, so that their vibrations are closely connected with the vibrations of the disk. In a double-side inlet impeller, two forms of vane and disk vibration are characteristic:

1) when the vanes located on one side of the disk are vibrating relative to a nonvibrating disk, and are deflected in the same direction as the paired vane on the opposite side of the disk (Fig. 57, a);

2) when a pair of disk vanes vibrate together with the disk, and are bent in opposite directions (Fig. 57, b). In this case, the disk is subject to bending vibration along with the vanes.

The lowest vibrational modes of the vanes of an impeller wheel, with their nodal lines, are shown in Fig. 57, c.

Coupled vibrations of the impeller vanes and the rotating guide vane assembly depend on the magnitude of the frictional forces between the vanes, i.e., on the tightness of the contact at the ends of the vanes. In an operating engine, the tightness is continuously reduced by reason of wear of the surfaces in contact, due to their vibration. In the case of very low contact pressure at the end, the amplitude of the vibrations is substantially increased, causing rupture of the blades due to metal fatigue. Some modes of coupled vane vibrations in a composite impeller wheel of a centrifugal compressor are shown in Fig. 58.

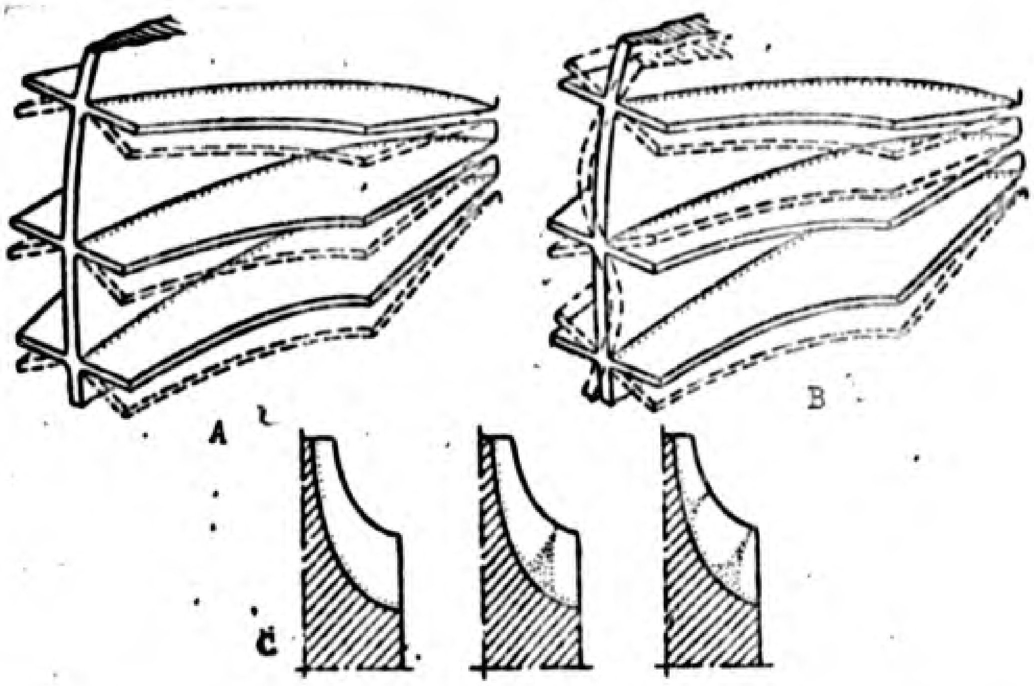

Fig. 57. Vane vibrations of a centrifugal compressor. a) Vanes are vibrating relative to a nonvibrating disk; b) the vanes are vibrating along with the disk; c) locations of the nodal lines for various modes of blade vibration.

For elimination or reduction of harmful resonance vibrations of centrifugal compressor vanes, the same methods are used as for elimination of harmful blade vibrations in axial compressors.

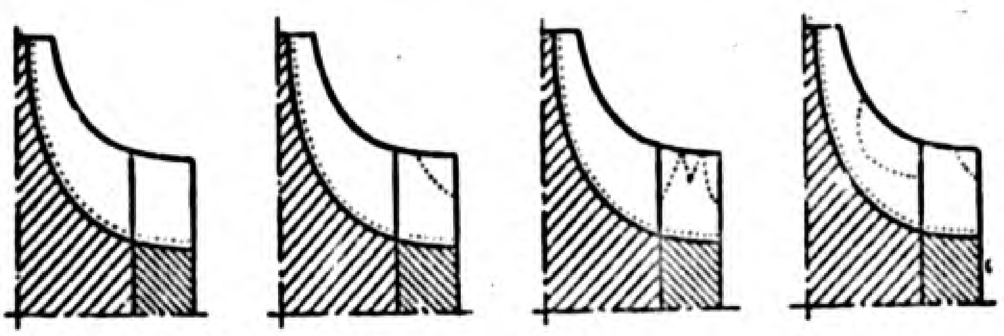

Fig. 58. Modes of coupled blade vibration of rotating guide-vane assembly and of the rotor of centrifugal compressor.

Part Two

GAS TURBINES

Chapter 3

GENERAL CONSIDERATIONS

1. OVER-ALL GEOMETRY OF THE GAS TURBINE.

The gas turbine is one of the basic parts of the gas turbine engine. It is used for driving the compressor of a turbojet engine; in the turboprop engine, it is used in addition for turning the propeller.

The source of the useful work of the turbine is the potential energy of the gas, obtained by compressing the air in the compressor and then heating it in the combustion chamber to a high temperature. The conversion of the potential energy of the gas into mechanical work at the turbine shaft is carried out in one of the turbine stages, particularly in a stage consisting of a stationary nozzle diaphragm, and a turning rotor (Fig. 59).

The nozzle diaphragm has nozzle blades 1, arranged in the form of an annular grid, and these form convergent curved channels for the passageexpansion of the gas. The turbine rotor consists of a disk 2, seated on the turbine shaft 3, and turbine blades 4, attached to the disk.

The nozzle diaphragm provides for conversion of a part of the potential energy of the gas into kinetic energy, and directs the gas into the rotor blading at a particular angle. A part of the kinetic

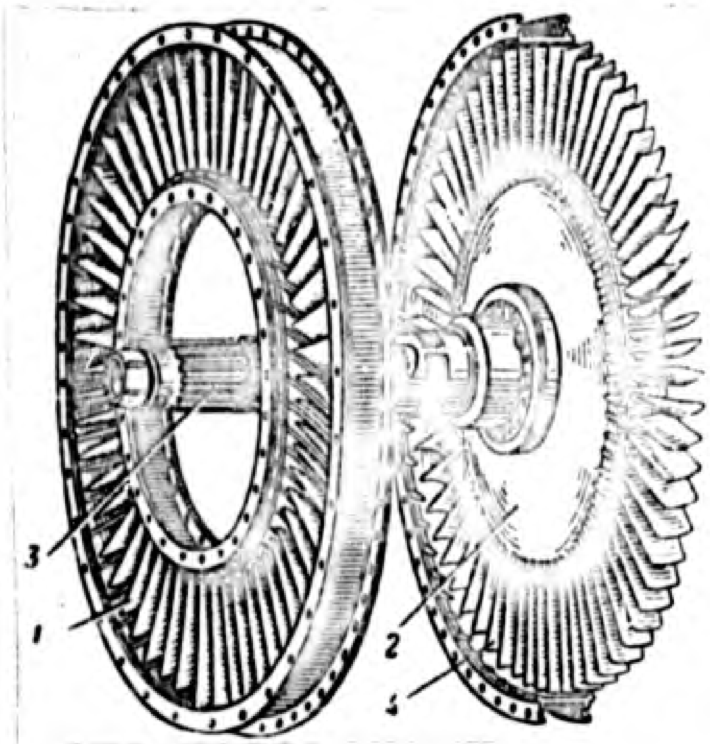

Fig. 59. Nozzle diaphragm and rotor of gas turbine. 1) Nozzle blades; 2) disk; 3) turbine shaft; 4) turbine blades.

energy acquired by the gas in its expansion in the nozzle diaphragm is used for turning the wheel.

The turbine-rotor blades also form channels in which partial expansion of the gas occurs thus effecting an increase in the power of the turbine.

2. STRUCTURAL DIAGRAMS OF GAS TURBINES.

In terms of structure (design), gas turbines are classified with respect to direction of gas motion as radial and axial types; with respect to number of stages, they are classified as single stage and multistage turbines. Multistage turbines may be single rotor and double rotor types.

In a radial turbine the gas is directed by the nozzle diaphragm into the turbine rotor in a radial direction, from the center toward the periphery, or conversely, from the periphery toward the center. In

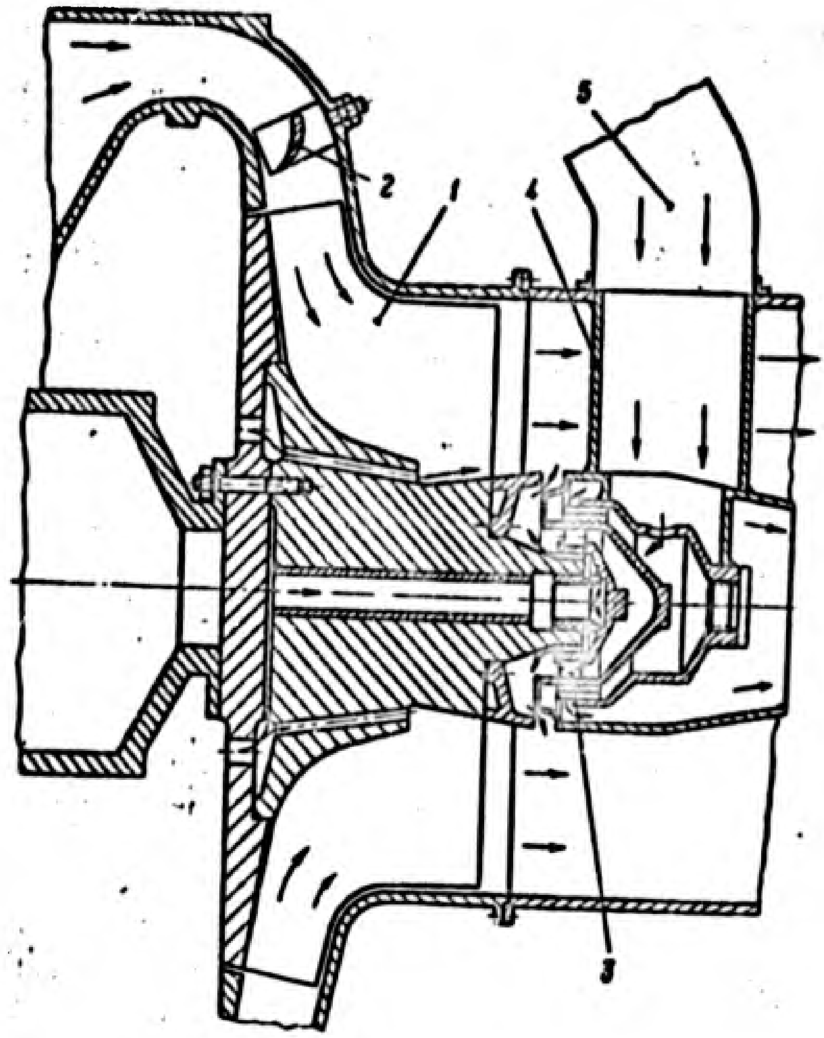

Fig. 60. Centripetal radial gas turbine. 1) Turbine wheel; 2) blade of the nozzle diaphragm; 3) turbine bearing; 4) turbine bearing housing; 5) connection for inlet of cooling air from the atmosphere.

the former case, the turbine is called a centrifugal radial turbine, and in the latter case, a centripetal turbine. In comparison with the centrifugal turbine, the centripetal turbine has, under equivalent conditions otherwise, a higher efficiency. A diagrammatic sketch of the centripetal radial turbine is shown in Fig. 60.

The simplicity of the design and fabrication of radial turbines is their principal advantage in comparison with axial turbines. The production of radial turbines is made particularly simple by the use of cast turbine rotors. Turbines of this type are used for driving

AC generators, in aviation turbocooling equipment, turbostarters, and in other auxiliary equipment of relatively low power. In gas-turbine engines of medium and great power, radial turbines ordinarily are greater in size and weight than axial turbines.

In low-power gas turbine engines with radial turbines where the turbine rotor is joined directly to the compressor impeller without any intermediate shaft (Fig. 61), the engine turns out to be expecially simple in design.

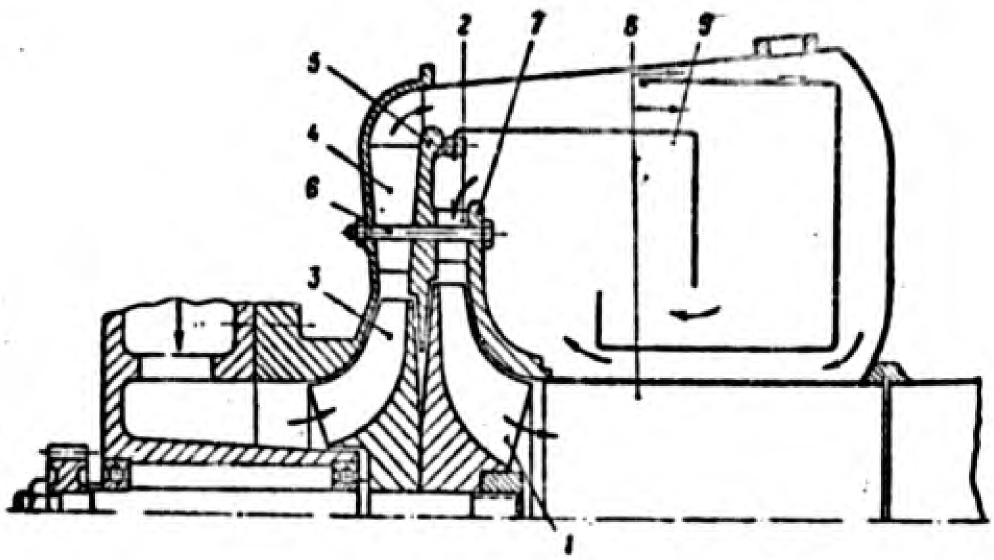

Fig. 61. Diagram of a low-power gas-turbine engine [GTD] with radial turbine, located on the compressor. 1) Turbine rotor; 2) nozzle diaphragm; 3) centrifugal compressor; 4) diffuser; 5) cast iron sealing ring; 6) bolt for attachment of nozzle-diaphragm ring and the turbine housing; 7) Turbine housing; 8) exhaust pipe; 9) inner liner of the combustion chamber.

<u>Axial gas turbines</u> [GTD] are the mose widely used types in modern turbojet engines. In such turbines the axial direction of gas motion is approximately maintained at the inlet to and the outlet from the turbine stage. The design of a single-stage axial turbine is shown in Fig. 62.

In multistage single-rotor turbines, a series of turbine rotors are mounted on one shaft (Fig. 63). In multistage two-rotor turbines, some of the wheels are mounted on one shaft, and some on another; these wheels are not kinematically connected, and turn independently at various speeds, (Fig. 64). Such types of gas turbines are used in two-rotor turbojet [TRD] and turboprop [TVD] engines, and also in gas-turbine installations for VTOL aircraft.

In ducted-fan [TRD] turbojet engines, it is possible structurally to have the axial fan joined to the turbine. Thus, for example, in the birotational axial turbine shown in Fig. 65, the blades of the fan are mounted on a shroud fixed to the tips of the turbine blades. Such a turbine design substantially reduces the size of the engine along the axis, but at the same time it increases the loading on the turbine rotor blades.

The design arrangements of gas turbines under consideration do not, without a doubt, exhaust the possibilities of design variations, differing in numbers of stages, kinematic relationships between them, and other design features. The selection of this or that design arrangement is determined by the basic engineering-economic turbine indices and by the ease of incorporating the turbine into the gas-turbine engine adapted for a specific type of aircraft.

3. REQUIREMENTS FOR THE DESIGN OF A GAS TURBINE.

In the preliminary design of a gas turbine, and methods of their realization one always strives to implement all the design measures, and those of a metallurgical and technological nature to ensure high values of adiabatic and brake horsepower efficiency, great power generated by turbines of small size and weight, reliability in operation over the required period of service between overhauls.

The adiabatic and brake horsepower efficiencies of modern turbine engines reach fairly large values, and amount to 0.90 - 0.93 and 0.80 - 0.85 respectively; any further substantial increase in

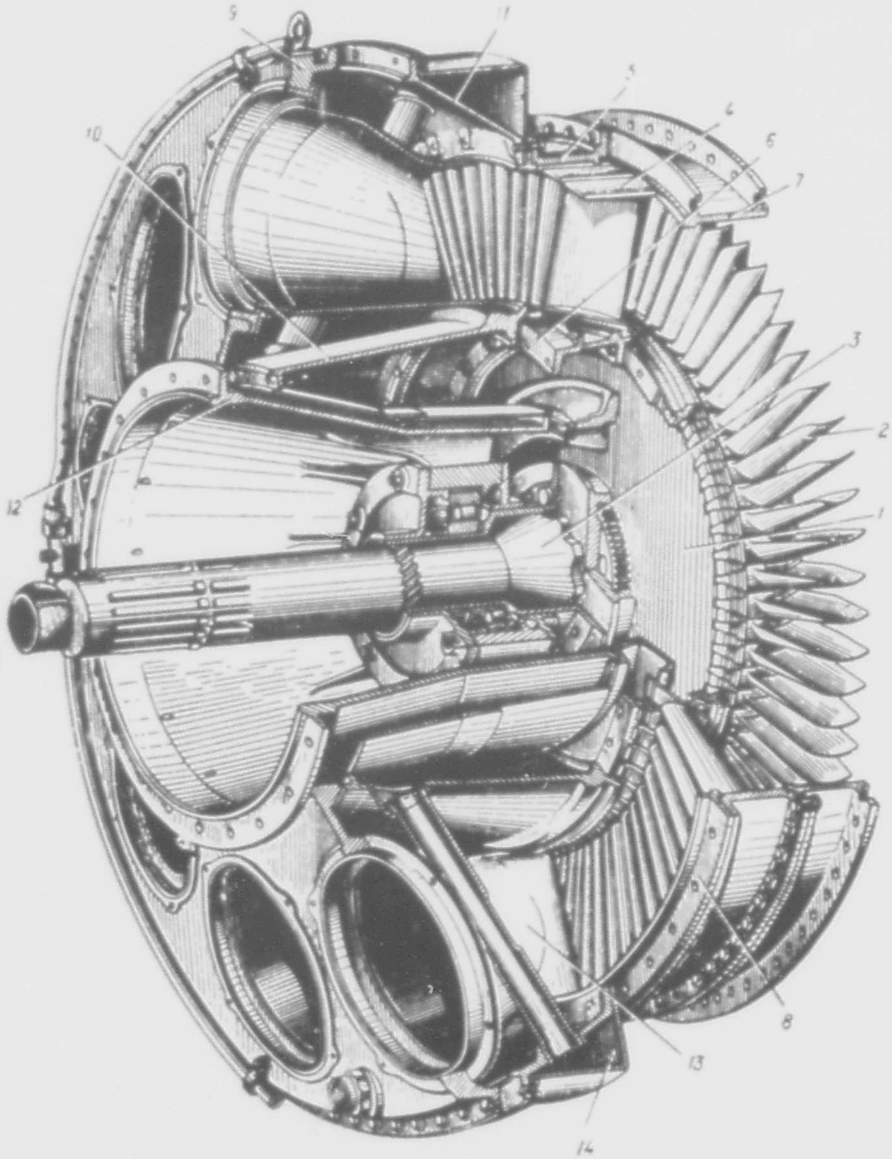

Fig. 62. Single-stage axial gas turbine. 1) Turbine disk; 2) rotor buckets; 3) turbine shaft; 4) nozzle blade; 5) outer shroud of the nozzle diaphragm; 6) inner shroud of the nozzle diaphragm; 7) turbine housing; 8) intermediate centering ring; 9) housing of the gas collector [discharge nozzle]; 10) drum for mounting of the inner nozzle-diaphragm shroud; 11) attachment ring for the external nozzle-diaphragm shroud; 12) turbine bearing housing; 13) gas collector [discharge nozzle] connection; 14) air removal compartment.

these magnitudes is hardly possible in practice.

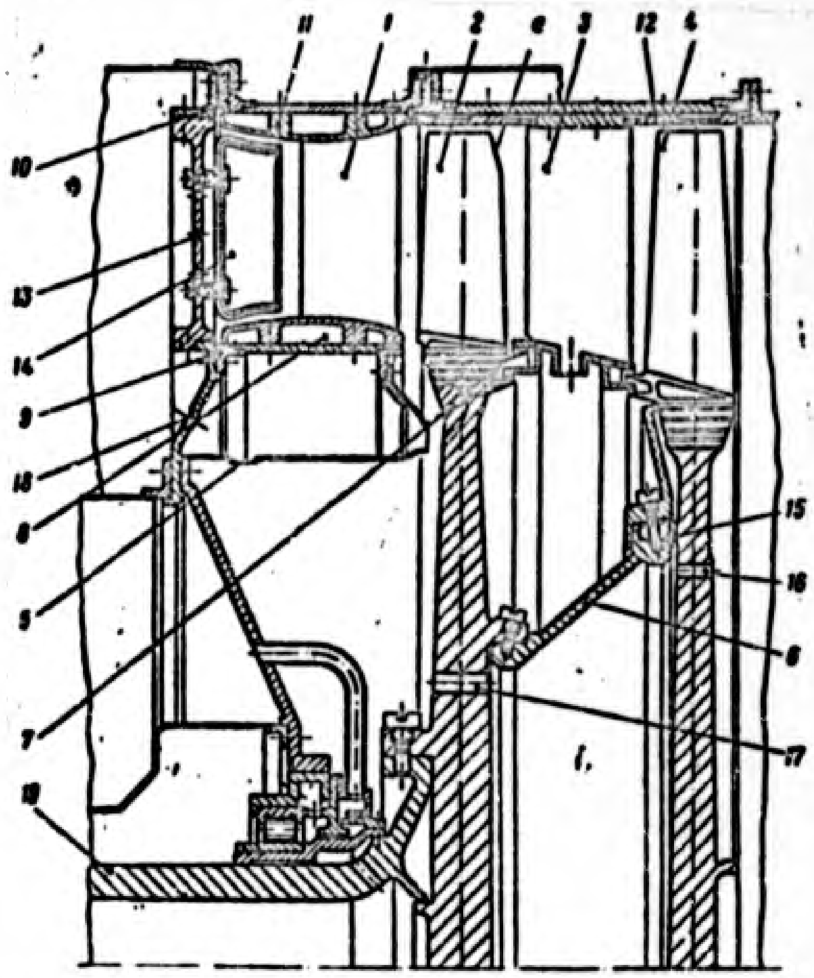

Fig. 63. Two-stage single-rotor axial gas turbine. a) cut; 1) nozzle [stator] blade of first stage; 2) turbine [rotor] blade of first stage (the blades are trimmed); 3) nozzle blade of second stage; 4) turbine blade of second stage; 5) distributor for cooling air; 6) load-bearing attachment ring of second stage disk; 7) inclined cooling orifices; 8), 11), and 12) lining shoes; 9), 10), 15), 16), 17) and 18) orifices for passage of cooling air; 13) support struts; 14) strut fairings; 19) shaft.

High efficiency values are provided by:

proper choice of the number of stages and the values of the operational parameters of the gas in the turbine slow passage;

efficient shaping of the turbine and nozzle blades to reduce radial gas flow in the axial clearance between nozzle diaphragm and turbine wheel, and also to reduce vortex flow of the gas downstream

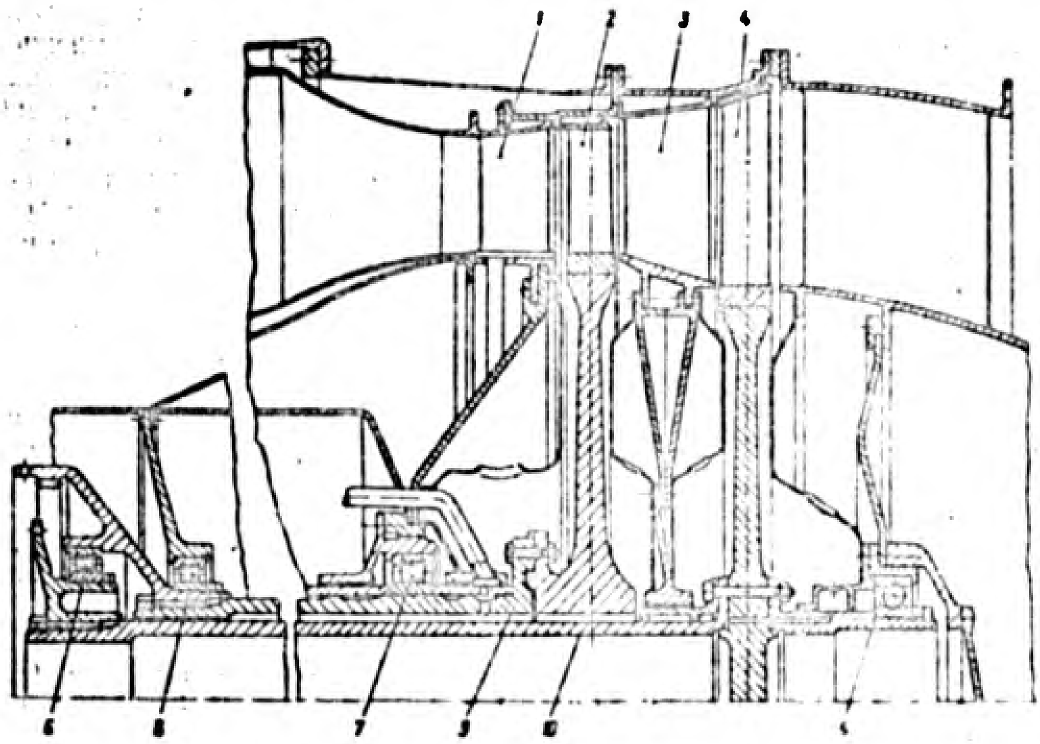

Fig. 64. Two-stage two-rotor axial gas turbine. 1) Nozzle diaphragm of first stage; 2) turbine wheel of first stage; 3) nozzle diaphragm of second stage; 4) turbine wheel of second stage; 5) and 6) bearings of the second rotor; 7) and 8) bearings of the first rotor; 9) and 10) shafts.

of the turbine;

decrease in the recirculation of the gas between stages and in the radial clearance between the casing and the blades, using for this purpose labyrinth seals, shrouding of the turbine blades, installation of special inserts in the turbine casing over the blading, and other design measures;

careful surface finishing of the blades for the purpose of reducing friction losses.

Modern turbines develop great power with relatively small sizes and weights, and this is accomplished by;

increasing the heat difference converted in one stage into mechanical work at the turbine shaft; effective conversion of large temperature differences per stage is associated with an increase in cir-

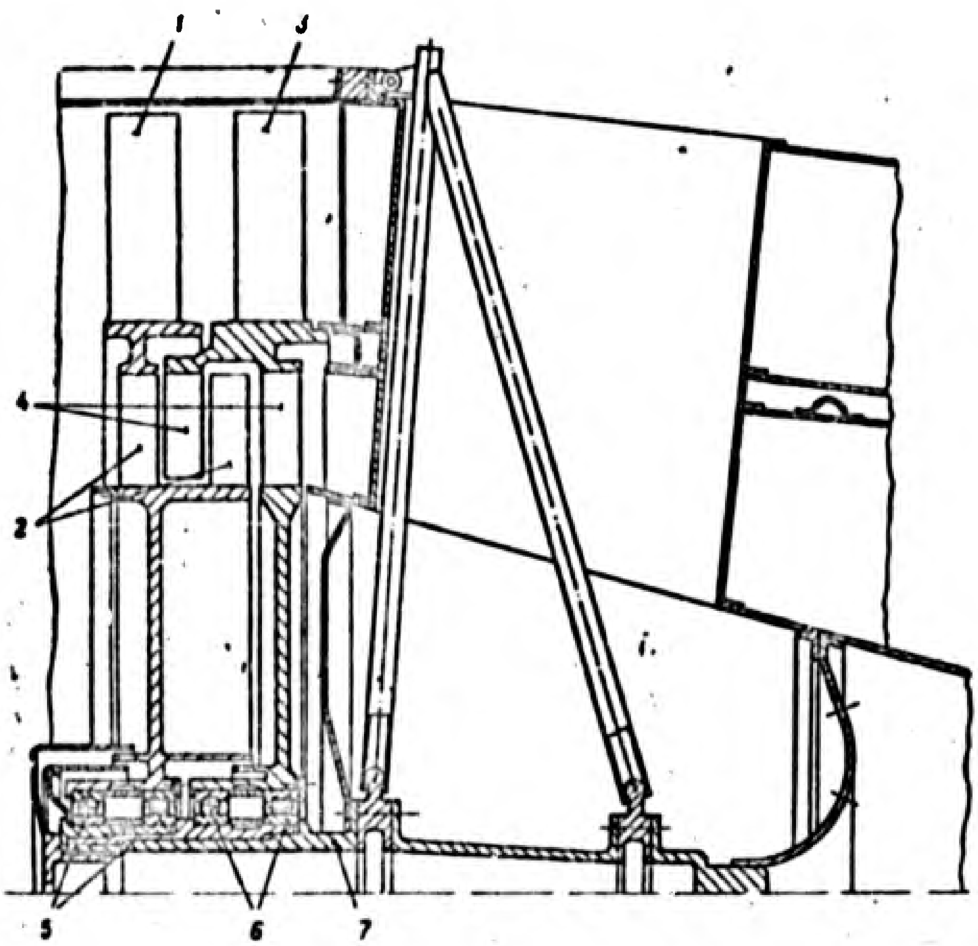

Fig. 65. Birotational axial gas turbine. 1) and 3) Fan blades; 2) and 4) turbine blades; 5) bearings of the first rotor; 6) bearings of the second rotor; 7) bearing support.

cumferential speed, changing at the mean radius of the wheel within a broad range from 330 to 450 m/sec, in which case the circumferential speed is limited by the stresses tolorated in the turbine rotor elements;

the use of high axial gas speeds (about 300 - 500 m/sec) in the flow passage at the turbine outlet;

increasing the gas temperature in front of the turbine nozzle diaphragm, leading to a sharp rise in the specific turbine power; in current engines, the temperature before the turbine amounts to 900 - 1000° C, and the specific power, to 300 - 400 hp sec/kg;*

increasing in the permissible stress in the turbine blades and turbine disks through improved mechanical characteristics of the construction materials, and also research on efficient design shape of these elements, corresponding to the requirement of equal design strength;

application in fabrication of turbine parts of heat-resistant construction materials with reduced specific-weight values.

Reliability of turbine operation in the course of the prescribed period between overhauls is provided by the application of:

heat-resistant steels and alloys, and also ceramic materials and high-temperature corrosion resistant coatings for turbine parts subjected to the action of high temperatures and high loads;

more up-to-date equipment and methods for parts inspection, insuring high quality fabrication of parts and completely eliminating flaws;

special design measures such as, for example, the cooling of parts which are subjected to high temperatures and loads, or decreasing their heating by use of thermal insulation and heat-reflecting screens, etc.

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[Footnotes]

121 * The adiabatic efficiency characterizes the hydraulic losses in the turbine. It is the ratio of the useful energy of expansion hydraulic losses accounted for to the adiabatic expansion work of the gas. The brake horsepower efficiency is the ratio of effective turbine shaft work to the energy expended.

Translator's note: This unit would seem to be in error. The unit hp sec/kg would hardly be referred to as a specific power (though possibly as a specific fuel consumption). It is surmised the author meant to write hp/kg.

Chapter 4

ROTORS OF GAS TURBINES

1. DESIGN SHAPES OF TURBINE ROTOR BLADES AND DISKS.

The profile of the shaped part of a turbine blade is determined by the geometric parameters shown in Fig. 66.

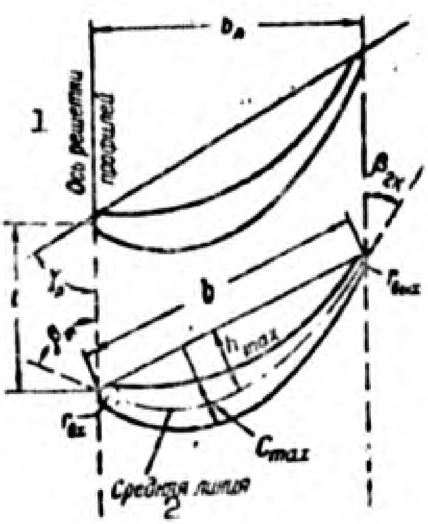


Fig. 66. Geometrical parameters of the profile of a turbine rotor blade. b) Profile chord; b_1) width of blade profile in the axial direction; c_{max}) maximum profile thickness; β_{1k} and β_{2k}) design angles for entrance and exit of gas flow; h_{max}) camber of profile; r_{vkh}, r_{vykh}) bending radii of leading and trailing edges; t) blade pitch; γ_1) profile setting angle. 1) axis of profile grid; 2) center line.

An important parameter is the profile chord. The magnitude of the chord is evaluated in relation to blade length and is expressed as the ratio of the blade length to the blade root.

In existing engine designs, the lengths in TRD (turbojets) from 2.3 - 4.0 and TVD (turboprops, from 1.5 - 4.0 and the smaller values correspond to the first turbine stage. Designers try to make the turbine blade length as large as possible in relation to the profile chord, because in this way the diameter of the disk rim is reduced, and consequently the turbine rotor weight is reduced. However, reduction in the chord is blocked by an increase in the bending stress at the blade root section and by the reduction of the normal blade vibration frequencies lowering the vibrational strength of the blade.

The maximum profile thickness of the blade is chosen by the designer to be as small as possible in order to reduce turbine weight and achieve high hydraulic characteristics of the flow passage. Its minimum value is limited by the conditions of strength and for the profile of the root section is equal to 20 - 25% of the chord length. In conformity with the equal strength principle along the blade, the fin thicknesses of the profile decrease in the direction root-to-tip. At the mean diameter the profile thickness amounts to 10 - 12%, and at the periphery to 4 - 6% of the chord. The ratio of the maximum profile thickness at the tip section to the maximum profile thickness at the root section determines the blade taper. Because of the taper, the blade cross-sectional area decreases from root to periphery. The free tip of the blade is sharpened to decrease the flow of gas through the radial clearance between the blades and the turbine casing. The maximum profile thickness is located at a distance of about 15 - 30% of the chord from the leading edge of the blade.

The leading and trailing edges of the blade are chosen in conformity with the design angles of entry and exit of the gas flow, which are the angles between the axis of the grid and lines tangent to the center lines at entrance and exit edges, respectively. These angles are determined by gasdynamic computations and differ from the hydraulic angles of flow entry and exit by the angles of gas-flow, inflow and discharge, respectively.

The largest ordinate of the mean line of the profile is called the profile camber. The profile camber decreases from the root section toward the tip in accordance with the change in the angles of inlet and exhaust of the gas flow.

At the leading and trailing edges of the profile, the shape is rounded to decrease the stress concentration. The radius of curvature of the leading edge is 0.02 - 0.06, and of the trailing edge, 0.01 - 0.02 of the profile chord. With decrease in thickness of the trailing edge, the hydraulic losses in the profile grid are reduced; However, the thermal stresses in the blades are increased in an operating engine in nonsteady regimes because of nonuniform heating of the blade fin.

The rotor buckets [blades] are arranged on the rim of the disk with a definite spacing [pitch] to insure the best hydraulic turbine characteristics. The blade pitch (or spacing) $\underline{t}$ and the diameter D_1 of the disk rim determine the number z_{r1} of blades which must be set on the disk. The computation is carried out by use of the formula $z_{r1} = \pi D_1/t$.

In a number of cases, the number of blades determined according to the optimum pitch value is changed for the purpose of removing harmful resonance blade vibrations from the range of operating rotor speeds, thus insuring the necessary vibrational blade strength. The number of blades may also be limited by the conditions under which they are set on the disk and by the increase in the stresses that arise within the disk at the point of blade attachment. The use of turbine grids with pitch somewhat larger than optimum, permits, at a cost of some reduction in turbine efficiency a reduction in the number of blades and in the weight of the turbine rotor, a reduction in the cost of fabrication, as well as in the need for expensive materials, a lowering of the stresses in the blades by thickening the root sections without distortion of the shape of the interblade channels, and a lowering of the expenditure of energy on the cooling of the blades.

Since the pitch of the lateral blade sections increases in the direction from blade root to the periphery, the profile chords in this direction are also increased or kept constant in order to keep the ratio between chord and pitch of the profiles as near the optimum value as possible on the longer length of blade. The ratio of the profile chord at the tip section to that at the root section determines the extent to which the blade is trapezoidal in shape. An impediment to the increase of the profile chord in the direction of the blade tip is that complex blade-fin vibration modes are thereby more easily excited; this fact makes it necessary in such cases to use a profile with decreasing chord, or to trim the edge of the shaped part of the blade (see Fig. 63). In current turbine designs the extent to which blades are made trapezoidal ranges from 0.8 to 1.35.

Efforts are made to design blade profiles so they will be simple to fabricate. They are therefore described by the simplest possible curves, i.e., conjugate circular arcs with straight-line seements. Most usually the convex surface of the blade profile is described by two circular arcs, and the concave side, by one circular arc (Fig. 67, a). The machining of the concave surface of such a blade is significantly simplified in that it has a cylindrical or conical surface, and the convex surface is machined by use of a template.

The lowest hydraulic losses are achieved, as experiments show, in profiles with continuously changing curvature, without discontinuities in the curvature at the points of contact. The lemniscate profile (Fig. 67, b) satisfies this condition.

As a result of blade shaping in accordance with the selected manner of gas flow, the blade twists along the height and there is an increase in the tapering of the channel with increasing distance toward the periphery.

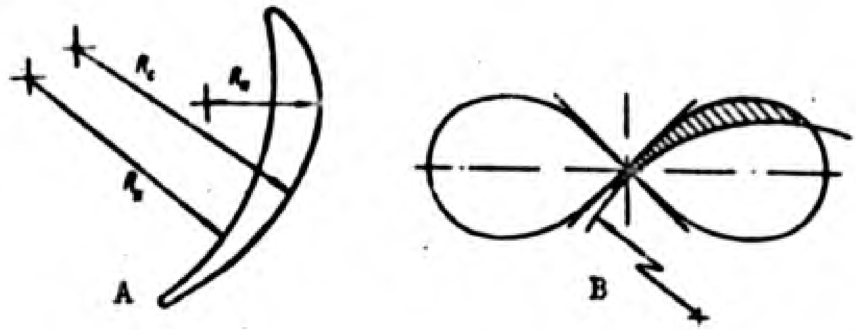


Fig. 67. The profile shape of turbine rotor blades. a) Profile of arcs or circles; b) profile formed by Bernoulli's lemniscate (convex surface), and an arc (concave surface).

The rotor blades of shrouded turbine wheels have flanges 4 at their ends (Fig. 68), fabricated integrally with the blades. In assembled state they form a shroud which decreases the radial flow of gas in the turbine.

Between the shaped 1 and key 5 parts of a blade there is a flange 2 which forms the flow passage of the turbine wheel, and there is the transition 3, by use of which the blade key can be brought toward the center of the disk, i.e., displace to a zone of lower disk temperature, so that the disk can be made of less heat-resistant material.

Chrome-nickel with an admixture of alloying components is used for fabrication of turbine blades. Turbine-blade blanks are usually stamped, and here it is necessary to make sure that the fibre of the material is longitudinal to the blade; or they may be precision cast. In the conventional stamping method, an allowance is left on the blank for subsequent machining. Precision casting makes it possible to obtain the desired fin shape and dimensions with accuracy to ± 0.1 mm. Machining in this case is limited to polishing.

To ensure high efficiency, the turbine blades are subjected to careful control during production; here the precision of positioning profile sections along the length of the blade is inspected, as is the positioning of the fin relative to the key, and also the precision of the shape sections themselves, and the machining finish of the blades.

Fig. 68. Blades of shrouded turbine wheel. 1) Shaped part of the blade; 2 root flange; 3) transition; 4) flange of shroud; 5) key part.

In order to simplify the process of balancing the rotor, we separate the blades according to weight, and for assembly the blades are chosen in such fashion that variations in weight for the various blades of the assembly do not exceed 5 - 10 g. Further, blades are set in opposite slots of the rim with differences in weight no greater than 1 - 2 g.

Blades are subjected to selective control as to frequency of the normal vibrations. In case the frequency falls below the set limit, some material is trimmed off the end of the fin of the blade, or the attachment stiffness of the keying groove is increased. In some turbine designs, the requisite frequency of normal vibrations of the blade is achieved by taking substantial cuts a (see Fig. 63) from the trailing edge of the fin.

Gas-turbine disks are intended for holding and supporting turbine blades on their rims. The rim itself consists of a widened peripheral part of the disk. The shape and dimensions of the rim are determined by the type of keying attachment of the blades to the disk, and the magnitude of the loads acting on the junction.

The rim is made as small as possible, since its size affects in

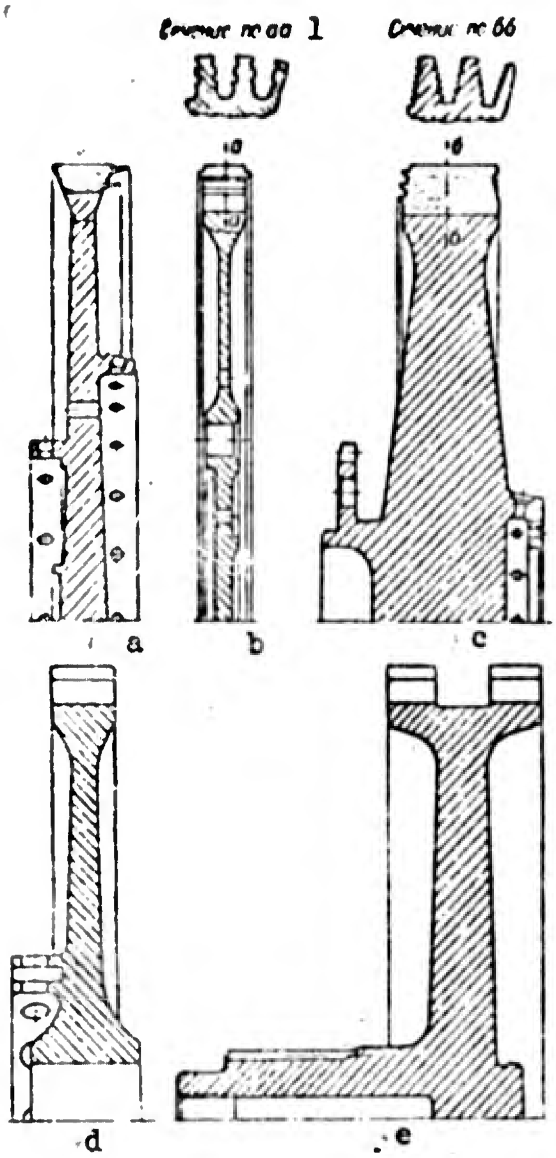


Fig. 69. Cross-sections of integral turbine disks: a) Conical disk; b) plane disk; c) disk with conical and plane parts; d) disk with central hole and hub; e) disk with tail and double rim; 1) section along .

turn the loads on the body of the disk and, consequently the thickness and weight of the disk. The transition from the body of the disk to the rim is faired, which substantially lightens the over-all turbine assembly.

To simplify the mechanical fabrication of the disk, its shape is formed of as simple lines as possible. Most frequently conical and plane disks (Fig. 69) are found in practice.

The body and rim of the disk are made, as a rule, in one piece. However, in some cases, the peripheral part of the disk with the rim is made up separately, and then set on the central part and welded to it (Fig. 70, a). Under operating conditions, circumferential breaking stresses in the rim due to the welding compensated the circumferential compression stresses occurring in the rim because of nonuniform temperature distribution along the disk radius. The composite welded disk reduces the requirement for high-alloy high hot-strength steel, and simplifies the coupling of the disk to the shaft.

For design reasons, in a number of cases disks are made with a central hole (for example, in double rotor engines, when the disk is

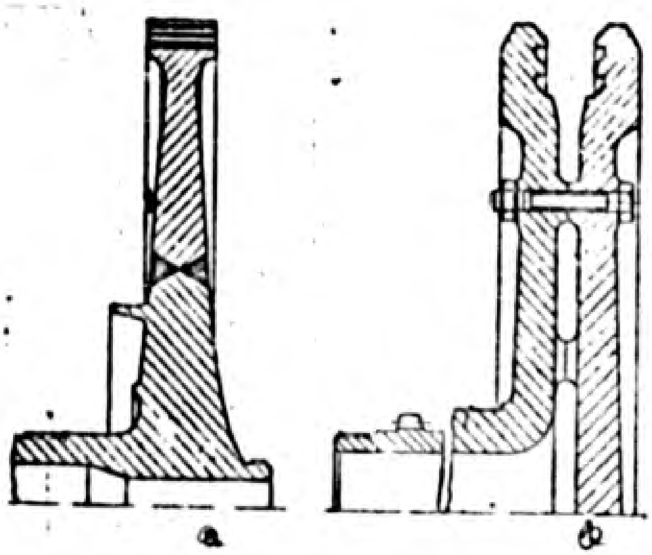


Fig. 70. Sections of composite turbine disks. a) Disk with peripheral part welded on; b) disk with joint in the plane of symmetry.

pressed onto the shaft, etc). In such cases, the disk has a broadened central part -- a hub -- reducing the magnitude of stress at the hole radius (see Fig. 69, d).

In operation, forces acting on the turbine disk are: centrifugal forces due to the blades attached to the disk, and due to its intrinsiec mass; gas forces transmitted from the turbine blades, and moments due to these forces; thermal forces arising because of the nonuniform distribution of temperature along the radius and thickness of the disk.

Axial gas forces P_{og}, acting on the turbine blades and the body of the disk, create a moment that bends the disk. In order to counter this bending moment by a moment due to centrifugal forces on the blades and the rim, the center of gravity of the rim is usually displaced in relation to the plane of symmetry of the disk in the direction of the action of the bending moments of the gas forces (Fig. 71). The magnitude of this displacement in the turbine disks of some engines amounts to 2 - 4 mm.

Thermal stresses arising in the disk originate from nonuniformity in the temperature distribution along the disk radius and thickness. The nonuniformity is paricularly large in the turbine disk in non-steady regimes of operation -- on starting up and shutting down the engine. At start-up of the engine, the rim is heated faster than the central part of the disk. On this account, the outer layers of the disk, as they are heated more intensively, tend to expand circumferentially and in the radial direction to a greater degree than the inner less-intensively heated layers which oppose such expansion. Therefore large circumferential compression stresses are generated in the rim of the disk. As the disk is heated up, the intensity of the thermal stresses is reduced. However, even in steady operational turbine regimes, the temperature of the central part of the disk remains less than that at the rim. Thus, for example, in the VK - 1 engine, the temperature of the disk is 150° C at the center, and 500° C at the rim. The total tangential compression stresses in the rim amount in this case to about 2500 kg/cm^2.

The other case that is dangerous to the integrity of the disk may be engine shut-down, when the rim is cooled faster than the central part of the disk. Especially fast cooling may take place in the case of engine shut-down in flight, at low ambient air temperature. In this case, thermal tensile stresses arise in the disk rim, and thermal compression stresses arise in the center.

Nonuniform distribution of the temperature across the thickness of the disk leads to the appearance of bending thermal stresses.

2. ATTACHMENT OF TURBINE BLADES.

Attachment of turbine blades to turbine disks is accomplished by use of keying attachments which are included among the especially highly stressed elements of gas turbine designs. Less frequently, the

the blades are attached by use of welding or soldering.

The largest load on a keyed junction is imposed by centrifugal forces due to blades, and there reaches 10 t and more. This load is therefore accorded special attention in the design and strength calculations of the keys. The relatively high temperature of the key (600 - 700° C) substantially lowers its mechanical strength.

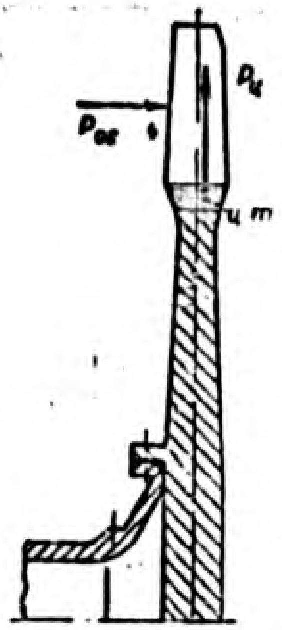


Fig. 71. Displacement of the center of disk rim gravity for purpose of countering the bending moment due to gas forces by a bending moment due to centrifugal forces. p_{ts}) axial gas forces; p_{ts}) centrifugal force;

The basic requirement imposed on the design of blade keys consists in the ability of the latter to transmit to the disk the large centrifugal forces due to the turbine blades, with small key weight. This last is very important, since increase in key weight leads to increase in the load on the rim and increase in weight of the disk. This requirement is satisfied by a reasonable distribution of the transmitted force over the surface [volume] of the key, elimination of "nonworking" material in the key, and making sure the design exhibits equal strength. A reduction in abrupt changes in key shape, resulting in the appearance of stress concentrations and leading to a reduction in the magnitude of the nominal stresses permissible in the key, will make itself particularly felt in this case. The attachment should insure precision in blade positioning with the requisite setting angles, and stability of their positioning under operating conditions. Design of the key parts should be technologically feasible: that is, their parts should be simple to manufacture, not requiring a large number of factory operations, and insuring ease of blade replacement under conditions of production and overhaul.

Figure 72 shows methods of attaching turbine rotor blades to a turbine disk.

Attachment of turbine blades by use of welding or soldering has the following advantages: 1) construction of the junction is simple and does not require large industrial outlay in fabrication and mechanical working of the blade shank and disk rim; 2) the design insures good thermal contact between the blades and the disk, thus promoting heat transfer in the disk and lowering the temperature of the blade root section; 3) because of the tight attachment of the blades to the rim, the adjustment of all the blades to the same frequency of natural vibration is simplified.

However, such a method of attachment is justified only with excellent organization of the welding or soldering process, because quality control in the case of welding (soldering) is difficult and requires the availability of complicated apparatus. Another deficiency of this method of attachment is the complexity of the replacement of defective blades. Further, the imposition of the requirement that the materials used for the manufacture of blades and disks exhibit good weldability forces us, in a number of cases, to use expensive materials for the disks, and these materials must be similar in cost and chemical composition to the materials used in the fabrication of the rotor blades. For this reason the attachment of blades by soldering or welding is used little at the present time.

The attachment of blades by use of key connections insures convenience in mounting them on the disk; however, low weight of keying arrangements and necessary strength are possible only with relatively complex key design shapes. In this case the blades can be positioned

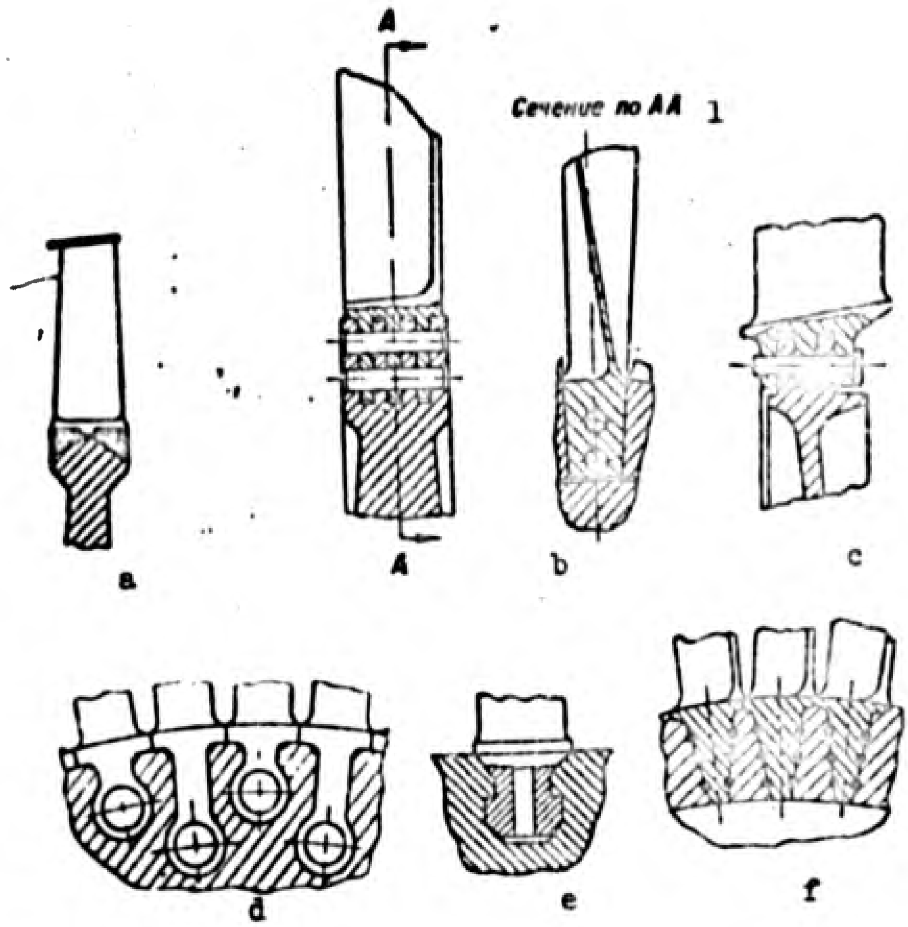


Fig. 72. Methods of turbine-blade attachment. a) Attachment by welding; b and c) key attachment in annular grooves in the rim; d, e, and f) key attachment in transverse grooves in the rim; 1) section along AA

in annular or transverse grooves in the rim.

In fitting the blades into the annular grooves of the rim (see Fig. 72, b), we attach each blade by two smooth dowels, working in shear, in press fit. Beacuse of the comb-like shape of the shank, there are a large number of shear planes on the dowels, as a result of which the stresses in them are relatively small. However, the holes beneath the dowels weaken the top of the rim and the shanks of the blades. In this method of attachment, excellent thermal contact of blades to rim is realized. The rigidity of the attachment also simplifies the adjustment of all the blades to the same natural vibration frequency.

In seating the blades in the annular grooves (Fig. 72, c) each blade may be attached in hinge-fashion to the rim by a single dowel, as a result of which the blade is relieved from the action of bending moments due to gas and inertial mass forces. The hinged attachment also eliminates the most easily excited bending vibrations of the blade. Extensive use of such a method of blade attachment is limited by the considerable loads on the dowels.

An advantage of the method of blade attachment in annular grooves in the rim is that the grooves can be made on a lathe. However, the necessity of providing a sufficient number of grooves leads to an increase in size of the rim and in the weight of the disk.

In case of blade attachment in transverse grooves, the groove may be round, T - shaped, trapezoidal, or some other shape (Fig. 72, d, e, and f).

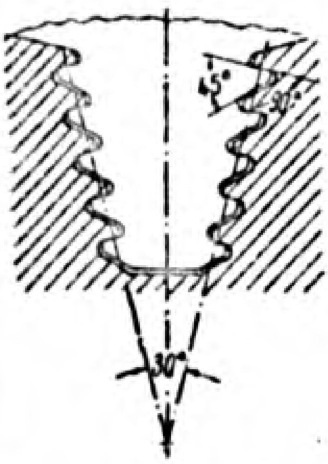


Fig. 73. Fir-tree blade attachment.

Especially widespread at the present time is the use of the so-called fir-tree key (Fig. 73). In this key the trapezoidal shank of the turbine blade and the matching channel in the disk have mating teeth for transmission of forces on their side surfaces. In comparison with other keys, the material is used most efficiently in the fir-tree key for transmission of loads.

The teeth of a key under the action of transmitted centrifugal force, work in shear, crumpling, and bending and, as tests have shown, they are loaded nonuniformly. The first and the last teeth are expecially highly loaded. In the fabrication of the key to equalize the load between the teeth of thekey, it is necessary to hold the necessary clearances in the coupling with high precision. Equalization of the loading between teeth is effected by the plastic deformation of the teeth observed at high temperatures.

Great influence on the magnitude of the stress in the key is also exerted by the shape of the transition to the first tooth, located closer to the shaped part of the blade. To lower the stress concentration, the radius of curvature in the first cavity of a tooth is made 1.5 to 1.8 times larger than in the other cavities.

The fit of the blades in the key may be free (with clearance) or tight (with stress). A fit with clearance is used to decrease tangential thermal forces arising in the rim because of nonuniform temperatures and coefficients of linear expansion of blade and disk materials, and nonuniform distribution of disk temperature along the radius. With clearance in the key, the blade can wobble in a direction perpendicular to the axis of the key.

A loose turbine-blade fit also provides damping of vibrations because of friction in the key, arising as a result of the displacement of the blade under the action of a changing bending moment.

As the disk turns the blade is tightened by the teeth of the shank (under the action of centrifugal force) into the teeth of the rim, so that displacement in the key due to vibration of the blade is substantially reduced.

Because of fabrication tolerances in the manufacturing precision of the keys, the pinching conditions of various blades in a given assembly are not the same. As experiments show, in case the blades are pinched in the upper part of the key, vibration stresses in oscillations are increased because in this case the damping conditions in the key are impaired. If, however, the key is pinched at a lower point, the vibration stresses in the key are reduced. The conditions of pinching the blades in the keys are influenced jointly by other factors, a precise calculation of which is very difficult. Therefore it is possible to have unpredictable increases of vibration stress in various blades of an assembly, and even failure. This situation exists for loose fitting turbine blades; it is not true, however, for blades that are tightly held (with stress).

Current methods of fastening to prevent shifting of the blades along the grooves are shown in Fig. 74. In a rotating turbine, the force of friction prevents shift of the blade along the groove, and in many cases such frictional forces, due to the action of centrifugal force, exceed the lateral force due to the gas flow many times. For example, if the coefficient of friction is equal to 0.2, and the centrifugal force is equal to 7000 kg, the frictional force in the key is equal to 1400 kg, while the component of the gas force along the groove does not exceed 100 kg. This circumstance permits use of relatively simple methods of fixation, i.e., bent plates or by punching (calking) of the blade shanks or the disk.

3. COUPLING OF DISKS TO ONE ANOTHER AND TO THE SHAFT.

Methods of coupling disks to one another and to the shaft vary, and to a certain degree, they have not yet been standardized. A number of structural and manufacturing factors affect the methods of coupling disks to one another and to the shaft, i.e., the number of turbine stages, the magnitude of external loads, methods of cooling turbine parts, conditions of assembly and disassembly of the turbine unit, priority in the utilization of attachment elements whose fabrication techniques have been mastered, etc.

The following loads act on the point of turbine disk coupling to the shaft (Fig. 75):

turbine torque;

bending forces due to weight of disks with blades;

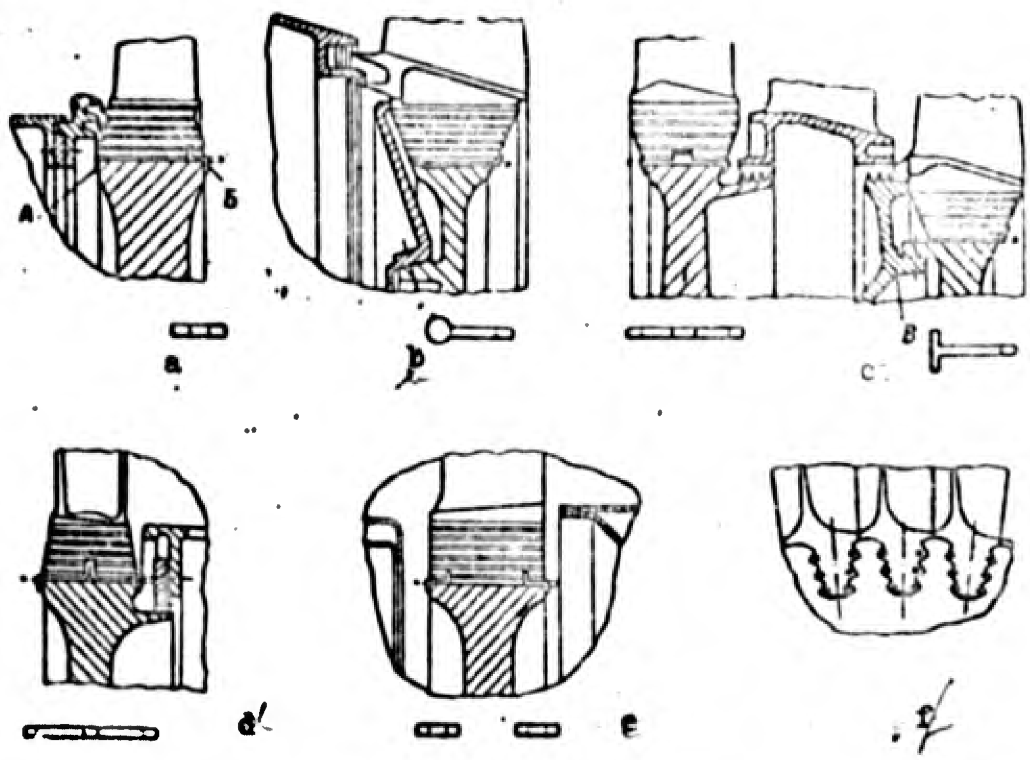


Fig.74. Fastening methods to prevent displacement of turbine blades along the groove. a) with projection A on the blade and with a bent-down plate B; b, d, and e) bent-down plates; c) bent-down plates and a turbine blade stop for the second stage in ring C; f) center punching.

bending inertial forces of rotor mass, originating in aircraft maneuvers;

gyroscopic moment, bending the turbine shaft;

stretching axial force arising in the nozzle diaphragm of the turbine, and on the side surfaces of the disks;

unbalanced centrifugal forces.

Especially unfavorable are the following inertial loads;

1) the regime of maximum gravity stress, corresponding to the aircraft coming out of a dive operating with maximum G - forces at maximum engine speed;

2) the regime of maximum aircraft angular turning rate, for example, the regime of a flat spin (relative to the vertical axis of the aircraft) with the engine working at maximum rpm.

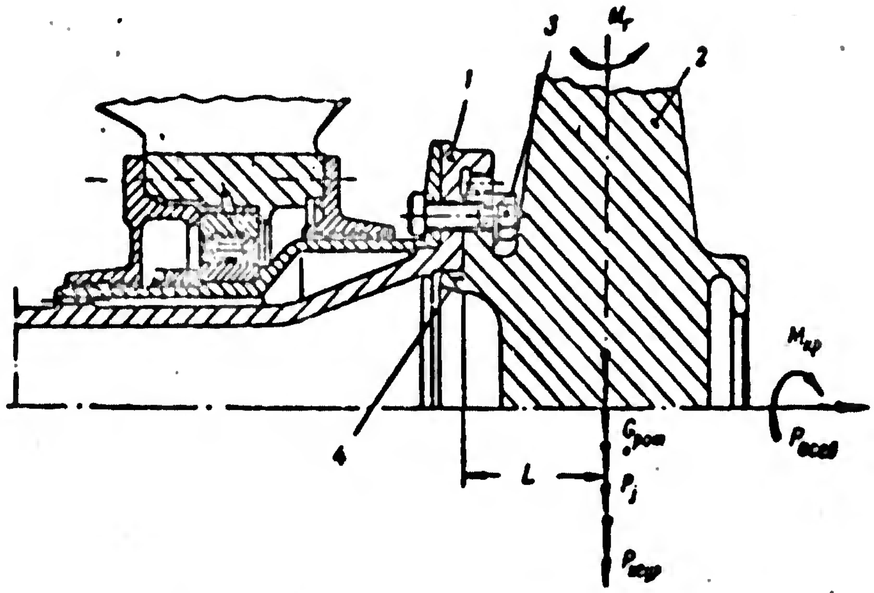


Fig. 75. Loads acting on the point of turbine disk coupling to the shaft. M_{kr}) Turbine torque; G_{rot}) rotor weight; P_j) force of inertia; P_{neur}) unbalanced centrifugal force; P_{osev}) turbine axial force; M_g) gyroscopic moment; 1) shaft flange; 2) turbine disk; 3) bolt; 4) collar.

The coupling of the disks to one another and to the shaft should provide for transmission to the shaft of all the torques generated in the rotor of the turbine during engine operation, without splitting joints or causing interference with the stator by deformation of the rotor. Departure from relative centering and positioning or rotating parts so as to impair the balance of the is completely inadmissible in coupling the disks to the shaft and to one another. Construction elements are therefore provided maintaining the centering of the parts in the hot and cold states. To decrease heating of the turbine shaft bearings the area of contact in the coupling between disk and shaft is reduced.

In the process of fabricating disks and the shaft, great care is taken to insure that the geometrical axis of the shaft is perpendicular to the planes of contact between disks and shaft. This is necessary so as to avoid extra bending moments in couplings.

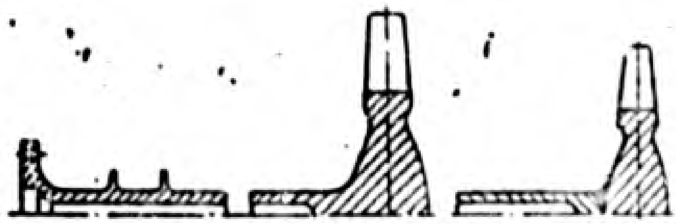


Fig. 76. Integral coupling of disks with the shaft. a) Single-forged rotor; b) welded rotor

Couplings of disk with shaft may be classified as integral or demountable.

Figure 76, a shows an integral turbine rotor whose dishes are made from a forging in one piece with the shaft. Such a rotor has the requisite rigidity at the transition from disk to shaft, and does not require any additional bracing parts. However, the shaft of such a rotor is fabricated from the same material as the disk, which makes its fabrication expensive. Therefore in some designs, the shaft is made of low-alloy steel, and is welded to the disk which is made of steel of high hot strength (Fig. 76, b). Welding provides high rigidity of the coupling; however, it requires careful control of the quality of the weld-seam. The materials of the shaft and disk must have high mutual weldability.

The use of <u>flange attachment</u> in detachable couplings, is especially widespread, that is, attachment by use of flanges or collars located on the shaft and disk. In this type of coupling, the torque and axial forces are transmitted from the disk to the shaft in various ways. The torque may be transmitted (Fig. 77, a) by the moment of frictional force which is provided at the contact surface as a result of the tightening of screws 1. The screws work only in tension under the action of the force due to initial tightening and the axial force acting on the disk. In order to obtain the necessary moment of frictional force, a considerable initial tightening of the bolts is required, or a large flange radius. Therefore, in order to reduce the force of the initial tightening, the flange is substantially extended, and this leads to increase in the heat transfer from disk to shaft, and further on to the bearing. These shortcomings are substantially reduced in the design shown in Fig. 77, b, because of the use of precisely fitted bushing 13, transmitting the torque.

A shortcoming of this design is the weakening of the disks because of the holes. This deficiency is removed in the flange coupling (see Fig. 75) in which the turbine disk 2 and the flange 1 of the shaft have evolute grooves which transmit the torque. The bolts 3 of the coupling work only in tension. Centering of the parts in the hot condition is maintained due to the fact that the centering collar 4 on the disk (the hotter part) is inserted into a boring in the shaft flange (the cooler part). On the flange of the disk there are flutes decreasing the flow of heat to the shaft.

Simpler from the technological standpoint are the flange doweled couplings. In these designs the disks are located on the shaft flanges by tightening, and at the same time by attachment, using axial (Fig. 78, a) or radial dowels (Fig. 78, b). In the first case, the torque is transmitted by six smooth axial dowels, and the axial force, by six threaded axial dowels. It should be noted that the axial location of dowels does not maintain centering in the hot operating condition, when the tightness between flanges of disk and shaft is relieved. In the second method of attachment, that is , with radial position of the dowels, the centering of the parts is maintained in the hot condition, and transmission of torque and axial force is accom-

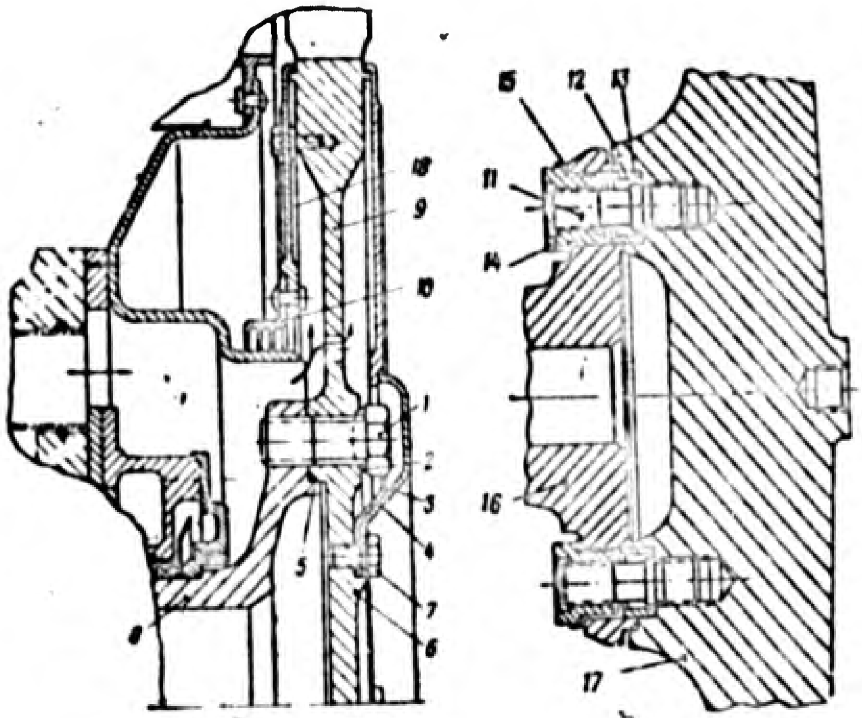


Fig. 77. Flange attachment of the turbine disk to the shaft. a) By use of screws; b) by use of pegs and bushings; 1) screws (6 of them); 2) spherical washer; 3) lock plate; 4) screw cover; 5) centering collar of the shaft flange; 6) collar for centering the lock plate and cover; 7) attachment screws; 8 and 16) shafts; 9 and 17) turbine disk; 10) throttling labyrinth; 11) pegs (6 of them); 12) centering collar; 13) bushings; 14) nuts; 15) lock; 18) deflector.

plished by the same dowels, working in shear.

In rotor designs of modern gas turbines, another method used for attachment of disks to shaft is by means of a grooved sleeve (Fig. 79, a and b). In such designs the reciprocal centering of shaft, disk, and sleeve may be effected by use of cylindrical collars, bushings, or centering cones, in which compression is provided by tightening of a coupling nut. There are also designs in which the turbine disk is mounted on the shaft by use of the so-called heat-resistant trapezoidal grooves with the disks centered by the side surfaces of the grooves, similar to analogous couplings in axial compressors.

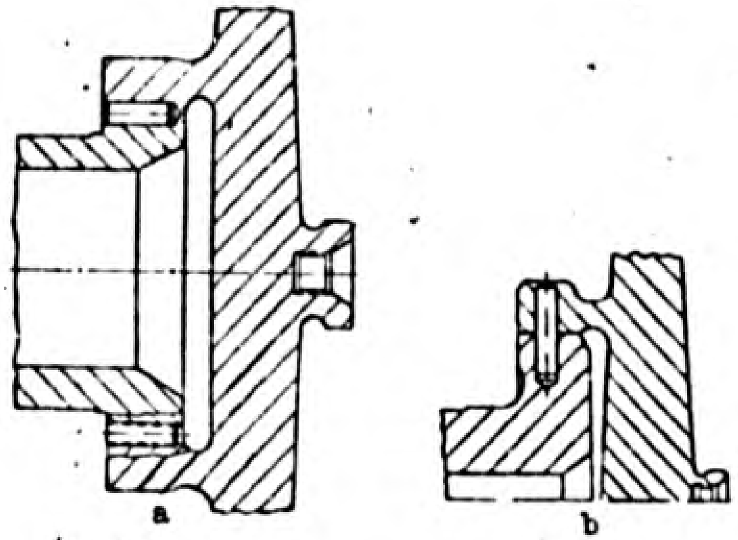


Fig. 78. Flange attachment of turbine disk to the shaft by use of dowels. a) Attachment by axial dowels (6 smooth and 6 threaded); b) attachment by radial dowels.

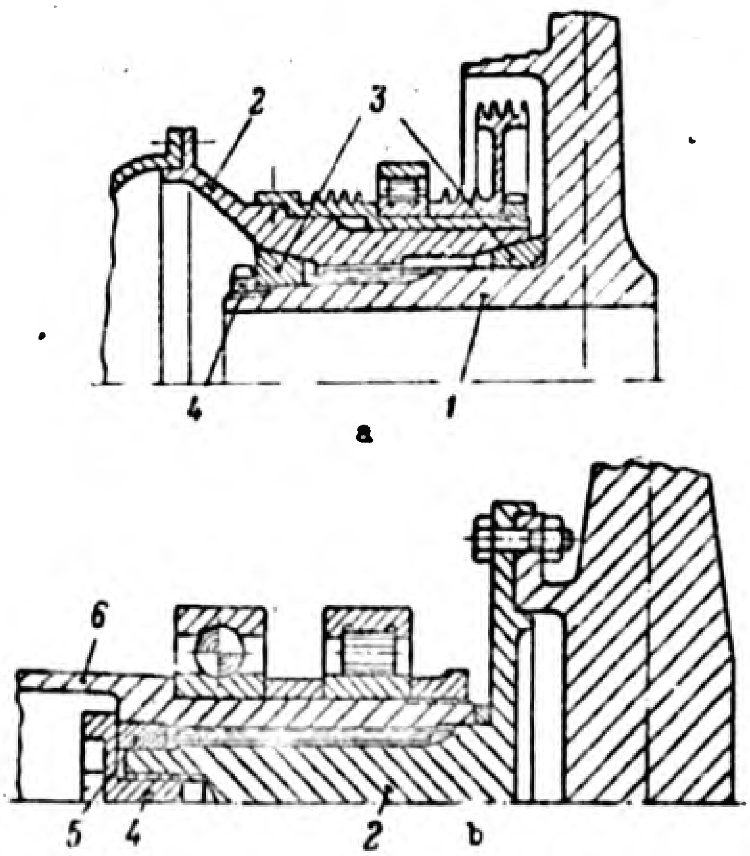


Fig. 79. Attachment of turbine disk to shaft by use of grooved sleeves. a) Centering by cone; b) centering by a cylindrical collar; 1) disk; 2) sleeve; 3) cones; 4) nut; 5) cylindrical ring; 6) shaft.

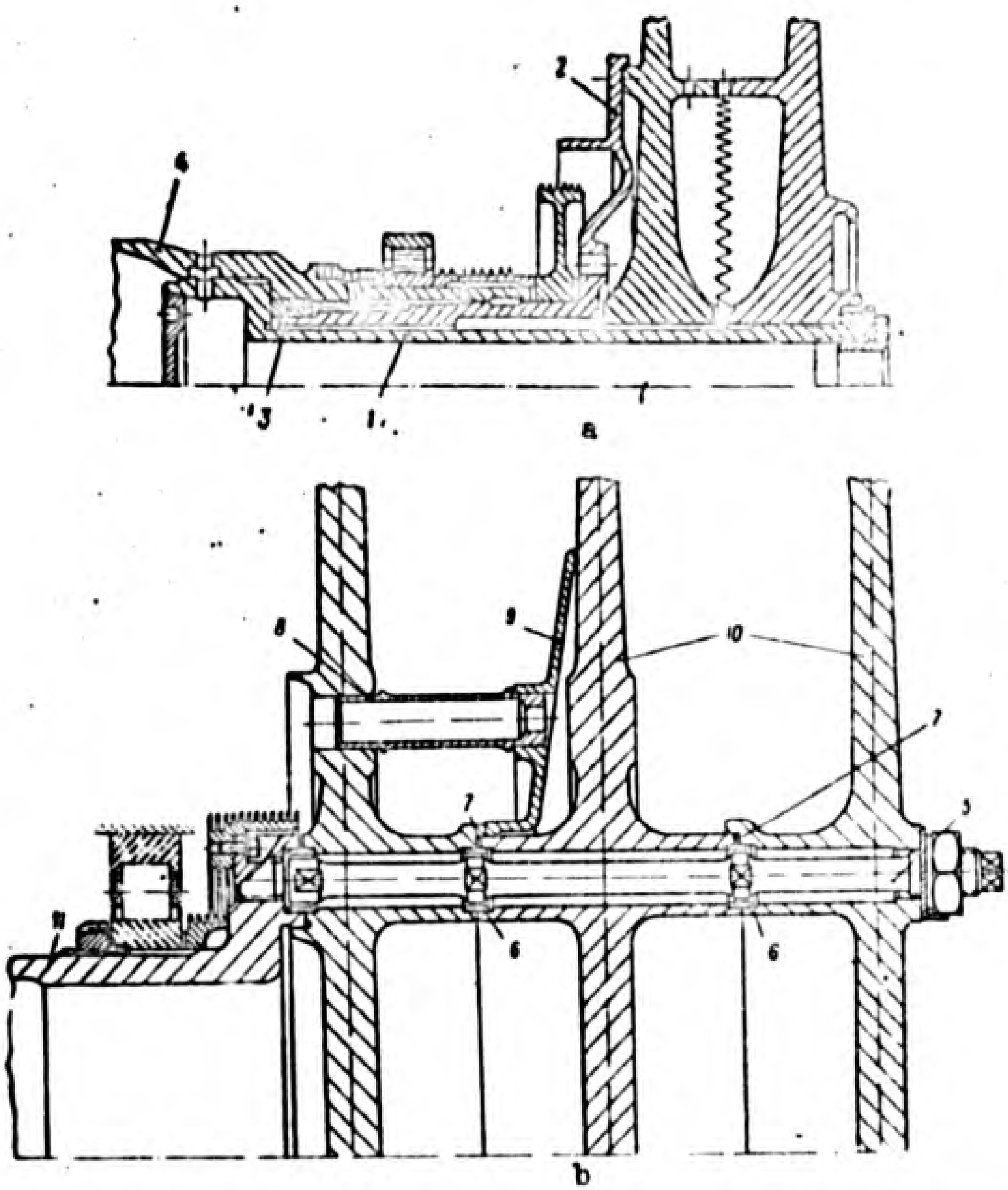


Fig. 80. Attachment of disks of multistage turbines. a) Attachment by use of a central tie bolt and triangular face grooves; b) attachment by use of pegs and precisely fitted bushings; 1) tie bolt; 2) flange of the grooved sleeve; 3) centering ring; 4 and 11) turbine shaft; 5) attachment pegs of disks to shaft; 6) bushings; 7) washers for regulating axial clearance between rotors and nozzle diaphragms; 8) tube for supply of air; 9) deflector; 10) turbine disks.

Figure 80 shows several demountable couplings for disks of multistage turbines. Coupling may be effected (see Fig. 80, a) by use of a central tie bolt and triangular face grooves, permitting free radial expansion of disks while maintaining their reciprocal centering. For effecting separable coupling of disks elongated pegs (see Fig. 80,c) may also be used, and precisely fitted bushings, transmitting torque.

Separable attachments simplify assembly of turbine units; however, balancing of the rotor on overhaul of the turbine is not retained unchanged.

The force of the initial tightening of the central tie bolt or elongated dowels should insure the tightness of the joint in the most unfavorable regimes of rotor operation. Such regimes include the pulling out of the aircraft from a dive with great G - forces and the rapid cooling of the turbine rotor, for example, in flight with the engine shut down, when a large mass of cold air flows through the engine. Under these conditions the disks are cooled faster than the central tie bolt or elongated coupling pegs.

The difference between the temperature of the disks and that of the bolt, reaching 200 - 350° C under steady conditions, brings about the appearance of a substantial force acting on the tie bolt. For reducing its magnitude, the tightness of the coupling is reduced by introduction in it of spring elements. In Fig. 80, a, the flange 2 of the slotted sleeve is such an element.

The turbine shown in Fig. 63 has a nondemountable coupling of disks. Coupling of disks is carried out by use of a stressed ring 6, the diameter of which is less than the outside diameter of the rim of the disk, in order to decrease the stress in the ring arising by reason of centrifugal forces.

Elimination of unbalance in the dynamic balancing of an assembled rotor is carried out by rearrangement of the blades, removal from the rim of part of the material by use of a grinding wheel, and most frequently of all, by use of balancing weights (this latter simplifies the balancing process, reduces the amount of blade rearrangement on the disk, and shortens the time for dynamic balancing).

Maximum permissible unbalance amounts to 10 - 50 g-cm.

Chapter 5

NOZZLE DIAPHRAGMS AND GAS-TURBINE HOUSINGS

The fundamental elements of the nozzle diaphragm are the nozzle blades forming an annular grid, and the shrouds forming the outer and inner walls of the annular gas passage. The shrouds are also used for blade attachment.

The turbine housing consists ordinarily of the housing for the turbine-shaft bearings and the attachment housing for the nozzle diaphragms.

1. DESIGN SHAPE OF THE NOZZLE BLADES.

The geometrical shape of the nozzle blades is characterized by values of the same parameters as were used for turbine blades (Fig. 81).

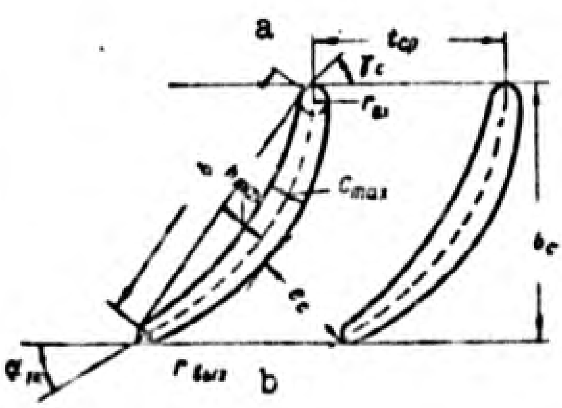


Fig. 81. Geometrical parameters of nozzle diaphragms. α_{1k}) Design angle of the gas flow outlet; e_c) minimum channel width (see other designations in Fig. 66).

The length of the blades and their mean diameter are determined on the basis of the gasdynamics calculation for the conditions that will insure passage of the specified quantity of gas.

The length of the nozzle blade is set about 1-2% less than the length of the turbine blade, so that there will be no gas shock at the outlet from the nozzle diaphragm by incidence of the gases on the turbine disk rim.

The aspect ration of the nozzle blades is equal to 1.8-2.2, that is, less than for the turbine blades. Consequently, for the same length of nozzle and turbine blades, the width of the former is larger, which is required by the necessity for turning the flow in the nozzle diaphragm through a larger angle than is required for the turbine wheel (rotor).

The ratio of chord to blade pitch in current designs of turbines is equal to 1.3-2.0. It is chosen on the basis of the results of wind-tunnel tests, from the condition of insuring the minimum hydraulic loss of energy in the grid. It is desirable that the number of nozzle blades should not be a multiple of the number of turbine blades. If the nozzle blades induce harmful resonance vibrations in the turbine blades, one of the methods of eliminating the vibrations is a change in the number of nozzle blades.

The profile of the lateral cross section is set constant with height along the blade, while the concave and convex surfaces are described by particular simple conjugate lines – arcs of circles and straight line segments. The greatest thickness of the profile amounts to 10-20% of the chord. The maximum curvature of the mean line is located in the fore part of the profile, approximately at a distance of 25-35% of the chord from the leading edge of the profile, and the thickness decreases steadily in the direction of the trailing edge with a pattern such that the fundamental turning of the flow toward the required angle should be accomplished in the forward part of the interblade channel, and the flow should be stable in the outlet end. The radii of curvature of the leading and trailing edges are, respectively, equal to 0.8-2.0 and 0.4-0.6 mm (VK-1). With decrease in radius of curvature, the hydraulic losses also decrease; however, the thermal stresses also increase in the nonsteady conditions of engine operation, where the temperature of the thin edges of the blades are changing more rapidly than that in their central parts.

For reduction of weight and thermal stress, cast blades of the nozzle diaphragms are very frequently made hollow.

Blades of the nozzle diaphragm, set in a grid, form a converging channel, and are not sensitive to departure of the operational regime from the design [rated] regime.

The design angle at outlet of the flow amounts to 20-30° in the first stages at the mean diameter. In the later stages, it increases to 50°, in order to decrease the height of the blades. With height, the angle of the blades may be kept constant, or it may change in conformity with the previously selected rule of shaping of the turbine blades with height.

The minimum flow cross section of the nozzle diaphragm is carefully controlled during fabrication by measuring the width of the channel at several points at the outlet from the nozzle diaphragm. The tolerance on the flow cross-section amounts to 0.5-1.5%. In case this limit is exceeded, an adjustment of the flow cross section is carried out by turning the blades, trimming part of the material at the trailing edge of the blades, or by replacing them.

In the hot state, because of the thermal deformation of the nozzle diaphragm, the minimum flow cross section is increased by 2-2.5%. Thermal deformation is caused by change in the gas temperature upon change in the operational regime of the engine, altitude, and speed of flight.

Nozzle blades are fabricated by the milling of special forgings, stamping of thinwalled steel plates, or most frequently, by precision casting. Precision casting permits obtaining sufficiently precise profiles with small allowances (of the order of 0.2 to 0.5 mm) for final machining (grinding and polishing). To increase resistance to heat, the turbine nozzle blades may be coated with a heat-resistant enamel or aluminum coating.

2. ATTACHMENT METHODS FOR NOZZLE BLADES AND EQUIPMENT

In the attachment of nozzle blades, the following fundamental requirements are imposed:

1) provision for rigidity, making certain that in the operation of the engine the design angle of the gas flow exit is constant, since that quantity effects the efficiency, turbine power, and gas flow rate through the nozzle diaphragm;

2) the attachment should insure free and unrestrained thermal expansion of the blades relative to the coupled parts thus avoiding the appearance of large thermal stresses and buckling of the structure, arising as a result of the temperature difference between the blades and shrouds, and differences in the coefficients of linear expansion of the materials of which they are made;

3) replaceability of defective blades in fabrication and in operation should be simple and convenient.

Because of the indicated requirements, the nozzle blades are not as a rule included in the supporting structure of the turbine housing as structural elements stiffening the housing.

The nozzle blades are located and attached between inner and outer shrouds, which in turn are attached to the housings of the nozzle diaphragms. In the nozzle diaphragms of some engines, the shrouds as constructional elements are missing, and their place is filled by the housings of the nozzle diaphragms.

The following fundamental arrangements of the attachment of nozzle blades are differentiated: doubly supported and cantilevered.

Doubly supported blades are used in the nozzle diaphragms of the first stage of gas turbines. Cantilevered attachment of blades is most often encountered in the nozzle diaphragms of intermediate stages.

In doubly supported attachment, the blades may exhibit:

rigid fastening in both of the shrouds; in this case, one of the shrouds has a split between each 4-6 blades, which reduces the resistance of the band to radial thermal expansion of the blades;

rigid fastening in one of the shrouds and a flexible support in the other;

flexible support in both shrouds.

Coupling of the nozzle blades with the shrouds may be classified as integral or demountable. Nozzle diaphragms with integral attachment may be fabricated by casting, welding, or with riveted blades.

For fabrication by casting, the method used for nozzle diaphragms is precision casting. The blades and rims are ordinarily cast in one piece, in sections of 2-4 blades each, or in the form of an entire casting. In the latter case, in order to decrease the thermal stresses, the inner shroud is cut at intervals between the blades at several places. The advantage of such design is its great rigidity. Its deficiencies include the difficulty of polishing the blades and obtaining a high quality of finished surface; the presence of thermal and residual stresses in the parts of the assembly; the difficulty of overhayl. In the light of these deficiencies; the casting of nozzle diaphragms is not used widely.

In welded nozzle diaphragms, the blades are welded to both shrouds 82,a one of which (the upper one) has thermal cuts, or to one of the shrouds, forming with the other a flexible coupling. Thus, for example, Fig. 82,b shows a design of nozzle diaphragm the blades of which are welded to the inner shroud.The blades fit with clearance into the shaped slots in the outer band. The rigidity of the shrouds is increased by use of cross-sections of box-like form.

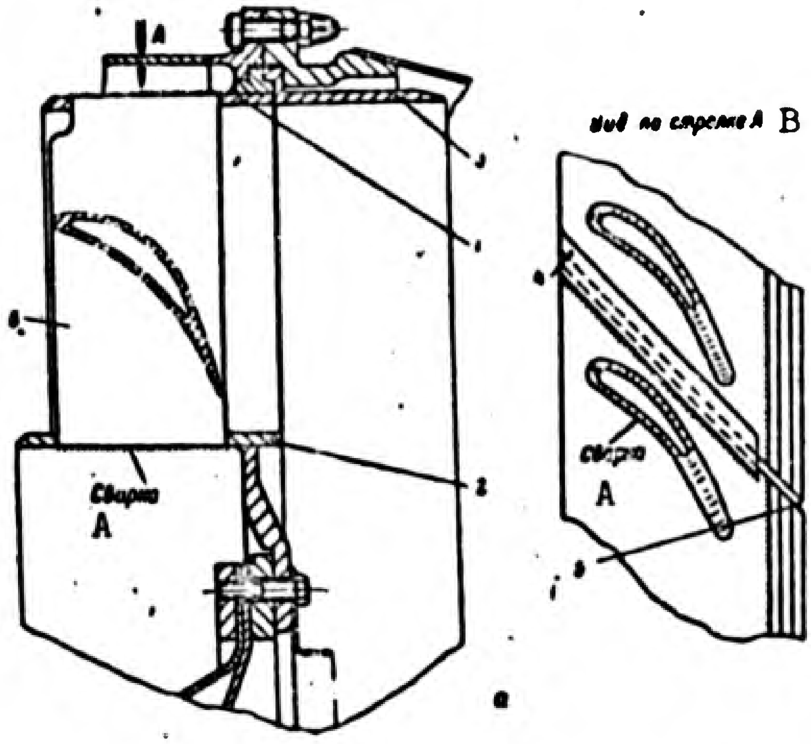
The design of nozzle diaphragms with welded blades is simple; however, in fabrication, great difficulty arises in the welding of a large number of blades. Also in such a method of blade attachment, the precise placing of the blading before completing the weld is difficult;and further, in the welding process, it is possible to buckle the blade and ruin the shape of the flow passage. Therefore, the welding of blades must be carried out in special fixtures.Replacement of defective blades is completely impossible in practice.

Technologically simpler is the attachment of blades to the shroud by riveting. Figure 82,c shows a nozzle diaphragm with blades riveted to the inner shroud by use of two projections. The projections from opposite ends of the blades project freely into grooves of the outer shroud, which holds the blades in a tangential direction. Between the blade ends and the outer shroud a radial clearance is established, and controlled during assembly.

The fundamental advantage of the integral couplings is the relatively small number of manufacturing operations in the machining of the attachment parts. Among the deficiencies one should mention the difficulty of replacing defective blades on overhaul.

In demountable couplings (with detachable blades) attachment of blades may be carried out in three ways as follows: 1) in grooves in the outer and inner shrouds; 2) by use of shoes attached by screws; and 3) in shaped recesses in the shrouds.

In the attachment of blades in grooves (Fig. 83,a) each blade has on its ends flanges which fit into slanted grooves located in the shrouds. The grooves are arranged at various angles to the engine axis. Protuberances on the inner flanges, bearing on the edge of the housing of the nozzle diaphragm, hold the blades, preventing their radial displacement. In assembly, the blades are attached to the inner shroud



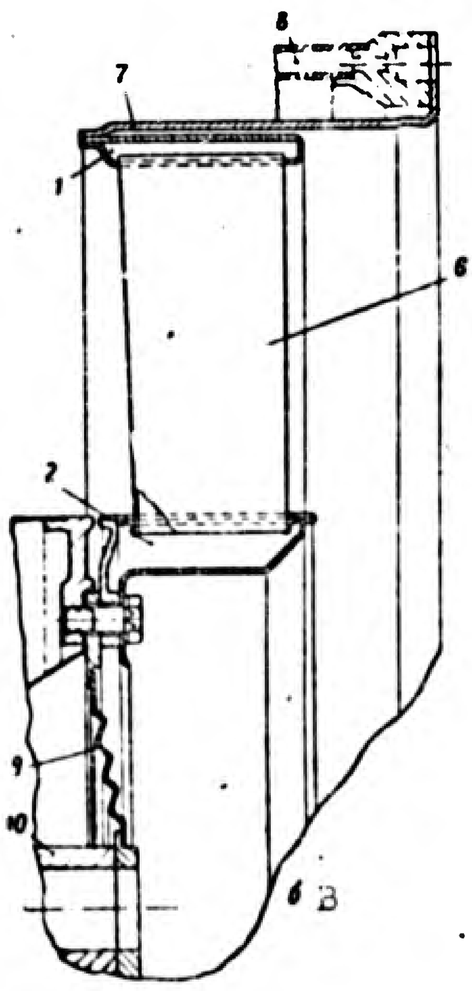


Fig. 82. Designs of non-demountable nozzle diaphragms. a) With blades welded to the outer and inner shrouds; b) with blades welded only to the inner shroud; c) with nozzle blades riveted to the inner shroud. 1) Outer shroud; 2) inner shroud; 3) ring; 4) cover; 5) thermal compensator (slit); 6) nozzle blade; 7) elastic shroud; 8) outer combustion chamber housing; 9) corrugated diaphragm; 10) turbine shaft bearing housing. A) Weld; B) view along arrow A.

after which the outer shroud is put on, shifting it simultaneously in the axial and circumferential directions. Between the outer projection and the outer shroud a radial clearance is established, insuring free expansion of the blade on heating.

Figure 83,b shows the attachment of nozzle blades by use of shoes, each of which has ahaped cuts to fit the shape, on the one hand, of the back of the blade, and on the other, of the concave side of the blade. In this way, two neighboring shoes form a shaped groove conforming to the blade profile, into which one end of a blade can be placed. The other end of the blade goes into an analogous groove formed by two shoes at the opposite end of the blade. Each shoe is attached by two screws, of which one is a lock screw.

The attachment of blades in shaped recesses in the shroud is shown in Fig. 83,c. The blades are prevented from moving radially by retaining ring 12. The shrouds are reinforced at the cuts by cuffs and profiles. The inner shell 18 forms the inner contour of the gas-flow passage and protects the blade attachments from the action of high temperatures. The outer shell 19 protects the outer shroud preventing the flow of secondary air.

Removable blades with cantilevered attachment have flanges at their tips to provide a stable position for the blades during operation (Fig. 84). The rigidity of the cantilevered blades is increased by installations (at the inner ends of the blades) of removable flanges 1, holding the blades in pairs.

Nozzle diaphragms with demountable blades have the following advantages:

the possibility of excellent surface finishing and shaping with adequate precision;

less thermal stress occurring in the nozzle diaphragm because of the temperature differences between the blades and shrouds;

the capacity of rapid replacement of a defective blade on overhaul.

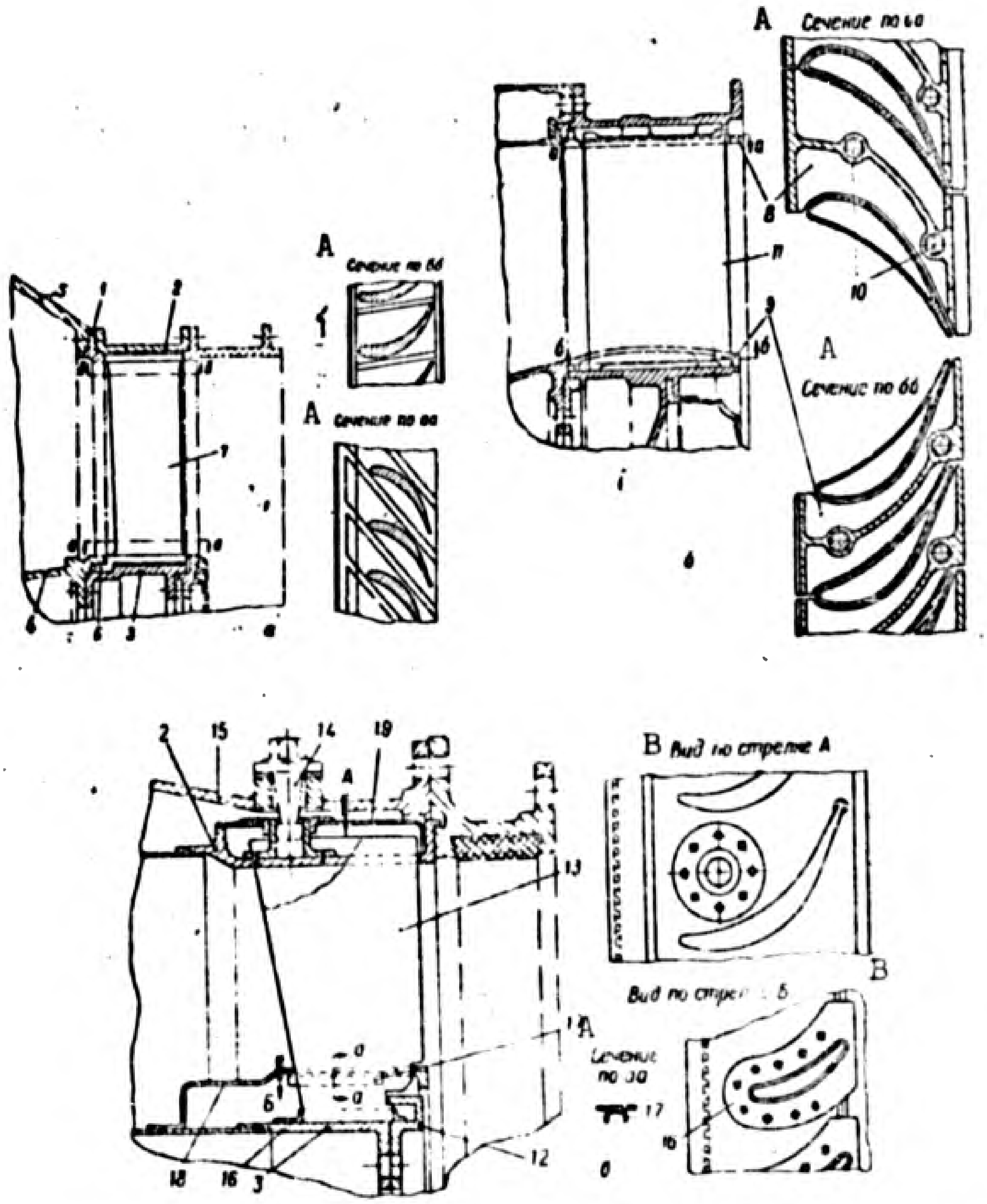


Fig. 83. Methods of demountable attachment of nozzle blades. a) In grooves of the outer and inner shrouds; b) by means of attached shoes; c) in shaped recesses in the shrouds. 1) Intermediate centering ring; 2) outer shroud; 3) inner shroud; 4) attachment ring of the inner shroud; 5) attachment ring of the outer shroud; 6) locative projection; 7, 11 and 13) nozzle blades; 8) outer shoe; 9) inner shoe; 10) screws for attaching shoes; 12) locating ring; 14) radial dowels; 15) outer liner of the combustion chamber; 16) cuffs; 17) cover; 18) inner shell; 19) outer shell. A) Section along; B) view along the arrow.

Because of this, nozzle diaphragms with demountable blades are at present used especially extensively.

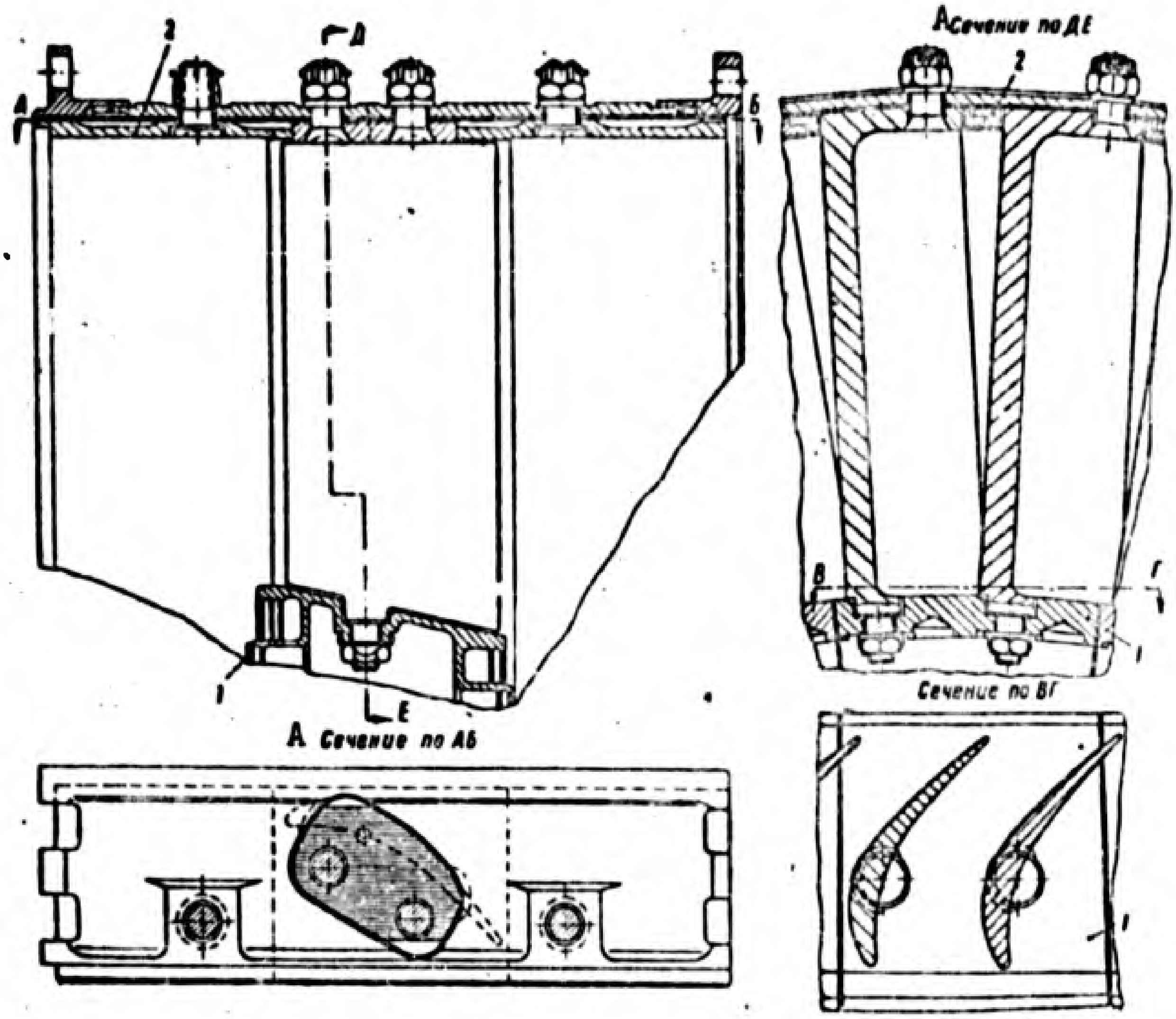


Fig. 84. Cantilevered attachment of nozzle blades. 1) Removable flange; 2) shoe. A) Section along.

The attachment key of a nozzle diaphragm absorbs and transmits the twisting moment arising in the nozzle diaphragm, axial forces arising by reason of the pressure difference upstream and downstream of the nozzle diaphragm, and by reason of the acceleration of the gas flow, as well as the gravity and inertial forces due to the mass of the nozzle diaphragm.

The nozzle diaphragms are attached to the nozzle-diaphragm housing which in turn is attached to support elements which are part of the system of supporting housings of the engine.

The structural support elements to which the nozzle-diaphragm housings are attached are include the turbine shaft bearing housing and the outer liner of the combustion chamber.

One of the methods of attaching a nozzle diaphragm to the turbine shaft bearing housing is shown in Fig. 82. A defect in such a method of attaching the nozzle diaphragm is that the bearing housing is subject to the entire load that acts on the nozzle diaphragm. Therefore the nozzle diaphragm in a number of designs is also attached to the combustion chamber liner. Thus, for example, in the design shown in Fig. 82,b, outer shroud 1 of the nozzle diaphragm is attached to outer liner 8 of the combustion chamber by use of shroud 7 (the outer housing of the nozzle diaphragm), and inner shroud 2 is attached by use of corrugated membrane 9 to housing 10 of the turbine shaft bearing. Loads from the nozzle diaphragm are in large part absorbed by the combustion chamber shell which exhibits greater rigidity with respect to bending and twisting than the bearing housing. The corrugated membrane does not interfere with the fhifting of the aluminum bearing housing as a result of thermal expansion relative to the steel combustion-chamber shell, so that the axial position of the nozzle diaphragm is maintained. The outer housing of the nozzle diaphragm 7 is made elastic so as to permit the outer shroud of the nozzle diaphragm, not attached to the blades, to expand radially on being heated.

In another design (see Fig. 83,c) part of the axial force and torque is transmitted to outer housing 15 of the combustion chamber by radial dowels 14. The radial dowels do not interfere with the free thermal expansion in a radial direction of outer shroud 2 of the nozzle diaphragm.

3. GAS-TURBINE HOUSINGS

The housings of gas-turbines are subject to loads from the rotor and nozzle diaphragms, and transmit them to the neighboring housings which is a part of the structural support system of the engine housings.

The basic requirement demanded of a housing is high rigidity in bending, reliably preventing contact between rotor and stator on deformation, and reduced tendency for harmful vibrations to appear in the rotor shafts.

The design geometries of the turbine housings depend on the arrangements of rotor support points, the method of attachment of the nozzle diaphragms, and the method chosen for fabrication of the housing.

The cantilevered arrangement of rotor disks is differentiated from the arrangement of disks with an end support. In the first case, the turbine rotor support is located forward of the disks, in the forward turbine bearing housing (Fig. 85). In the second case the support is located behing the disks, in the rear turbine housing (Fig. 86). A cantilevered arrangement of disks turns out to be difficult if the number of disks is greater than three, and the bending rigidity of the casing and rotor is inadequate.

In most designs the forward rotor bearing housings are traced to the housing of the compressor rotor bearing and do not project through the engine flow passage (see Fig. 85). The designs are reasonably simple. However, designs are well known in which the housing of the forward rotor bearing is located inside the engine flow passage. Such a housing in comparison with one having cantilevered position of the disks, is more rigid; however, it is necessary to provide protection from the action of hot gases. An analogours problem arises also in case the rear bearing housing is flushed by hot gases. The structural support elements of the housing may be protected by double or by sinlge screens, between which air flows; or else a screen is mounted between the walls of the housing for thermal insulation.

In order to increase the bending rigidity of the cantilevered housing and to decrease its weight, part of the radial loading from

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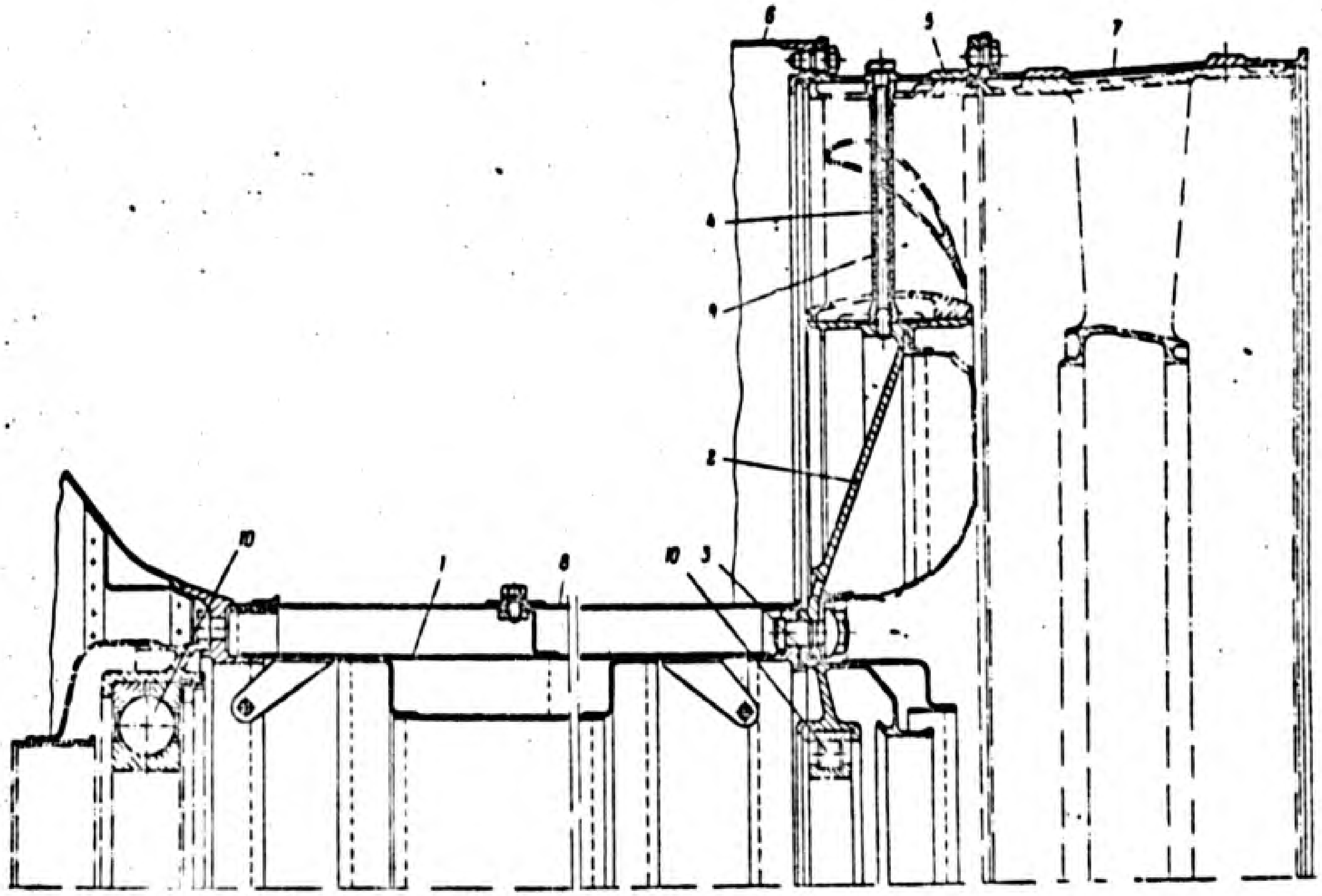

Fig. 85. Gas-turbine housing with support [stress-hearing] coupling between the shrouds of the first-stage nozzle diaphragm. 1) Turbine shaft bearing housing; 2) membrane of the inner shroud of the nozzle diaphragm; 3) membrane of the turbine shaft bearing housing; 4) support coupling screws; 5) outer shroud of the first-stage nozzle diaphragm; 6) outer combustion chamber shell; 7) outer shroud of the second-stage nozzle diaphragm; 8) heat reflecting screen; 9) bushing brace; 10) rotor bearigs.

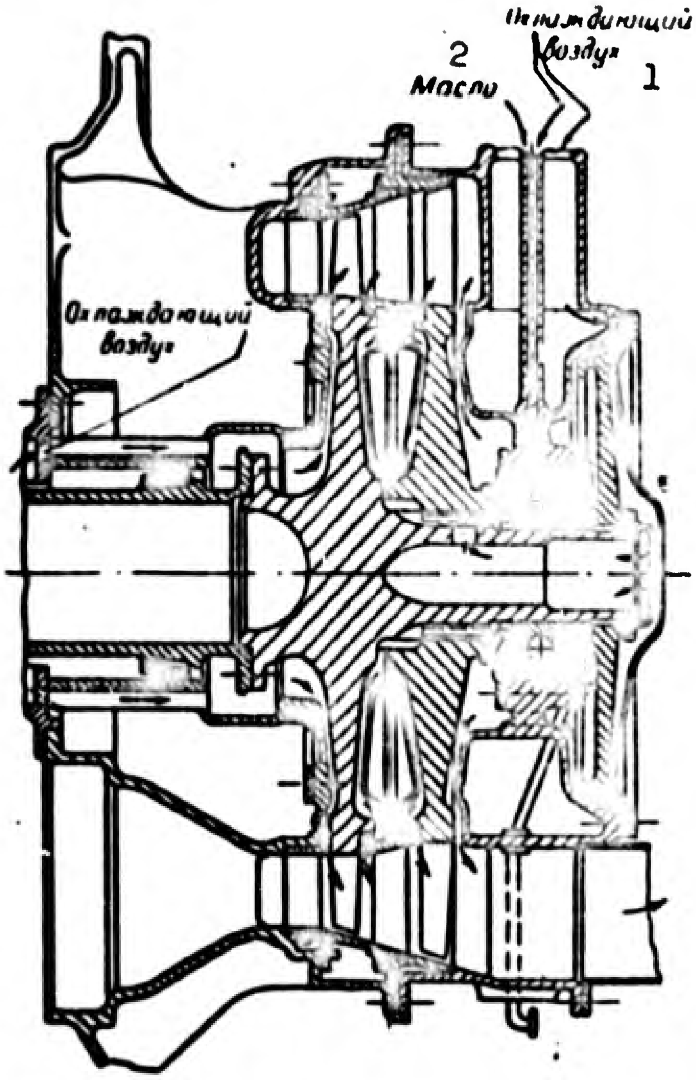

Fig. 86. Two-stage gas turbine with rotor bearing positioned behind turbine disks. 1) Cooling air; 2) oil.

the rotor is transmitted to the outer stress-bearing engine housing by means of screws 4 (see Fig. 85). Increase in rigidity of the housing without substantial increase of the weight is provided by longitudinal or transverse strengthening ribs, made integrally with the housing or welded to it.

Housings are centered with reference to one another by use of collars. Buckling of the housings when they are heated is prevented by choice of materials with proper values of the coefficient of linear expansion, and by providing for their cooling. In case of large differences in the values of the coefficients of linear expansion of the housing materials, leading to intolerable relative deformations, coupled housings may be separated by parts with intermediate values of the coefficient of linear expansion (for example, ring 1, see Fig. 83,a), or the holes for the fastening bolts may be made elliptical with the larger axis disposed in the radial direction.

Housings of aluminum alloys have only limited application because of their large weight and insufficient rigidity at high tempertures.

Casings of gas turbines may be fabricated by casting in molds or by centrifugal casting with subsequent machining, by the method of rolling, welding of steel sheets, and stamping.

The precise location of abutting surfaces is achieved in the machining of the housings. The required coaxiality of holes for the turbine shaft bearings are obtained by machining the holes all at one time, with the casing set up in the machine.

4. RADIAL AND AXIAL CLEARANCES AND SEALS IN THE TURBINE

Radial and axial clearances insuring freedom from contact between the rotor and stator in all turbine operating regimes are established between the turning part of the turbine (rotor) and the stationary part (stator).

Especially important are the radial clearances between the turbine blades and the turbine housing, and to a great extent the energy losses of the gas jet and, in the final analysis, in engine efficiency depend on the magnitude of these clearances. Energy losses in the gas jet are due to flow of gas the radial clearance from the concave side of the blade to the convex side, and in the axial direction, from the region of high pressure ahead of the blades to the region of low pressure behind them.

Differences in the conditions of heating and cooling the housings, disks and turbine blades under different operational conditions, lead to considerable changes in the magnitude of radial clearances.

Thus, for example, at start-up of the engine, the housings of the nozzle diaphragm are heated more than the disk, so that the initial radial clearance is increased. In proportion to the heating of the disk and turbine blades, the clearance is somewhat reduced and at shut-down of the engine, when the cooling of the housings takes place faster than that of the disk, the clearance decreases and may become less than it was originally. Therefore the magnitude of the initial radial clearances in the cold state of the engine (the "cold" clearance) is set so that at engine shut-down there will be no contact of the rotating parts with the stationary parts, and so that at start-up, the "hot" clearance will be as small as possible. In this case we must bear in mind the influence of the magnitude of radial clearance in a bearing,and the elastic and residual deformation of the turbine blades and the disk, due to the centrifugal force acting on their intrinsic mass; we must also bear in mind the following: elastic deformation of the housing by reason of the action of internal pressure; contraction of the material of the turbine housing due to periodic heating and cooling off, and also the deformation of the housings and rotor shafts on overload and wobble caused by factory tolerances, on the precision of fabrication of linked parts of the rotor and housing.

In modern designs of turbines the minimum radial "cold" clearance amounts to 1.5-3% of the blade length.

In order to decrease losses, intense cooling of the housings and the turbine disk rim is employed in changing the radial clearance, and also the appropriate materials are chosen for the required coefficients of linear expansion and small shrinkage (for turbine housings).

The housing may be cooled by air from the inside or the outside.

The air for cooling the housing from the inside is brought from the combustion chamber and mixed with the gas flow flushing the inner surface of the housings. In outside cooling, the air is led from an intermediate stage of the axial compressor or from the atmosphere (under dynamic pressure)beneath the metal screen in which the housing is enclosed. With such methods of cooling the housing, in order to lower substantially its temperature and noticeably decrease the magnitude of range of variation in radial clearance, a large amount of air is required. With gas temperature in front of the turbine at 850-900°C, the housing temperature is equal to about 700°C. A considerable lowering of the housing temperature (to 350°C) with reduced expenditure of air is achieved by installation of shoes 2 (see Fig. 84) on the inner surface of the housing, forming with the housing a cavity for air flow.

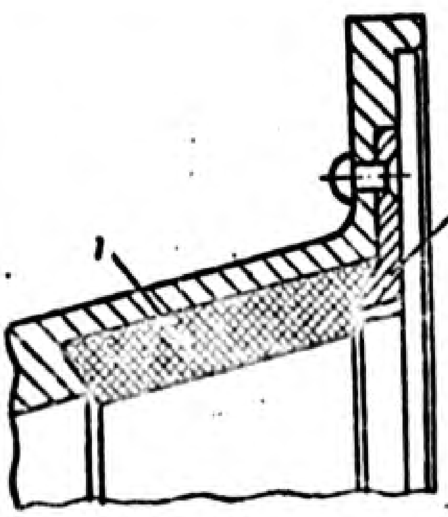

Fig. 87. Insert in the turbine casing. 1) Insert; 2) key.

An exact analysis of all the factors that influence the magnitude of radial clearance is very arduous. Therefore in order to prevent damage to the turbine in case of contact between the housing and the turbine blades, sometimes [metalloceramic] inserts sintered from metallic powder are installed inside the housing (Fig. 87). The material of the insert has sufficient strength to permit the blades, in case of contact with the casing, to cut away enough material to insure a minimum clearance. The inserts are put in through special slots in the circumferential grooves of the housing, and these are "dovetailed"; after installation they are bored out on their inner surface in order to provide the necessary radial clearance in the turbine. The slots are closed by keys 2, riveted to the housing.

The choice of the final magnitude of radial clearance in the process of turbine adjustment is very easy to accomplish by use of easily removable fixed ring 3 (see Fig. 82,a) installed over the turbine blades. Through proper choice of material for the ring and the requisite inside diameter, the minimum permissible clearance may be established to guarantee reliability of turbine operation. Decrease in radial clearance as a result of application of the methods considered, substantially lowers the specific fuel consumption (up to 2-3%). Effective methods for decrease flow of gas through the radial clearance include the use of labyrinths and shrouding for the turbine blades (Fig. 88).

Shrouding the turbine blades decreases the leakage of gases at the blade tips from the front to back, and decreases the level of vibrational stress in the blades, which permits the use of a thin profile providing high efficiency. The additional weight of the shrouding in this case is made up for by reduction in the height of the blade fin. Shrouding also evens out the vertical flow distribution behind the turbine blades, particularly in the upper part of the blade. The efficiency of a turbine with shrouding of blades is 1.5-2% higher than that of a turbine without shrouding, and when labyrinths are also used, the turbine efficiency increases up to 2.5-3%.

Ordinarily shrouds are found only in the first stages of the turbine, where the height of the blades is less than in the last stages, and the relative effect of radial clearance is greater. Most usually the shrouds are formed by the flanges of the blades on assembly. An unbroken ring is not pressed around the tips of the blades as a shroud, in view of the large circumferential stresses that arise shrouds during turbine operation. Particular attention is devoted to the proper placement of shrouds relative to the nozzle diaphragm, to prevent the projection of the shroud into the gas flow and the consequent spoilage of the smoothness of the turbine flow passage.

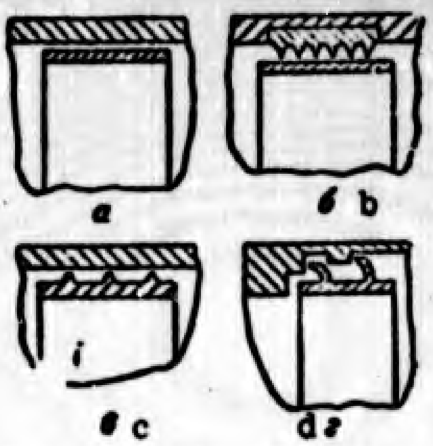

Fig. 88. Methods of reducing gas leakage in the radial clearance. a) Installation of shrouds on the turbine blades; b) installation of shrouds on the turbine baldes and a labyrinth seal in the casing; c) and d) installation of shrouds on the turbine blades with labyrinth seals.

The axial clearances in the turbine in another category of clearances. The axial clearances are divided into clearances between the disk rims and the shrouds of the nozzle diaphragms, and the axial clearances between the trailing edges of the nozzle blades and the leading edges of the turbine blades (the axial interblade clearances).

Normal axial clearances between the disk rims and the shrouds of the nozzle diaphragms insure the absence of contact between the rotor and the stator due to thermal deformations.

The magnitude of the axial clearances in the turbine change in relation to the operational regime of the engine. On start-up of the engine, the stator is heated more rapidly than the rotor shaft, so that the forward axial clearance between the stator and the rotor is decreased, and the rear clearance is increased. On engine shutdown in flight, when large masses of cold air flow through the engine, more rapid cooling of the stator relative to the rotor shaft leads to an increase in the forward clearance and a decrease in the rear clearance.

The necessary magnitude of axial clearance insuring absence of contact between rotor and stator under the most unfavorable operational conditions is determined in the light of the magnitude of thermal deformations of the rotor and stator and the axial displacement of the rotor as a result of the presence of axial clearance in the support-thrust bearing and in the sleeve coupling of the compressor and turbine shafts.

The magnitude of axial interblade clearance in present designs amounts to 0.1-0.4 of the chord. With decrease of this clearance, energy losses in the airflow in front of the turbine blades are reduced, the losses being related to equalization of the velocity field behind the nozzle blades and friction of the flow over the housing bounding the flow passage. However, at the same time, the excitation of turbine blade vibration is increased. As has been demonstrated by experimental research, some increase in axial interblade clearance is an effective method of reducing the intensity of vibration caused by nonuniform flow behind the nozzle blades; however, this is accompanied by some reduction in the turbine efficiency.

The magnitude of axial clearances should be made as small as possible, since the axial dimension of the turbine depends on it. In new designs it is chosen on the basis of statistics, and then refined by experimental test.

Labyrinths disposed in the radial or axial directions, are used as seals to reduce gas losses in the axial clearances between turbine stages.

In the radial arrangement of a labyrinth (see Fig. 74,a), the magnitude of the axial clearance and, consequently, the leakage of gas in the labyrinth is affected by the axial thermal deformation of the rotor and housings. In the axial arrangment of the labyrinth (see Fig. 74,c) the axial thermal deformations of the rotor and housings do not affect the gas leakage; however, the radial thermal deformations do have a noticeable effect, leading to change in the magnitude of the radial clearances in the labyrinth. Therefore for achieving minimum radial clearances both in the cold and in the hot states, two-sided axial labyrinths are used (See Fig. 74,b).

Chapter 6

THE COOLING OF GAS-TURBINE PARTS

The cooling of the structural elements of a turbine insures maintenance of temperature within permissible limits for the most critical parts, guaranteeing the necessary mechanical strength of those parts. Further, cooling of the parts permits increasing the time between engine overhauls, and if it is possible to avoid the use of high-alloy and high-priced steel and alloys (based on cobalt,nickel, etc.) the expense of gas-turbine fabrication can be signicantly reduced.

The basic cooling systems for the gas-turbine engines [GTD] widely used at present include the system of cooling hot parts by removal of heat into the turbine disk, and a system of internal cooling of the parts by air led through special channels located inside the cooled parts.

1. BLADE-COOLING SYSTEM WITH REMOVAL OF HEAT INTO TURBINE DISK

The system of blade cooling by removal of heat into the turbine disk is the simplest from the structural standpoint and does not require large expenditure of power; it is therefore used on most modern GTD [gas-turbine engines]. The heat from the more intesely heated fins of the turbine blades is conducted to the less-vigorously heated bucket shanks and through the keys into the rim and disk of the turbine The blade keys, the rim, and the disk of the turbine lose heat in turn into the surroundings. The extent to which the temperature of the blade bases is lowered depends on the intensity of heat removal by the turbine keys and disks into the surroundings.

In order to increase the heat flow from the keys and disk, they are cooled by air jets blown out under a pressure of a magnitude not exceeding the pressure of the air behind the compressor. In order to avoid large power losses in cooling, engines with axial compressors may take air from some intermediate compressor stage; and in engines with centrifugal compressors, the cooling air is ducted, as a rule, by use of specially designed low-pressure fans drawing air in from the atmosphere. In order to increase the efficiency of cooling, the air is directed toward the side surface of the disk by use of special deflectors.

After cooling the disk, the rim, and the keys, the cooling air is ejected into the flow passage. When the pressure of the cooling air is less than the gas pressure in the axial clearances of the turbine flow passage the cooling air cannot pass into the engine flow passage but must be ejected into the atmosphere by special channels provided for the purpose.

Two basic methods of supplying cooling air to the disk and buckets may be differentiated:

supply of air to a sleeve, with subsequent radial air flushing of the side surfaces of the disk and removal of heat from them;

supply of air directly to the rim, with subsequent ejection through holes in the rim or installation clearances in the turbine-bucket keys.

In the former case, the central part of the disk is more intensely cooled than the rim, which induces great nonuniformity in the temperature distribution along the radius of the disk, and leads also to the appearance of large thermal stresses in the disk that are of the same order as the stresses due to centrifugal forces. Ducting air through installation clearances reduces the temperature difference between the rim and the disk, and, consequently, decreases the thermal stresses. For the same temperature of the blade root sections, the flow rate of cooling air, when ducted through the installation clearances, is smaller by a factor of 8 to 9 than the flow rate in the case of radial flow along the disk, and does not exceed 0.5% of the [turbine] gas flow-rate, which is explained by the large surfaces of the slit channels and high intensity of heat transfer in the installation clearances.

Figure 89 shows a layout of two-sided cooling of a turbine disk by air led from intermediate stages of an axial compressor. The forward surface of the turbine disk is cooled by air taken from the eighth stage of the compressor through labyrinth seal 2 and through two flow passages 5 in the support casing. Air for the rear surface of the turbine disk comes from the fourth stage through six channels 11 in the compressor housing, through internal cavities 12 of the outer combustion-chamber liner, and through the hollow struts of the exhaust nozzle and orifices 10. After cooling the disk and its rim, the air is ejected into the flow passage through axial clearances, since the pressure of the air is greater than the gas pressure in the axial clearances.

The layout for single-side cooling of the turbine disk by air supplied by a special fan is shown in Fig. 90. Air in an amount 0.8-1% of the over-all airflow through the engine is sucked in from the nacelle through aperture 2 located in the rear supporting structure, and thereafter flows to the fan impeller 1 mounted on the shaft of the compressor rotor. The power required by the fan is equal roughly to 0.4% of turbine power. The impeller air inlet is formed by an inlet 3 and the forward wall of housing 4. Air is compressed to a pressure of 1.3-1.5 kg/cm^2 in impeller 1 and in vaned diffuser 5 and then through aperture 6 in the wall of the housing, it flows into the supporting housings. The temperature of the cooling air at outlet from the fan is equal to 70-80°C. In order to prevent leakage of air at the inlet to the vaned diffuser, labyrinth 7 is installed between the housing and the impeller. Thus an excess pressure of 0.3-0.5 kg/cm^2 is established inside the housing enhancing the packing of the oil bearing labyrinths (pressurizing the labyrinths).

The primary flow of cooling air is toward the turbine disk. In the couse of its flow toward the turbine disk, the air cools the turbine shaft, the bearing housing, and the support housing.

Decrease in the heating of the inner bearing ring by heat conducted along the turbine shaft is achieved by installation of 2 bearing on stepped sleeve 9. Air floes in the clearance formed by the sleeve and shaft; the air is then ejected through end slots in the sleeve flange by reason of the centrifugal effect.

Cooling air, and gas leaking through labyrinth 10, flow from the turbine disk through nine tubes 11 into the air-dicharge chamber 12, from which it is then ejected into the atmosphere. To decrease the heating of the turbine bearing housing with air flowing from the disk, screen 13 is used. Air ejected through holes 14 in the casing insures intense air circulation in the clearance between the screen and the housing.

A two-stage gas turbine cooled by air conducted through slits in the fir-tree key is shown in Fig. 63. Part of the secondary air bled from the combustion chamber in three streams is used for cooling.

The first airflow enters the space between the outer housing of the nozzle diaphragm of the first stage and lining shoes 11, and cools the outer bases of the nozzle blades. In this arrangement, part of the air is ejected into the flow passage through the clearance between the

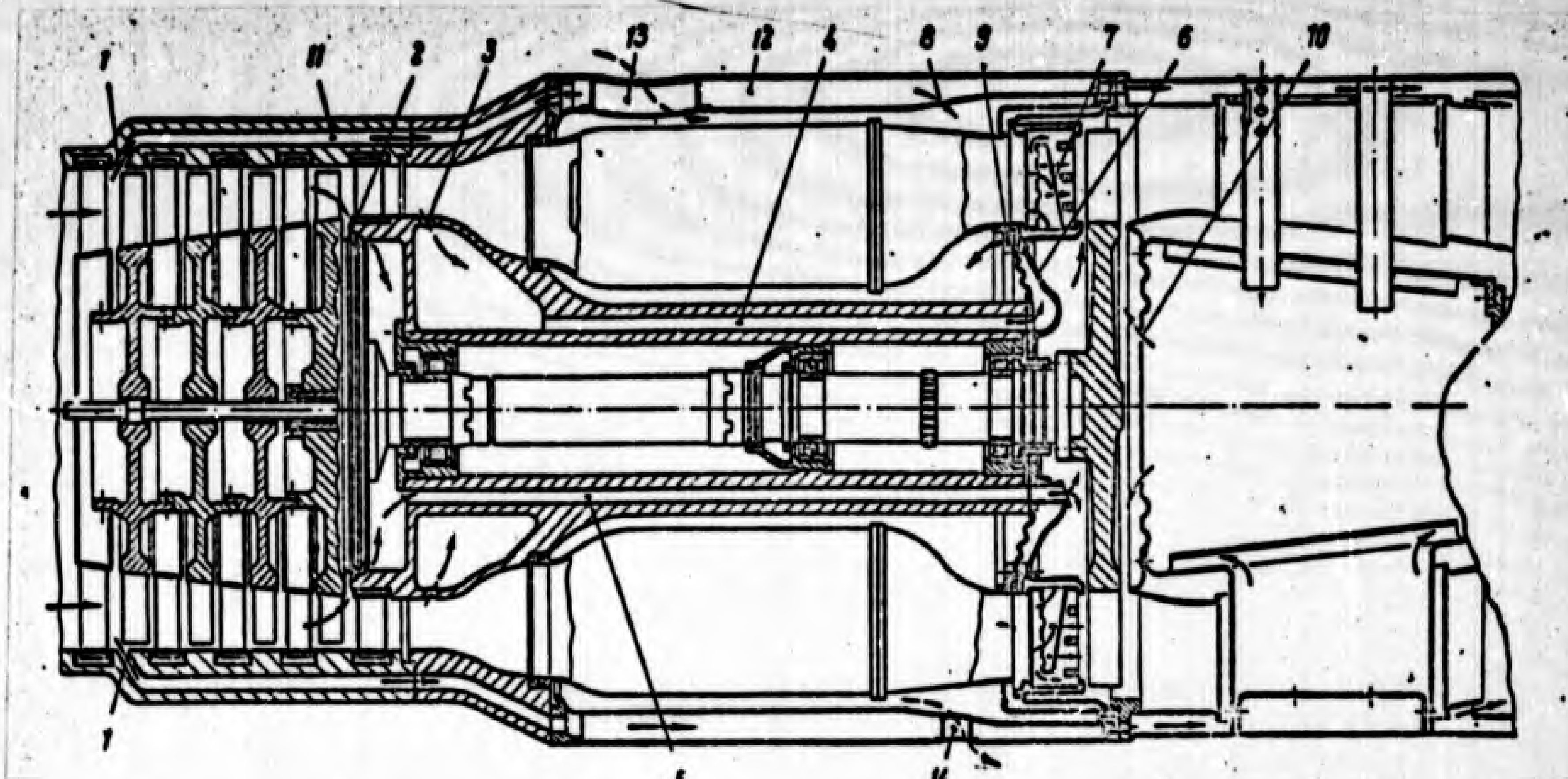

Fig. 89. Layout of turbine cooling by air bled from intermediate stages of an axial compressor. 1) Annular air scoop behind the fourth compressor stage; 2) labyrinth seal; 3) air outlet behind the compressor; 4) airflow channel for cooling blades of nozzle diaphragm (three channels); 5) airflow channels for cooling turbine disk (two channels); 6) separating wall; 7) nozzle blade deflector; 8 and 9) air inlet for cooling gas collector; 10) aperture for taking in air; 11) channel for air; 12) cavity in the combustion-chamber housing; 13) aperture for inlet of air from the atmosphere; 14) air outlet to the atmosphere.

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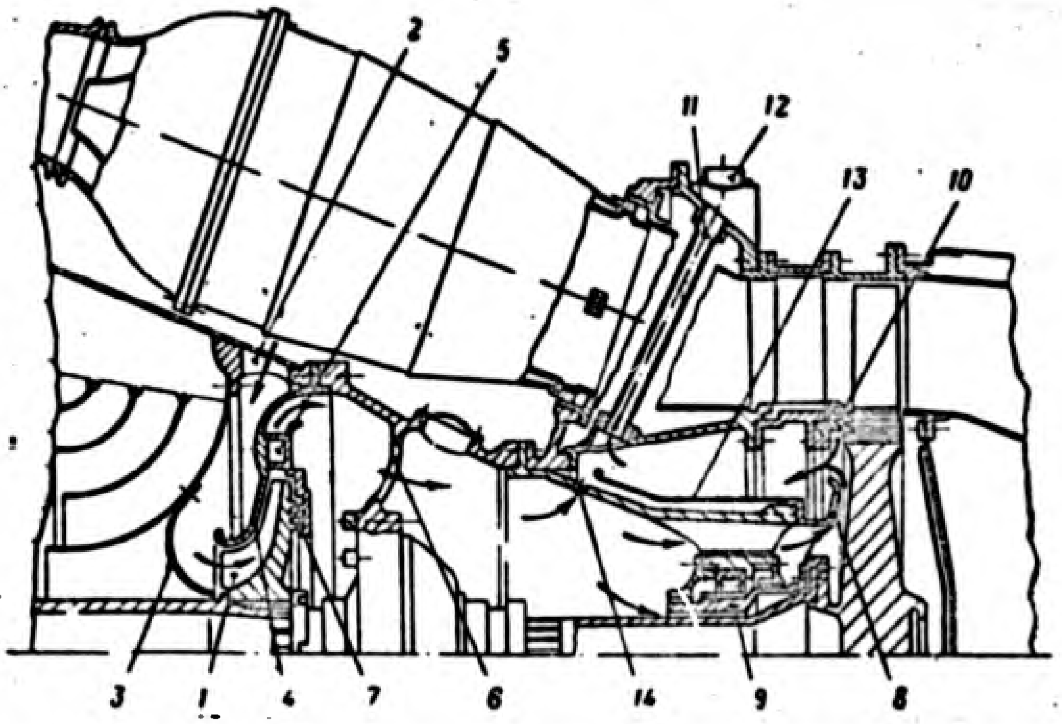

Fig. 90. Diagram of turbine cooling by air supplied by centrifugal compressor (motion of air is shown by arrows). 1) Fan impeller; 2) aperture; 3) inlet; 4) fan housing; 5) vaned diffuser; 6) aperture in the housing wall; 7,and 10) labyrinths; 8) deflector; 9) stepped sleeve; 11) tube; 12) air-discharge chamber; 13) screen; 14) hole.

nozzle blades and the shoes, while a large part flows into the clearance between the housing and the lining shoes 12 of the second-stage nozzle diaphragm, cooling the bases of the second-stage nozzle blades, after which it is ejected into the flow passage. Because of the air layer between the shoes and the nozzle-diaphragm housings the temperature of the latter is maintained at a relatively low value (350-350°C).

A second current of air in an analogous manner cools the inner bases of the nozzle blades and the inner housing of the nozzle diaphragm of the first stage, after which it is ejected into the engine flow passage so as to produce a zone of low temperature gas flowing past the root section of the turbine blades of the first stage of the turbine. A part of the air cools strut 13 and strut fairings 14 of the supporting structure.

The third airflow enters air distributor 5 with holes carefully selected as to size and location, and these distributes the air over the parts of the turbine, i.e., part of the air is directed toward the keys of the turbine buckets of the first stage, and part to holes in the disk of the first stage, through which the air is ducted inside support ring 6 to cool the disk and the keys of the second stage. The first part of the air proceeds through the clearances in the keys, intensively removing the heat from the base of the blades, and through slanted holes 7 to the overhang of the disk, (cooling the latter), into the cavity between the disks, and thereafter into the flow passage through the axial clearances.

The air from the inside cavity of the support ring flows through the radial holes in the ring shoulder to the keys of the second-stage blades, cooling the bases of the turbine buckets and proceeding through holes in the disk of the second stage, cooling the back surface of that disk, after which it is ejected into the flow passage of the engine.

The heat outflow from a blade affects only the temperature of its base (at a distance of about 25% of its length). The temperature of the middle and tip parts of the blade remains unchanged and is practically the same as the gas temperature.

In spite of the insignificant reduction of the temperature of the middle and tip parts of the blades, the efficienty of the blades and their service lives are considerably increased, since the strength margin is increased in the relatively less heated root section in which the greatest total tensile stresses are present, these being caused by centrifugal forces in addition, there are the bending stresses caused by aerodynamic forces.

These temperature of the gases in front of the turbine buckets is kept nonuniform in the radial direction, i.e., lower near the roots and tips of the blades, and higher near the center. The nonuniform change in temperature increases the strength margin of the especially highly loaded root sections of the turbine buckets and lowers the temperature of the turbine housing. The creation of local zones of low-temperature gas is achieved by supplying relatively cold secondary air from the combustion chamber to these zones through holes in the flame tubes or through clearances between the flame tubes and the combustion-chamber shells.

Computations show that the lowest gas-temperature value at the blade root corresponds to a maximum-temperature position at 2/3 of the blade length from the root section.

In present turbine designsm the maximum gas-temperature difference in the radial direction is equal to 100-400°C. In this case, the gas temperature at the root section of the turbine buckets is 450-700 °C. With no heat outflow into the stem, the root section would have the same temperature. Because of the heat outflow from the stem, the temperature of the root section of the blade may be lowered by 50-100°C.

The temperature of turbine disks of modern GTD [gas-turbine engines] in relation to the gas temperatures and the conditions of cooling comes to 450-700°C at the rim and 150-600°C in the center.

2. EQUIPMENT OF INTERNAL SYSTEMS FOR COOLING TURBINE BUCKETS WITH AIR

The basic elements in the design of an internal system for the cooling of turbine buckets by use of low-pressure air include an inlet for the cooling air, air ducts to the turbine, seals at the points where the air is transferred from stationary parts to rotating parts, deflectors at the turbine disk, insuring motion of air along the turbine disk and directing the air toward the buckets, bucket deflectors intended for cooling bucket fins, and equipment for the keying of the cooled buckets.

Cooling-air inlets may be annular or made in the form of apertures. With ring diversion of air, the flow distribution in the compressor is disturbed as little as possible, so as to reduce any danger of generating vibration in the compressor vanes.

Ducting of air from the compressor to the disk is accomplished by use of outer tubes or channels, located in the housings, or through a hollow turbine shaft.

In order to avoid large pressure losses in the ducted air, the channels should not have sharp turns or expanded sections. The channels are positioned at equal intervals about the circumference of the turbine disk,which insures uniform distribution of air about the disk and buckets. Heating of the air on its path from the compressor to the turbine disk should be minimum.

Since the required flow rate and pressure for cooling air is different for nozzle and turbine buckets, the bleeding of air may be accomplished at various compressor stages. In order to prevent mixing of these two flows, separating diaphragms 6 (see Fig. 89) are installed.

In the design development of a cooling system, great care must be taken to insure good seals at the points at which the air transfer from stationary parts to rotating parts and at the spots where the buckets are joined to the disk, since only when good seals are provided will the design volume of cooling air actually flow across the buckets being cooled. Failure to consider leakage of cooling air along the path to the buckets may substantially lower the effect of cooling. By way of seals at locations of air transfer from fixed to rotating parts throttling labyrinths 10 (see Fig. 77,a) are installed.

Turbine disk deflectors may be fixed or rotate with the disk. Figure 90 shows fixed deflector 8, and Fig. 77,a shows deflector 18 which turns with the turbine. Deflectors increase the effectiveness of disk cooling and decrease the quantity of air required.

Cooled turbine buckets are provided with channels for ducting of cooling air, the total flow cross-section of which is determined by the design air requirement. The number, shape, and distribution of cooling cavities are chosen in such a fashion as to guarantee the greatest possible cooling efficiency with minimum hydraulic resistances and the necessary uniformity of the temperature field in the bucket cross-sections. Uniform cooling of the buckets is provided by bringing the cooling channels close to the places having the highest temperature, that is, to the inlet and exit edges.

Increase in the efficiency of cooling, that is, lowering of the required air flow for given cooling intensity, is insured by increase of the cooling surface and the coefficient of heat flow from the bucket walls to the cooling air.

Considerable increase in the cooling surface is achieved by use of fonning at the internal cooled side of buckets, or the use of finned inserts 1 of relatively small diameter, introduced into cooling channels 2 (Fig. 91,a).

Increase of the coefficient transfer from the wall to the air is achieved by reducing the diameters of the cooling channels. For this purpose special inserts, i.e., deflectors guiding the air along or across the buckets, are installed in hollow buckets.

In the first case the air flows along the clearance between the wall of the bucket and the deflector, that is, in the longitudinal channels, and is discharged into the flow passage at the end of the bucket through the end cross-section (Fig. 91,b). Examples of transverse sections of buckets with longitudinal channels for ducting of cooling air are shown in Fig. 92.

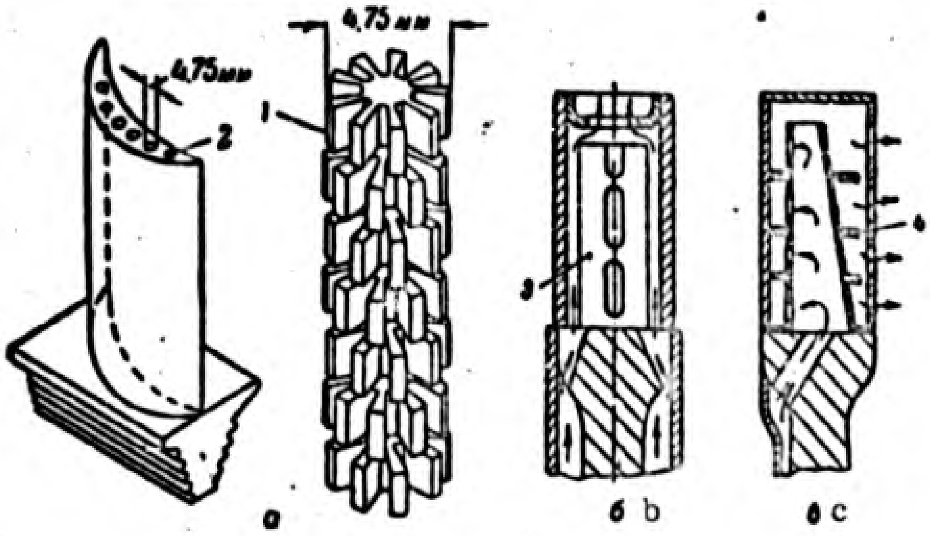

Fig. 91. Schematic diagram of inserts and deflectors for buckets with internal air cooling. a) Finned inserts; b) deflector guiding the air along bucket; c) deflector guiding the air across bucket. 1) Insert; 2) cooling channel; 3 and 4) deflectors.

In case of transverse flow of air, the latter initially enters inside deflector 1 (Fig. 93), after which it flows through a series of slits in the deflector into the nose, the more intensely heated part of the bucket (see Fig. 91,c) and thereafeter is ejected into the flow passage through slits along the trailing edge. The methods of ejecting air are shown in Fig. 93,a and b. In such an arrangement of the motion of the air, excellent cooling is achieved at the inlet and exit edges of the bucket, and the distribution of temperature in the cross sections of the bucket turns out to be more uniform than with longitudinal motion of the air.

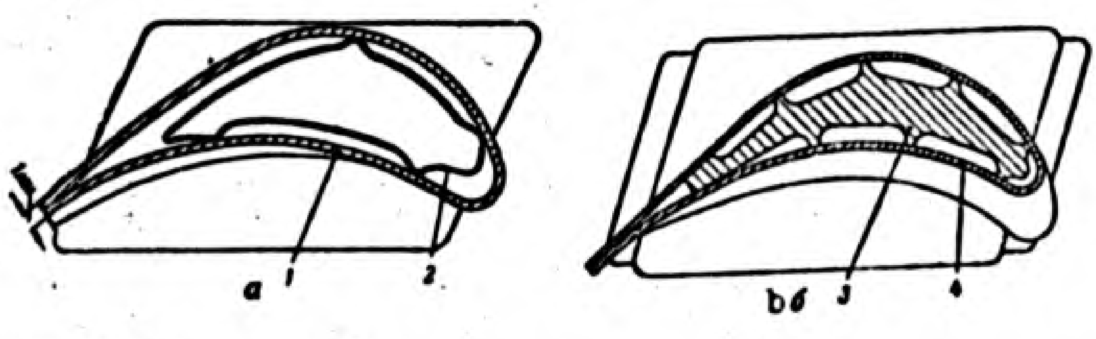

Fig. 92. Cross-sections of buckets with inner air cooling, in which the cooling air flows longitudinally. a) Bucket with load-bearing wall and nonload-bearing deflector; b) bucket with load-bearing core and inoperative screen. 1) Wall; 2) deflector; 3) core; 4) screen.

Cooled buckets are differentiated as to the loading arrangement:

1) arrangement with load-bearing wall and nonload-bearing wall and nonload-bearing deflector;

2) arrangement with load-bearing core and nonload-bearing wall (screen).

In the buckets of the first arrangement (Fig. 94,a) all the load is borne by the wall, which at the same time forms the bucket profile. The deflector is loaded only by centrifugal forces due to its own intrinsic mass. Buckets are formed by bending from sheet steel. The bucket key is made in one piece with the load-bearing wall. The deflector is attached by use of dowel 1 and finished by welding. The

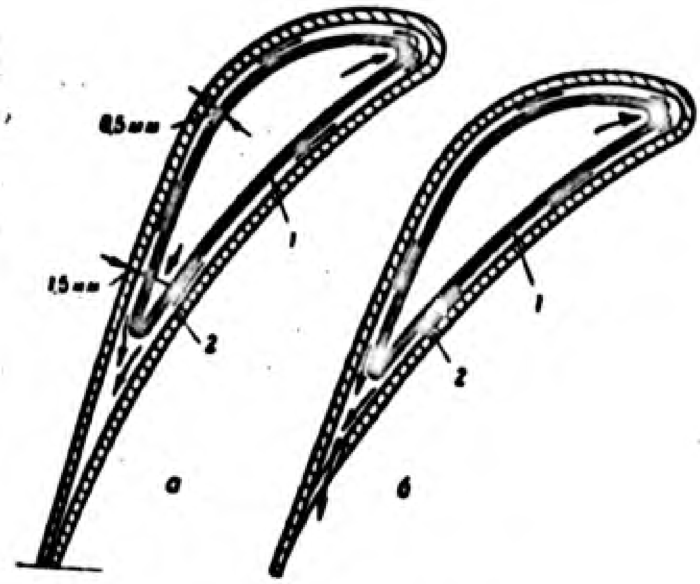

Fig. 93. Methods of discharging cooling air with transverse flow. a) Exit slit in the edge; b) exit slit in the concave surface of bucket. 1) Deflectors; 2) walls.

clearance between the deflector and the bucket wall is ordinarily not greater than 0.5-0.7 mm.

Buckets of the second arrangement consist of a nonload-bearing wall, functioning as a screen. In the design shown in Fig. 94,b, the load-bearing core, made in one piece with the key, consists of an involuted blade, on the body of which there are longitudinal ribs. The screen made of steel sheet of thickness 0.5-0.8 mm is first bent according to the shape of the core so that it can be mounted on the core, and finally it is driven in place and welded to the core at the base.

The screen is subject to gas forces and transmits them to the load-bearing core, which, further, is loaded by centrifugal forces of the intrinsic mass. There are channels for the supply of air to the clearance between the screen and the core in the key and rim. Cooling air flows in the longitudinal direction, cools the screen and the supporting core, and flows out through the end of the blade. The temperature of the supporting core, shielded by the cooling screen, is considerably lower than the screen temperature. The reliability of this turbine bucket is found to be quite high.

A drawbach of cooled buckets with a supporting core and non-load-bearing screen welded to the core, is the difficulty in the choice of materials with the requisite coefficients of linear expansion, insuring the absence of large thermal stresses in the screen for various operational conditions of the engine.

Cooled nozzle blades have much in common in design with cooled turbine blades. Differences consist only in the methods of attaching the blade deflectors of the air-cooled blades. For example, Fig. 95,a shows deflector 1 welded to the base of the blade and Fig. 95,b shows deflector 1 welded to the outer end by use of core 4 and plate 5. Slits for exit of cooling air are formed by use of wedge-shaped plates 2, welded by spot welding to the wall of the bucket (Fig. 95,a) or by milling the ends of the plates joined together before welding (Fig. 95,b). In cast blades, the slits are made in the exit edge by use of electro-erosion methods.

The keying devices for cooled turbine buckets have channels with

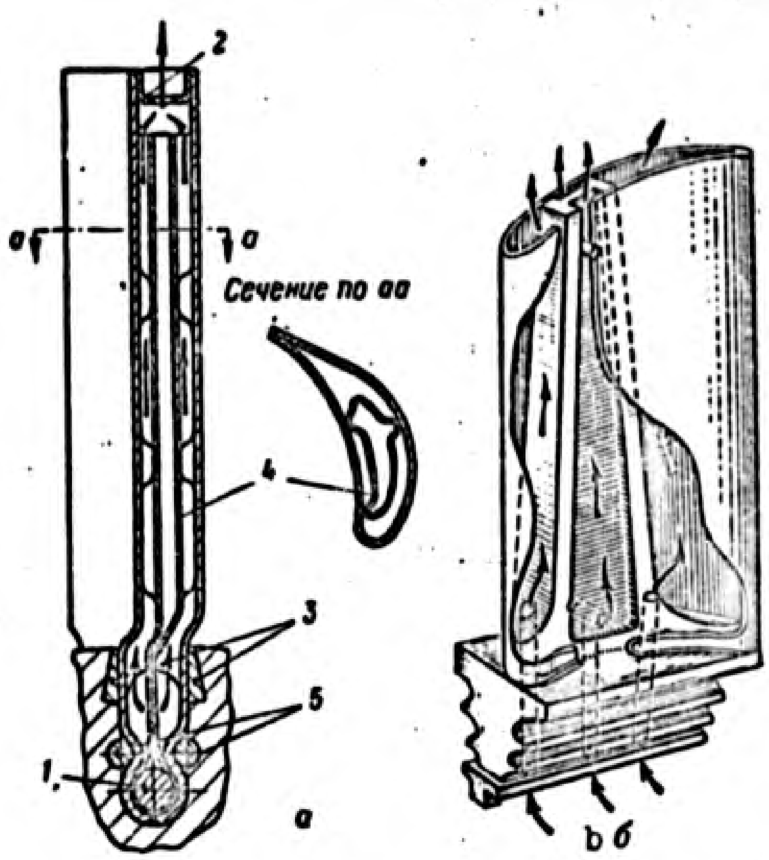

Fig. 94. Load-bearing layout of turbine blades cooled by air. a) Layout with load-bearing wall; b) layout with load-bearing core. 1) Attachment dowel for the deflector; 2) bottom plate; 3) wedges; 4) deflector; 5) load-bearing dowels for bucket attachment.

flow – passage areas sufficient for introduction into the bucket of the required quantity of air, without significantly lowering the strength of the key. In the process, the hermetic sealing of the bucket coupling to the turbine disk must be insured, because failure to take account of leakage of air through the key coupling may noticeably impair the cooling of the buckets.

Figure 94,a shows the attachment of a cooled bucket by use of a cylindrical key. In order to increase the airflow-passage area, the air it is ducted into two sides of the key. The design of the cylindrical key of the air-cooled bucket turns out to be simple. A deficiency of such a design is its reduced air-tightness due to the difficulty of achieving an airtight fit of the bucket into the socket in the disk. Increase in rigidity of attachment and sealing is accomplished by

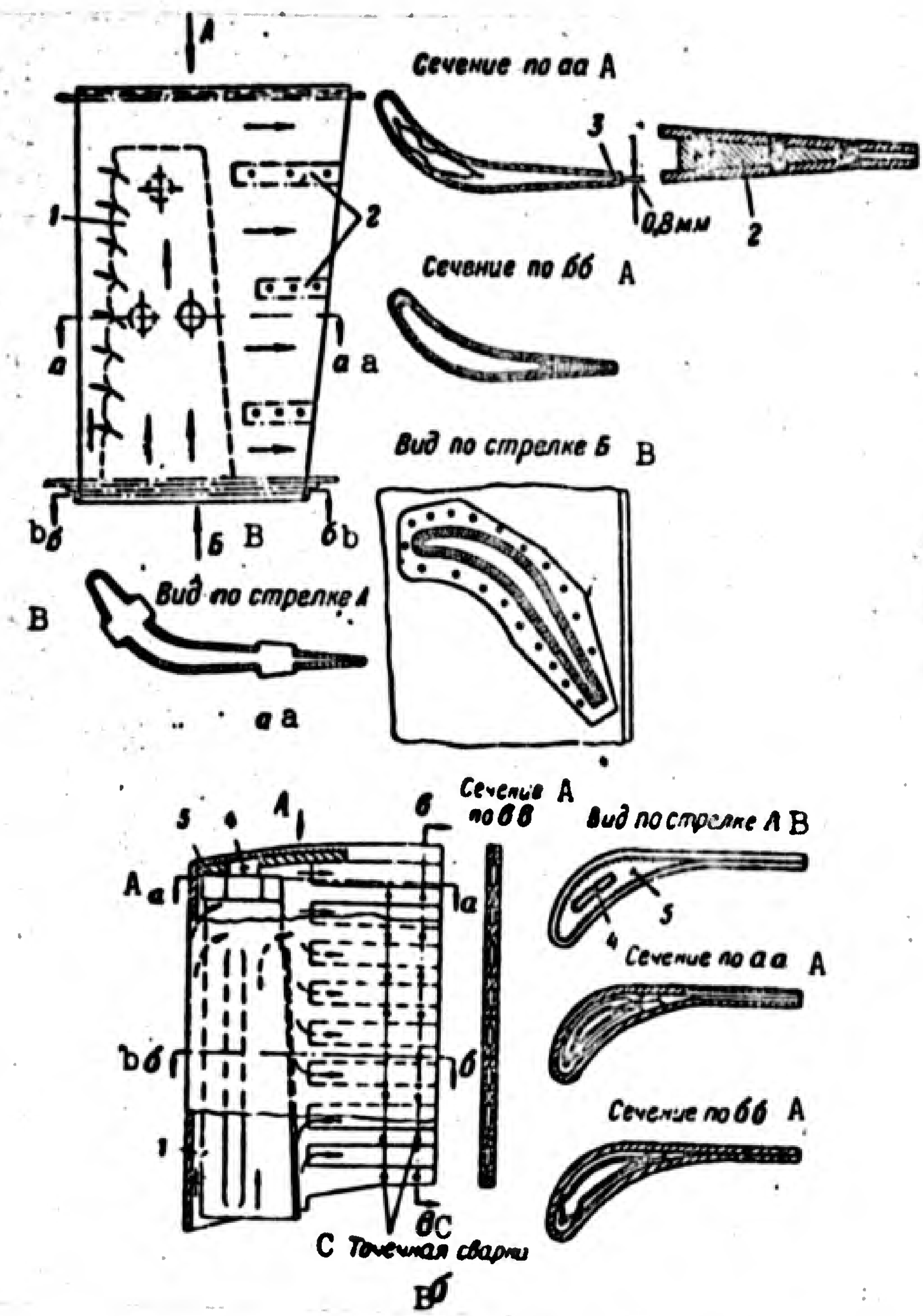

Fig. 95. Nozzle diaphragm blades cooled by air: a) With a deflector guiding the air across the blade; b) with deflector guiding the air along and across the blade. 1) Deflector; 2) plate; 3) slit for ejection of air; 4) core; 5) plate. A) Section along; B) view along the arrow; C) spot welding.

press-fitted wedges 3.

Figure 94,b shows the design of a cooled fir-tree key.

Calculations indicate that in the case of internal cooling of turbine buckets with a cooling-air requirement of about 6-7% and a bucket-wall temperature equal to 700°C, the permissible temperature of the gas through the turbine buckets amounts to 1050-1100°C, and through the nozzle diaphragm of the turbine, 1200-1250°C, i.e., the cooling of the bucket by air permits of increasing the temperature in the combustion chamber ahead of the turbine by 300-350°C in comparison with the temperature used at present (850-900°C). A further increase in the flow of air in an open cooling system is not feasible, since air emitted into the engine flow passage results in considerable losses from the viewpoint of doing work in the turbine buckets. In this case the power losses in cooling and the complexity of the design amount to more than the power gained by reason of the increased gas temperatures in front of the turbine [27].

Part Three

MAIN COMBUSTION CHAMBERS AND AFTERBURNERS. EXHAUST SYSTEMS

Chapter 7

MAIN GTD [GAS-TURBINE-ENGINE] COMBUSTION CHAMBERS

1. COMBUSTION-CHAMBER TYPES AND RELATIVE RATINGS

GTD combustion chambers are classified on the basis of: air and (gaseous) combustion-product flow direction, method of supplying fuel to the combustion zone, and chamber construction and location on the engine.

GTD combustion chambers are classified as straight-through flow or reversed-flow engines on the basis of air and combustion-product flow direction.

On the basis of the method used to supply fuel to the combustion zone, chambers are classified as vapor-phase-fed and liquid-phase-fed chambers (in the latter case, the liquid is supplied as finely dispersed drops).

On the basis of construction and location, chambers are classified as individual-tube (can), annular, or cannular.

Straight-through flow chambers (Fig. 96) are the most widely used since in comparison with reversed-flow chambers the hydraulic losses are substantially less while chamber dimensions (diameters) cannot exceed the maximum diameters of the axial compressor and turbine.

A drawback to this type of chamber is the increased distance between turbine and compressor, which increases the over-all engine length and, chiefly, the distance between the fore and aft engine-rotor supports with a corresponding complication in design and increase in engine-rotor weight.

Reversed-flow chambers permit a reduction in engine and rotor length, since they are usually placed around the turbine and tailpipe (Fig. 97). In isolated cases (Fig. 98), the combustion chamber is located between turbine and compressor. In this case, the flame chamber in which the fuel is burned and the combustion products mixed with air are so shaped that the distance between compressor and turbine is nearly halved in comparison with the corresponding distance for an engine having a straight-through flow chamber.

Reversed-flow combustion chambers are best used where engine weight and size are decisive factors. Here we are primarily concerned with combustion-turbine starters.

Combustion chambers with vapor-phase fuel supply are fairly seldom used in GTD. This is primarily explained by the complexity of building a reliably operating vaporization system, which must take the form of a set of vaporizing tubes 4 (Fig. 99), located in the combustion zone. The tubes carry a highly enriched fuel-air mixture, heated until the fuel has completely vaporized. Mixture composition, the form and dimensions of the vaporizing tubes are dictated by the need to prevent coking or charring while maintaining stable operation of the chamber under all service conditions.

The basic advantages of a vapor-combustion chamber are:

– a reduction in the required maximum fuel pressure ahead of the nozzles and, consequently, throughout the fuel-delivery system, which increases the operating reliability of the fuel-delivery and metering apparatus;

– increased chamber operating reliability at high altitudes, as there is no longer any need for fine atomization of the fuel by the injectors at the low fuel flow rates required at high altitudes;

– within the chamber, combustion proceeds without flame tongues

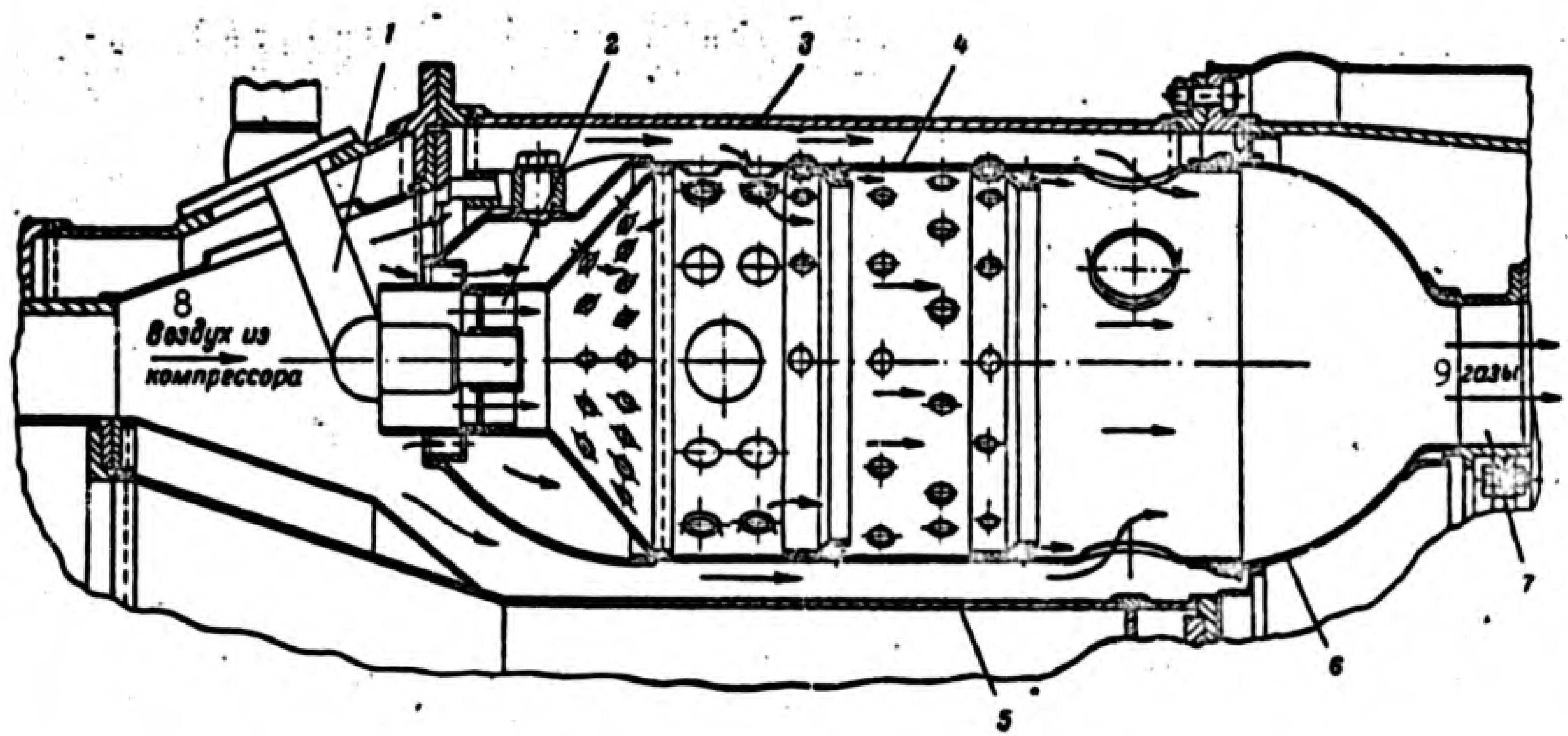

Fig. 96. Straight-through flow cannular combustion chamber: 1) nozzle; 2) swirler; 3) outside chamber; 4) flame tube; 5) inside chamber; 6) gas collector; 7) turbine nozzle diaphragm; 8) air from compressor; 9) gases.

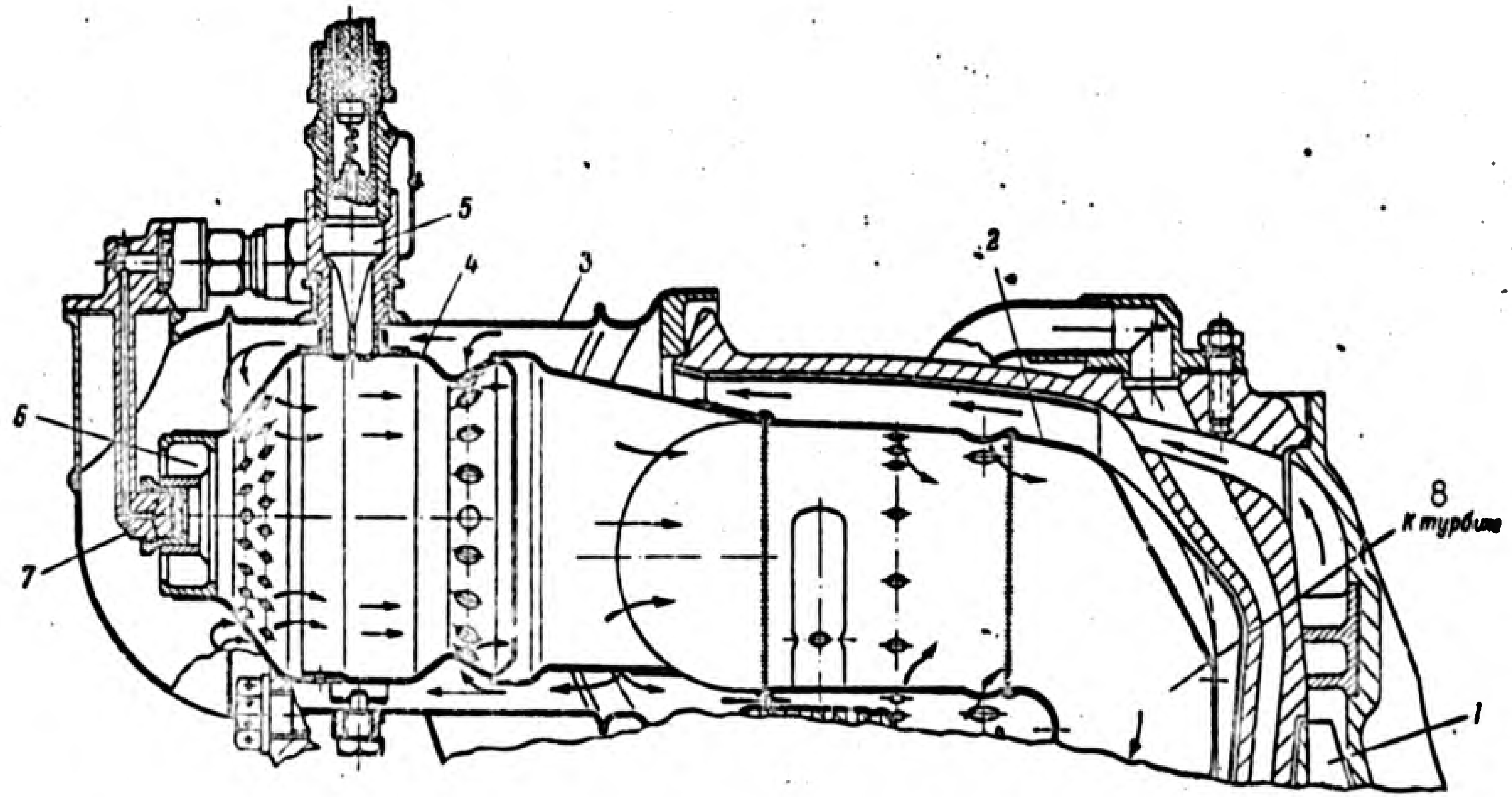

Fig. 97. Reversed-flow can-type combustion chambers: 1) compressor runner; 2) gas collector; 3) chamber housing; 4) flame tube; 5) ignition plug; 6) swirler; 7) nozzle; 8) to turbine.

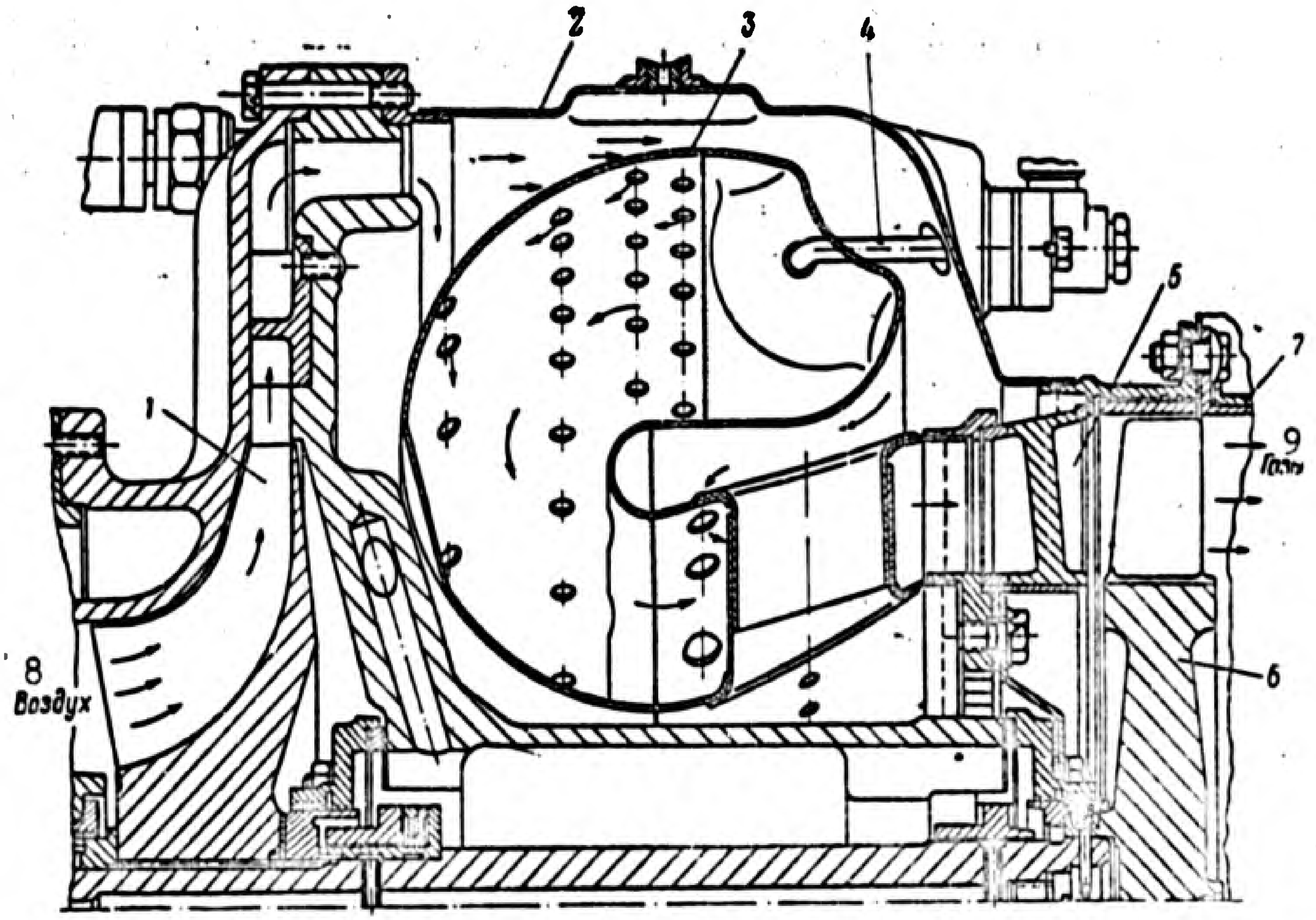

Fig. 98. Reversed-flow annular combustion chamber: 1) compressor runner; 2) chamber housing; 3) flame tube; 4) nozzle; 5) turbine nozzle diaphragm; 6) turbine runner; 7) exhaust section; 8) air; 9) gases.

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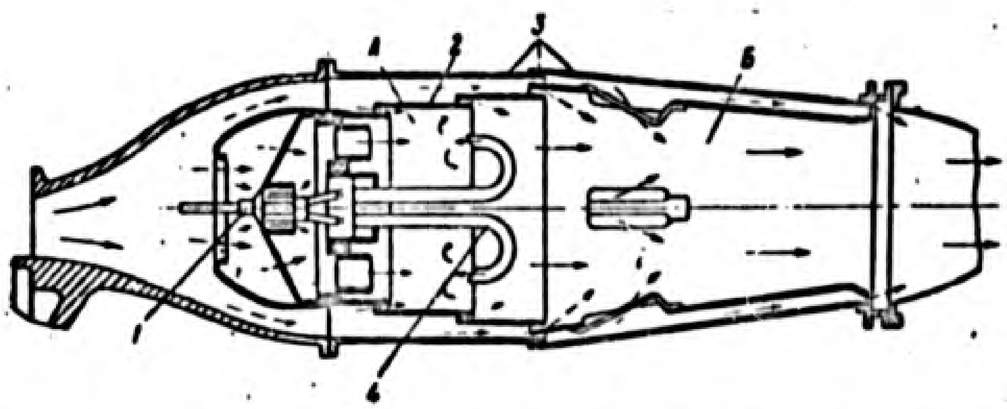

Fig. 99. Straight-through flow combustion chamber with fuel supplied as the vapor: A) combustion zone; B) mixing zone; 1) nozzle; 2) flame tube; 3) chamber housing; 4) vaporization tube (for vaporized-fuel injector, see Fig. 208).

or deposition of soot, and this reduces radiative heat transfer to the chamber walls and improves chamber cooling conditions.

The drawbacks of vapor-combustion chambers include:

– the complexity of starting up the chamber and bringing it into vaporization operation, which requires a large number of starting injectors and plugs (as many as 6-7);

– a reduction in maximum combustion efficiency (2-3%) as compared with chambers to which fuel is supplied in atomized form;

– the difficulty of ensuring a uniform distribution of fuel vapor throughout the chamber cross section, which makes it necessary to increase the chamber cross section.

As a rule, an advantage to vapor-combustion chambers is reduced length (35-40%), owing to the fact that the vaporization processes occur within the vaporizers, rather than at the front of the chamber. There is no accompanying weight reduction, however, owing to the additional weight of the tubes and the increase in chamber cross-sectional dimensions.

Combustion chambers with fuel delivered in liquid phase (as fine drops) are predominantly used for GTD. In such chambers, the fuel is supplied through nozzles designed to produce the best dispersion and

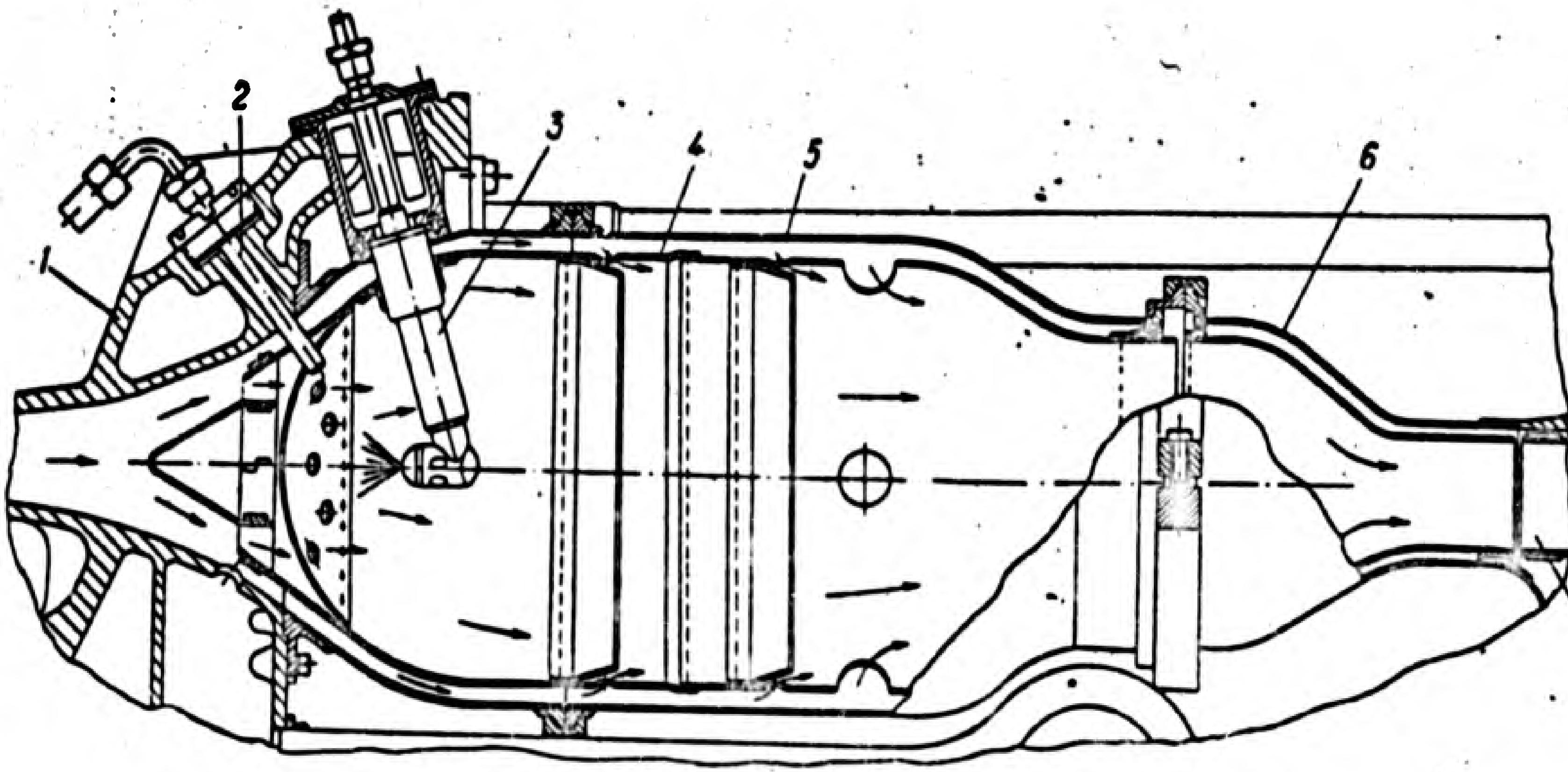

Fig. 100. Straight-through flow combustion chamber with fuel flowing against the air stream: 1) compressor housing; 2) plug; 3) nozzle; 4) flame tube; 5) chamber housing; 6) gas collector; 7) turbine diaphragm.

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mixture of the fuel. Fuel-drop diameter usually runs from 10 to 300-400 microns.

The great store of experience that has accumulated in designing and operating combustion chambers with nozzles permits reliably operating GTD combustion chambers to be made fairly rapidly. Design and implementation work is easier for chambers using atomization owing to the fact that fairly accurate design methods are available for nozzles, and a large number of investigations have been carried out into the processes of atomization, mixing, and combustion of a fuel supplied to the chamber through atomizing nozzles.

Fuel is sometimes injected against the direction of gas flow (Fig. 100) in order to improve fuel dispersion and mixing. In such chambers, the nozzles are subject to vigorous heating. For this reason, in most engines fuel is injected into the chamber in the direction of gas flow.

Can-type straight-through flow chambers (Fig. 101) are usually installed around bearing housings, with no connection between chamber elements and the engine stress-bearing system. One engine may have 6 to 20 chambers.

The advantages of can chambers include:

– simplicity of experimental development work, since the air- and fuel-flow rates required for each chamber are small;

– the possibility of rapid chamber removal and installation under service conditions for chamber inspection, elimination of flaws, or replacement.

At the same time, these chambers have the following drawbacks:

– increased weight of engine stress-bearing system elements and of the engine as a whole;

– inefficient use of the volume included between compressor and turbine, as spaces are left between the combustion chambers and in com-

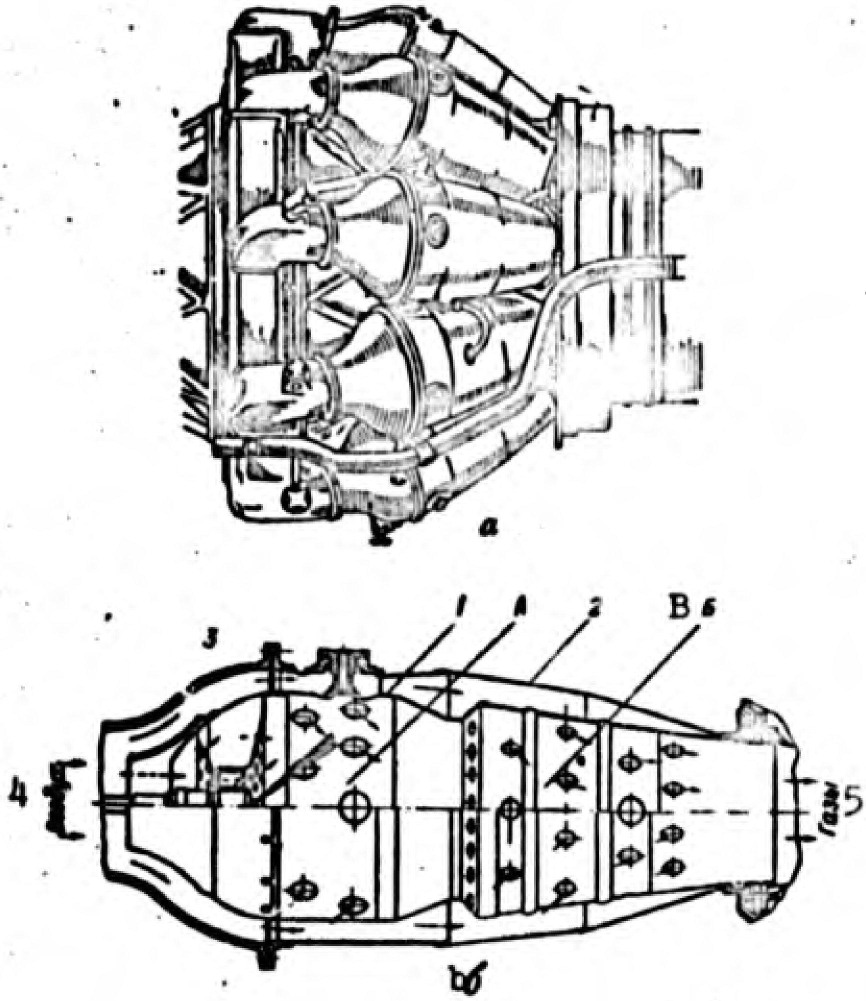

Fig. 101. Can combustion chambers: a) arrangement of chambers on engine; b) diagram of combustion chamber; A) combustion zone; B) mixing zone; 1) flame tube; 2) chamber housing; 3) chamber diffuser; 4) air; 5) gases.

pensation, engine diameter is increased at the combustion chamber;

– a less uniform temperature field on the periphery of the chamber outlet ducts;

– more complicated combustion-chamber connector-duct design and the need to install a gas collector ahead of the turbine nozzle diaphragm.

Can chambers are widely used for engines with centrifugal compressors. The desirability of using can combustion chambers for these engines lies in the fact that this permits full utilization of the advantages of can chambers with respect to their experimental development; chamber inspection and replacement is also easier. The basic drawback of increased engine size at the combustion chambers is not

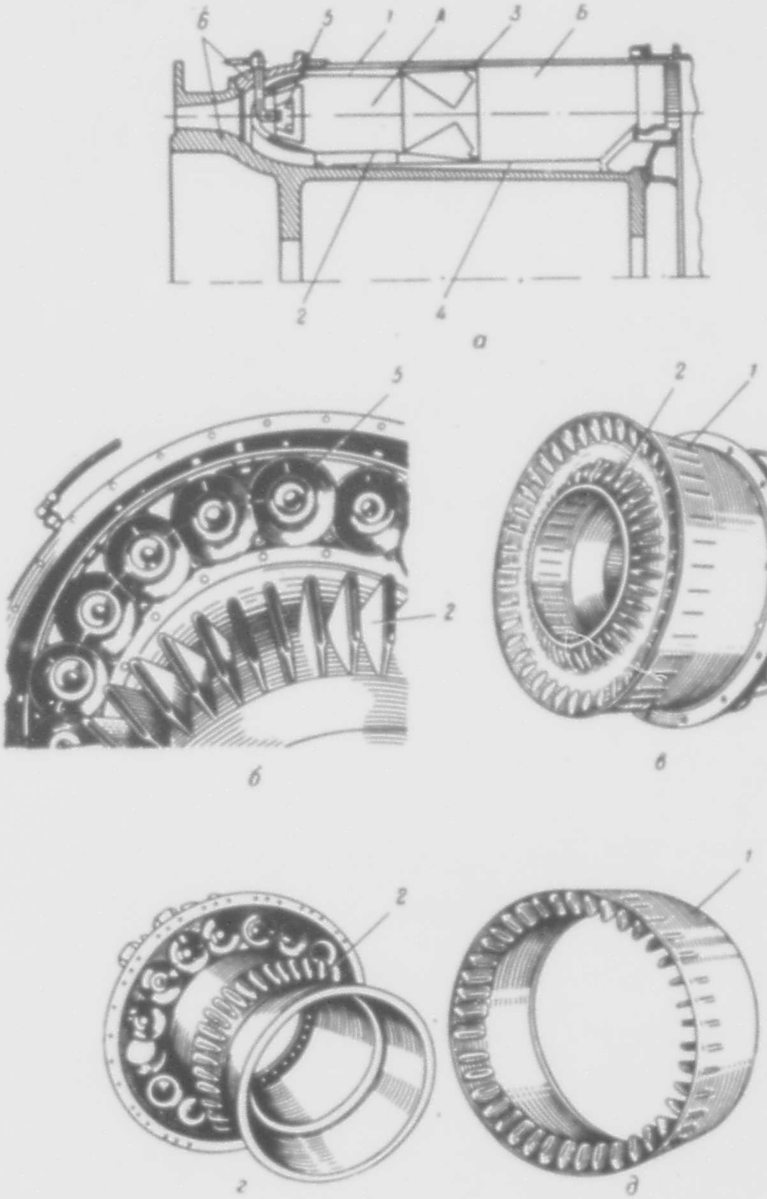

Fig. 102. Annular combustion chamber: a) diagram of chamber; b) arrangement of chamber flameholders; c) over-all view of flame tube; d and e) inside and outside flame-tube casings; A) combustion zone; B) mixing zone; 1) outside flame-tube casing; 2) inside flame-tube casing; 3) outside chamber casing; 4) inside chamber casing; 5) flameholder; 6) chamber diffuser.

very important, since the maximum engine diameter is determined by the compressor dimensions.

Annular combustion chambers (Fig. 102) differ from other types of chambers in having a minimum number of parts and in being compact. Annular chambers are also considerably cheaper to manufacture than can

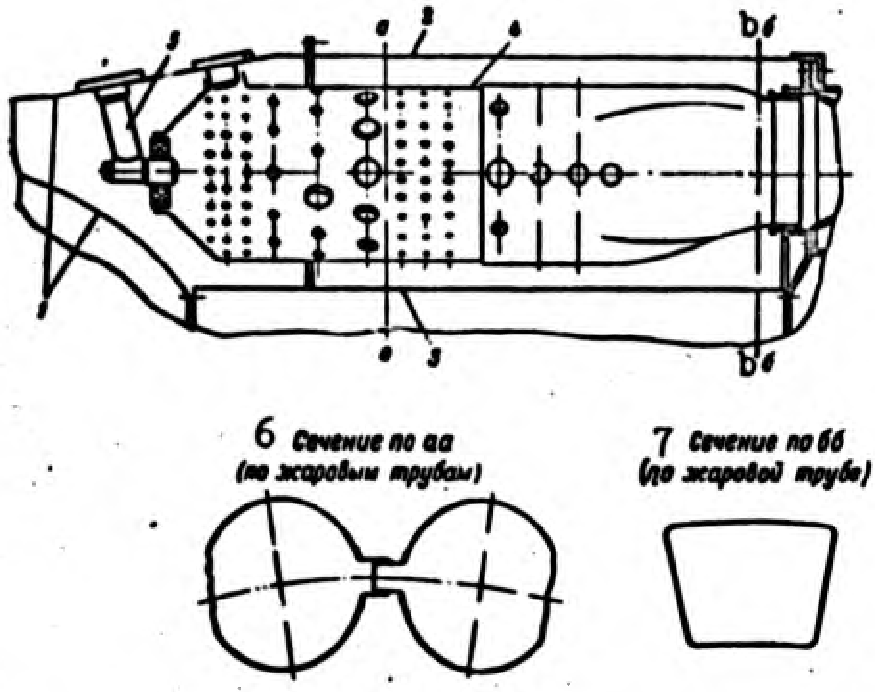

Fig. 103. Diagram of cannular combustion chamber: 1) chamber diffuser; 2) outside chamber casing; 3) inside chamber casing; 4) flame tube; 5) nozzle; 6) section through aa (along flame tubes); 7) section through bb (along flame tube).

combustion chambers. The chamber housings are usually connected into the engine stress-bearing system. This increases the stiffness of the entire engine and reduces its weight.

The difficulties involved in experimental development of annular chambers, which require full-scale testing on installations delivering large air flow rates reduce the possibility of testing a large number of chamber models for selection of the best design. In this connection, the time required to create and develop GTD will be increased, as a rule.

Cannular chambers are an intermediate chamber type which include elements of annular and can chambers. In the annular space included between the outside casing 2 (Fig. 103) and inside casing 3 of the chamber, there are the flame tubes 4 at the fronts of which, as in the flame tubes of can chambers, the processes of fuel vaporization, mixing, and combustion occur; at the tube rears the gases are mixed with diluent air and the gas stream is directed to the turbine nozzle diaphragm.

Cannular chambers are the most commonly used type for engines, as they combine the advantages of both can and annular chambers.

2. CONSTRUCTION OF COMBUSTION-CHAMBER ELEMENTS

The main elements of combustion chambers are chamber diffusers, flame tubes, and housings or casings within which the flame tubes are located.

A combustion-chamber diffuser is a diverging duct, in which air speed is reduced from 120-160 m/sec (at the compressor outlet) to 60-70 m/sec (at the inlet to a chamber flame tube). The reduction in chamber stream velocity helps to improve combustion stability and to reduce hydraulic losses.

In construction, diffusers 3 (see Fig. 101) are cast aluminum-alloy ducts (throats), while in annular or cannular chambers, they are specially shaped cast (see diffuser 6 in Fig. 102) or steel welded (see diffuser 1 in Fig. 103) throats. In comparison with cast diffusers, welded diffusers are lighter in weight and are thus very commonly found in present-day GTD.

Flame tubes are designed to facilitate the processes of combustion and mixture of combustion products with diluent air. The combustion process takes place in zone A (see Figs. 101 and 102), and the mixing process in zone B.

In designing flame tubes, it is necessary to see that:

– the flame is stable under all flight conditions;

– the fuel is burned as fully as possible;

– gases and air are mixed reliably and the necessary stream temperature distribution is obtained over the tube length and cross section;

– chamber parts are cooled to the degree necessary and there is no extreme nonuniformity in heating, which would result in impermissible thermal stresses.

For stable combustion of kerosene with high combustion efficiency, the best possible atomization, mixing, and fuel-vaporization conditions must be created for the air stream, and a mean stream velocity not exceeding 15-20 m/sec must be provided in the combustion zone together with the maximum possible gas temperature, i.e., roughly 1900-2000°C.

Maximum gas temperatures are obtained by burning the fuel with an air-fuel ratio close to unity. The temperature ahead of the turbine is subsequently reduced (to the temperature permitted by turbine strength conditions) through the addition of diluent air to the combustion products in the mixing zone. As the temperature increases, the relative amount of diluent air is reduced.

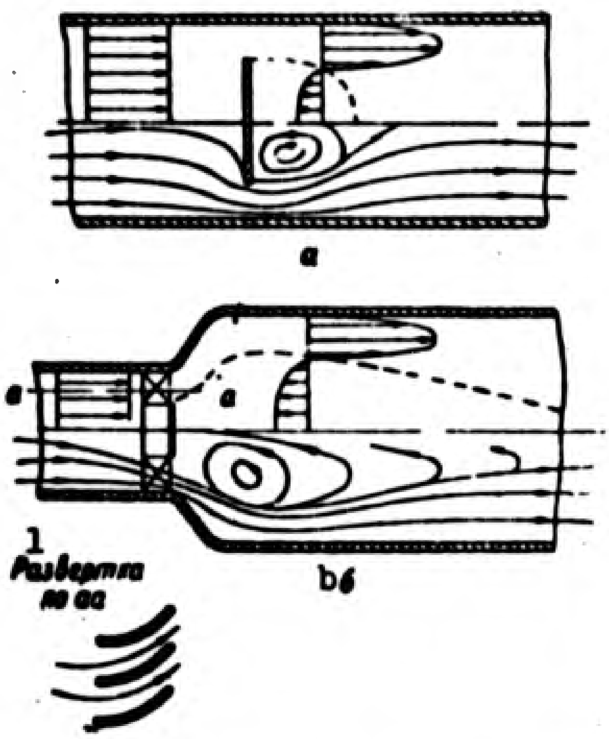

Fig. 104. Formation of flow-reversal zones: a) behind plate; b) behind swirler. 1) Section through aa.

By setting up flow-reversal zones, it is possible to obtain the required flow-velocity reduction at the front of the flame tubes together with improved air-fuel mixing and provision for the required supply of heat to raise the fuel temperature and vaporize it. Flow-reversal zones appear behind a blunt body (Fig. 104a) or after a swirler (Fig. 104b). The nature of the air stream in flow-reversal zones is clear from the figure, which shows the flow lines and the pattern of variation in the axial stream velocity components over the tube cross section. The boundaries of the flow-reversal zones are indicated by dashed lines.

The devices used to create flow-reversal zones have come to be called flameholders.

Swirlers are most frequently found in main GTD chambers as the flameholders. The hydraulic losses in swirlers are considerably less than in other types of flameholder. Swirlers (Fig. 105a) are blades stamped from sheet material and welded to an inside and outside ring. The outside ring of the swirler is in turn welded to the wall of the flame-tube head at the front, while the inside ring is used to center the fuel-burner case. About 8-10% of the total air flow passes through the swirler.

The dimensions of the flow-reversal zone behind a swirler is determined by the intensity with which the air is deflected. This depends on the number of blades and the angle at which they are mounted.

In chambers in use, there are usually 5 to 12 swirler blades installed at angles φ ranging from 30 to 80°.

The flame may be stabilized by means of cones installed at the front of the flame tube so that reverse air flows form behind them. Such a device is shown in Fig. 105b.

A basic drawback to this type of flameholder is the increased hydraulic resistance and insufficient operating reliability owing to the possibility of deposition of carbon in the space between the cones, with resulting interruption in the formation of flow-reversal zones.

It is also possible from the design viewpoint to stabilize the flame by forming reverse flows near the front wall of the flame tube,

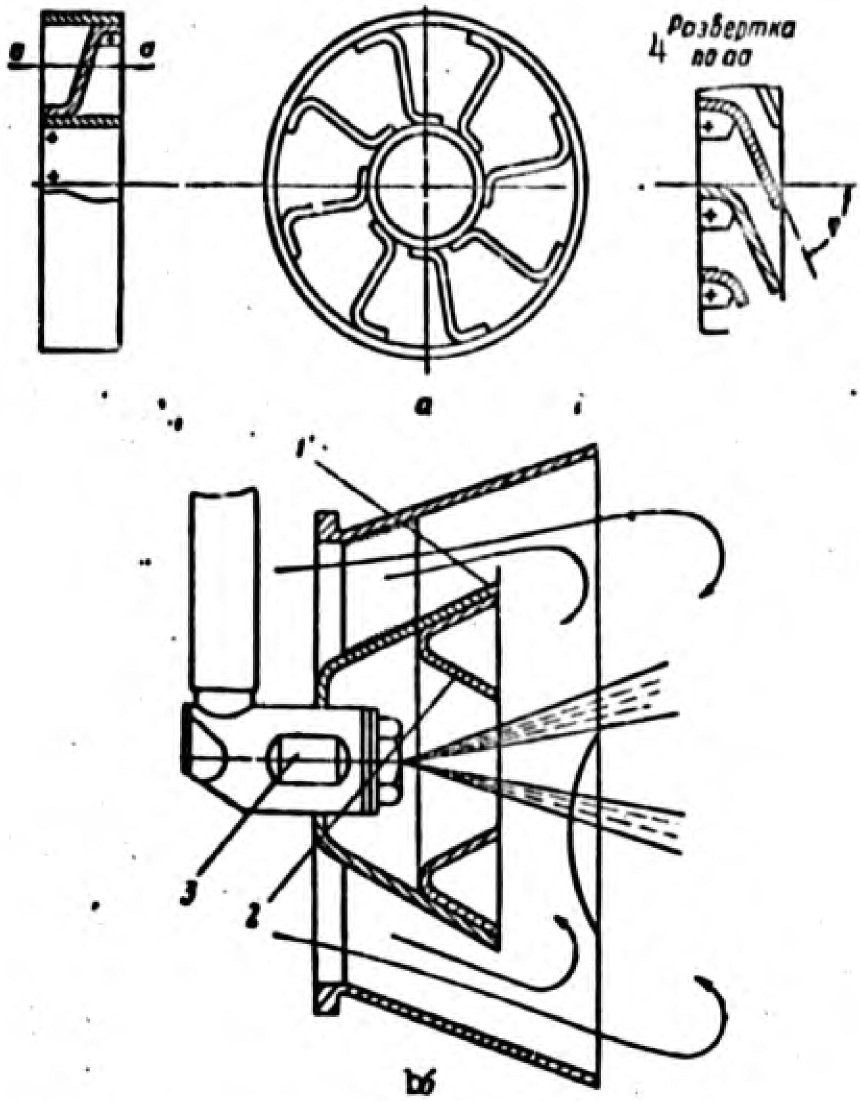

Fig. 105. Flameholders: a) swirler; b) cone system; 1) outside cone; 2) inside cone; 3) injector; 4) section through aa.

which has special apertures 1 (Fig. 106). The apertures are made by cutting through the wall and bending the material. The flow cross sections, shape, and arrangement of the apertures, which determine the dimensions of the flow-reversal zone are obtained experimentally. The techniques required to make flame tubes with such flameholders do not raise major production difficulties; in service, however, they may deform owing to loss of stability in the presence of a slight temperature rise or impairment of cooling. Thus combustion chambers using such flameholders are not widely used.

Mixing of combustion products leaving the combustion zone with the flow of diluent air usually occurs at the rear of the flame tubes. The diluent air may enter through ports in the flame-tube wall, which may be quite varied in shape and size (Fig. 107a, b, c, d, e, and f). In annular combustion chambers, mixing tubes are frequently used for this purpose (Fig. 107g and h).

It is advantageous from the viewpoint of tube-design simplification to have air enter through ports in the flame tubes. Port size and shape should be so selected as to provide good air-gas mixing and the least possible nonuniformity in tube-wall heating. It is necessary to ensure uniform tube heating in order to reduce thermal stresses. This is done by using additional vents of small size (Fig. 107f) between the main tube ports or by using special tubing (Fig. 107c) of heat-resistant material to frame the aperture edges.

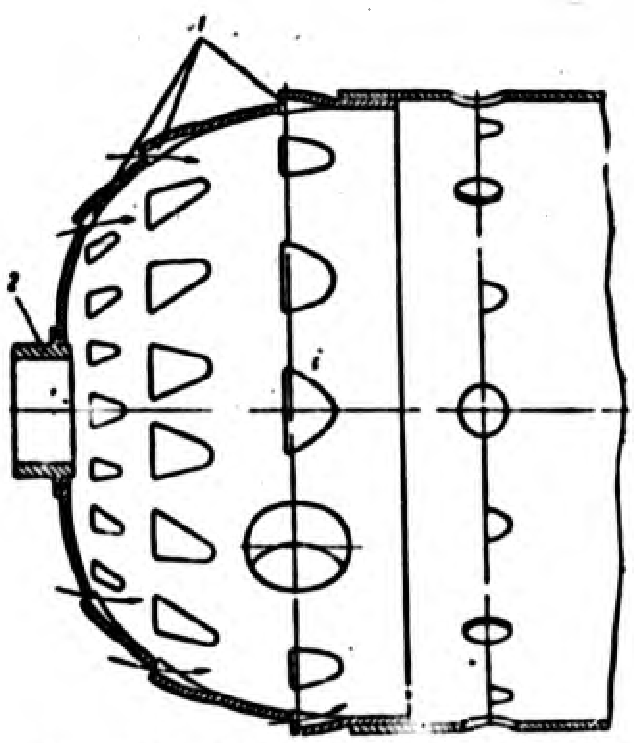

Fig. 106. Flame-tube head: 1) air-inlet apertures; 2) bushing for centering nozzle.

The nonuniform heating of the tube wall owing to better cooling of the wall material near the vents leads to the appearance of thermal stresses. This is explained by the fact that the cooler portion of the tube hinders expansion of the hotter portion. Thus compressive stresses appear in the relatively hot sections of the tube (between the apertures), while the cooler sections (at the edges of the apertures) are subject to tensile stresses that facilitate the formation of cracks.

The drawbacks inherent in the method of supplying air through ports in the flame tubes include the short range of the air stream. In this connection, it is difficult to obtain the required reduction of temperature in the center of the stream, especially with large tube cross sections. It is for precisely this reason that the method of supplying air through ports in the flame tubes is most commonly used for can and cannular combustion chambers, where the cross section is small and air can be supplied around the entire tube perimeter.

In flame tubes of annular combustion chambers in low-power engines, we also sometimes find ports used to supply air to the mixing zone; elongated ports (Fig. 107b) are used to provide some increase in air-stream range.

The use of mixing tubes (Fig. 107g and h) to supply air to annular chambers makes it possible to obtain more uniform velocity and temperature fields over the cross section at the combustion-chamber outlet.

The installation of tubes in a high-temperature air stream requires special measures to be taken for cooling the tubes and protecting them against gas corrosion, especially at the front. To do this, we usually increase the speed of the air washing the front wall of the tube by reducing the duct cross section with the aid of a special insert A (Fig. 107h).

In addition to internal tube cooling, it is also possible to create a protective layer of cold air that washes the outside surface. To do this, special attachments are installed ahead of the tube, or vents are made to admit cooling air. To protect mixing tubes against gas corrosion, their inside surfaces are sometimes coated with a layer of heat-resistant enamel [27].

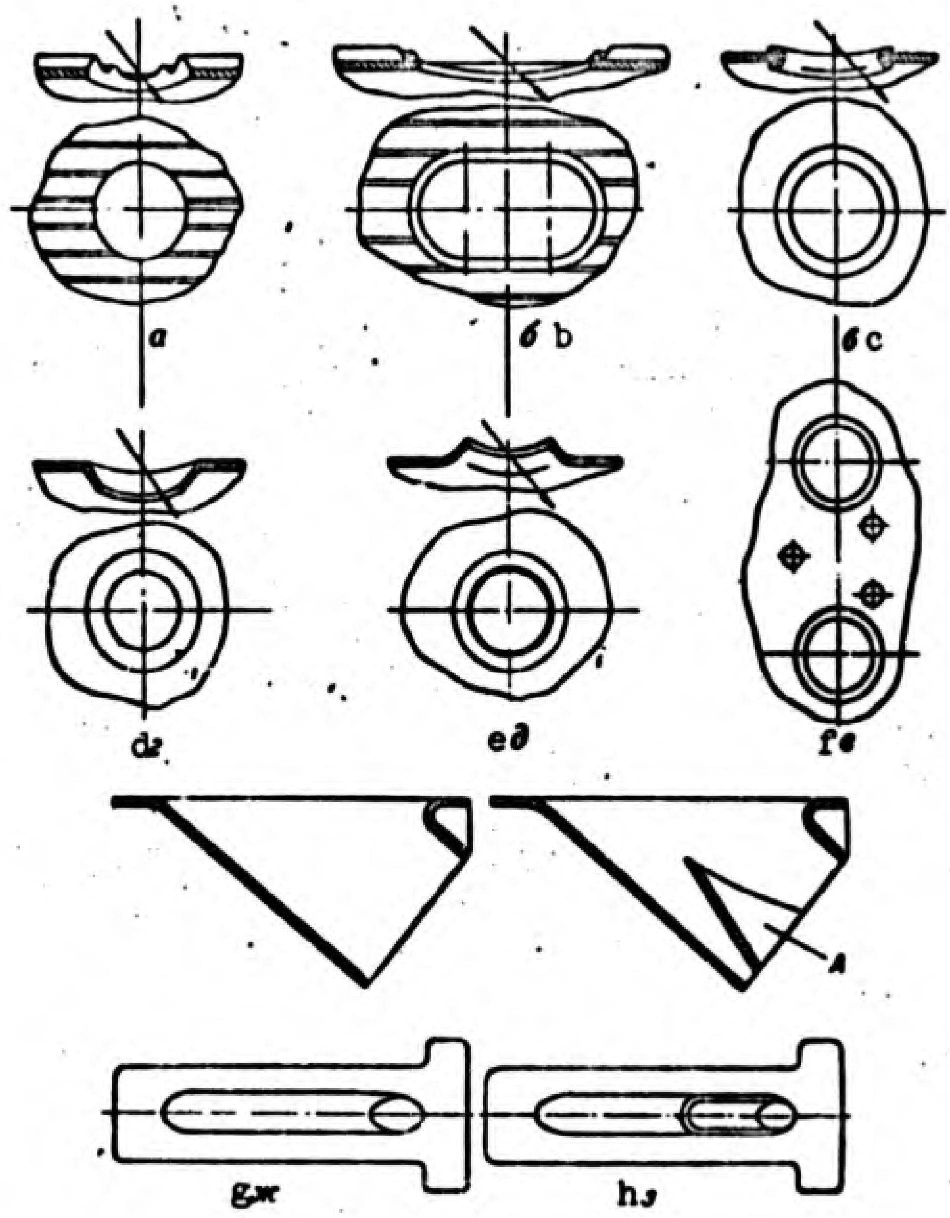

Fig. 107. Shapes of ports and ducts used to supply air to flame-tube interiors: a, b, c, d, e, and f) various shapes of ports in flame-tube walls; g and h) mixing tubes; A) insert to reduce tube-channel cross section.

Cooling of flame-tube parts to increase operating reliability and life is one of the most important problems to be solved in designing a combustion chamber. The hottest part of a flame tube is its center, since this is where combustion is completed and mixing of gases begins. The tube walls are cooled externally by the stream of secondary air flowing through the annular space between the flame tube and combustion-chamber housing (casing). External cooling frequently proves inadequate and it becomes necessary to provide for internal boundary-layer cooling. This essentially consists in creating a layer of air between the flame-tube wall and the gas flow. The air stream is di-

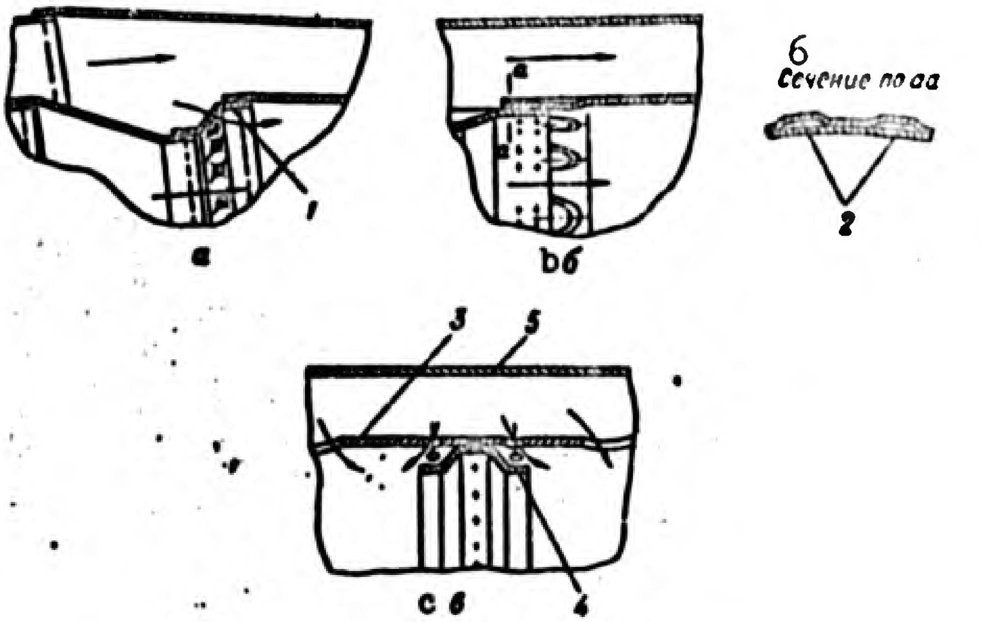

Fig. 108. Methods of supplying cooling air for boundary-layer cooling: 1) ring with vents; 2) slots; 3) duct; 4) shaped ring; 5) housing; 6) section through aa.

rected along the inside flame-tube wall to rings 1 (Fig. 108a), which connect the separate tube sections; they have vents, external slots 2 (Fig. 108b) stamped at the tube-section junctions, or shaped rings 4 (Fig. 108c) welded to the inside of the flame tube 3.

Nonuniform heating of flame tubes either along the surface or within the walls will lead to the appearance of excessive thermal stresses in the tubes or to loss of tube stability.

In order to improve cooling efficiency, flame tubes are made with small wall thickness (0.8-1.2 mm) from heat-resistant sheet material.

The greatest stiffness and stability are possessed by flame tubes having milled external longitudinal fins (Fig. 109). By bending the edges of the ports toward the inside or outside of the tube (see Fig.

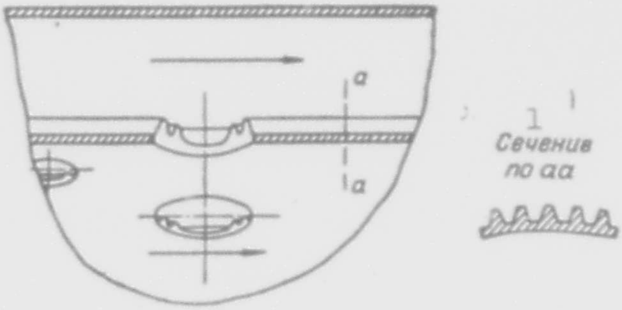

Fig. 109. Flame tube with longitudinal fins on outside surface. 1) Section through aa.

107d and e), or by installing rings (Fig. 107c), it is possible to increase the stiffness of smooth tubes.

The increased tube-wall thickness at welded seams increases thermal stresses, which can be lowered by introducing the temperature-compensating slots A (Fig. 110).

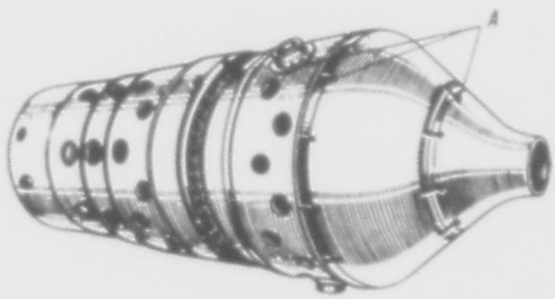

Fig. 110. Outside view of flame tube with temperature-compensation slots A.

Tubes are sometimes covered with a layer of heat-resistant enamel to increase their heat resistance.

Type EI435 chrome-nickel alloys are usually used in the manufacture of flame tubes; they have good welding and drawing properties.

<u>Combustion-chamber housings or casings</u> are cylindrical or conical shells within which the flame tubes are mounted. In contrast to casings, combustion-chamber housings are included in the engine-stress bearing system, and thus carry additional forces and moments that appear in other engine elements.

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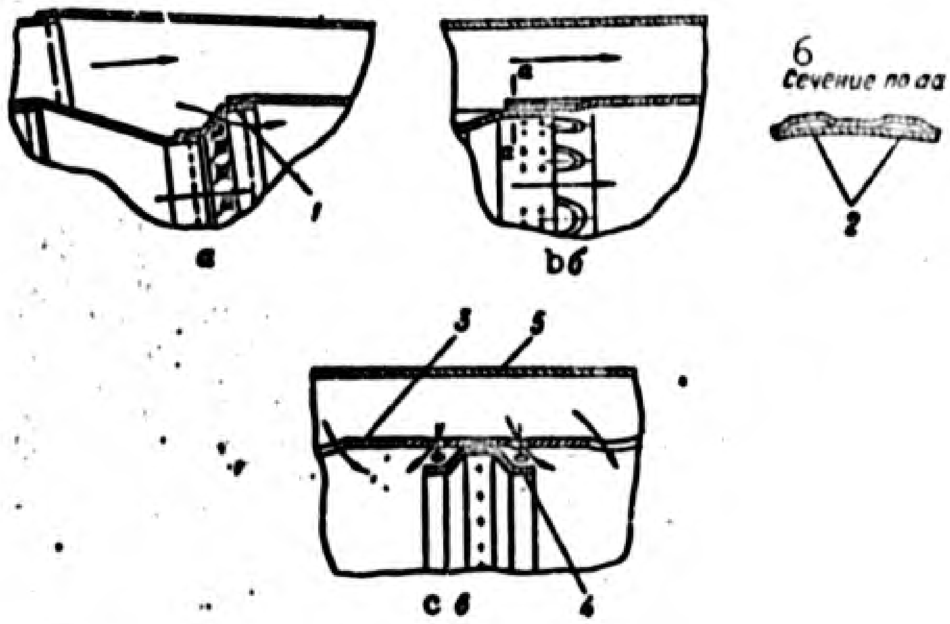

Fig. 108. Methods of supplying cooling air for boundary-layer cooling: 1) ring with vents; 2) slots; 3) duct; 4) shaped ring; 5) housing; 6) section through aa.

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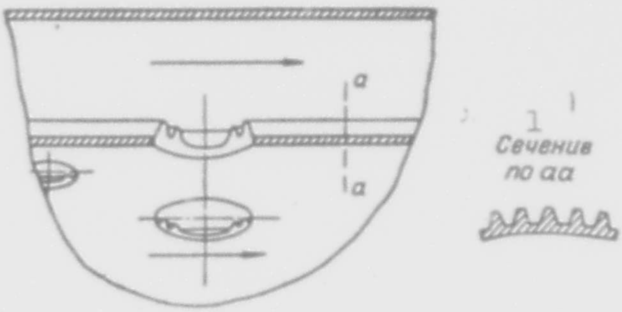

Fig. 109. Flame tube with longitudinal fins on outside surface. 1) Section through aa.

107d and e), or by installing rings (Fig. 107c), it is possible to increase the stiffness of smooth tubes.

The increased tube-wall thickness at welded seams increases thermal stresses, which can be lowered by introducing the temperature-compensating slots A (Fig. 110).

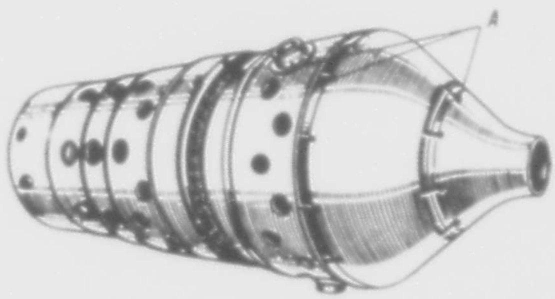

Fig. 110. Outside view of flame tube with temperature-compensation slots A.

Tubes are sometimes covered with a layer of heat-resistant enamel to increase their heat resistance.

Type EI435 chrome-nickel alloys are usually used in the manufacture of flame tubes; they have good welding and drawing properties.

Combustion-chamber housings or casings are cylindrical or conical shells within which the flame tubes are mounted. In contrast to casings, combustion-chamber housings are included in the engine-stress bearing system, and thus carry additional forces and moments that appear in other engine elements.

Casings are used in can-type combustion chambers and housings in annular and cannular chambers.

Casings of can chambers usually have a movable telescoping connection at the front (Fig. 111a) or rear (Fig. 111b) ring. The movable connection for the front part of the casing may be sealed with the aid of a rubber ring 4 installed under slight tension in a groove in the casing 3 (see Fig. 111a).

The movable joint at the rear end of the casing, which is at the hottest part of the combustion chamber, requires a more complicated sealing device. As we can see from Fig. 111b, a machined ring 6 with a spherical chrome-plated outside surface is welded to the end of the conical casing section for this purpose. A bushing 5 with chrome-plated inside surface is installed in housing 7 of the gas collector. Ring 6 and bushing 5 are assembled under slight pressure. The chrome-plated mating surfaces of the seal increase the wear resistance of the parts and prevent the gap from enlarging during operation of the chamber. The nonmovable front end of the casing is mounted as a restricted hinge (see element A in Fig. 111b). Here flange 8 of the chamber is attached to flange 9 of the compressor duct with the aid of two bolts. The connection remains sealed under possible rotations of the chamber owing to the installation of ring 10, which has a spherical contact surface.

Chamber housings in present-day GTD are shells welded to flanges designed to attach the housing to other stress-carrying elements of the engine. This connection is usually so made that it is possible to remove the flame tubes for inspection or replacement without substantial disassembly of the engine. Certain chamber designs, however, make it possible to connect the outside housing 1 of the chamber to the turbine housing 2 by means of flanges (Fig. 112) so that by moving

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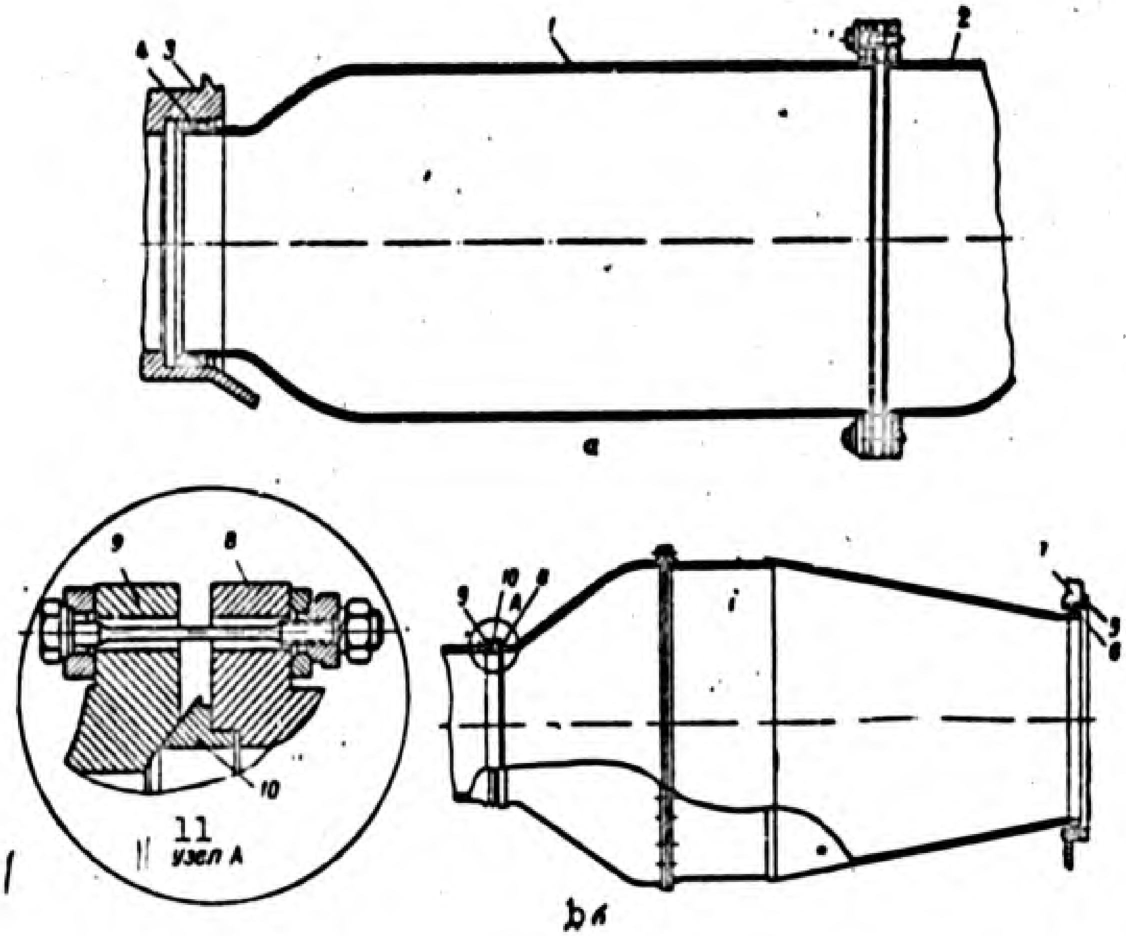

Fig. 111. Casing mounts for can-type combustion chambers: a) axial positioning of rear ring; b) axial positioning of front ring; 1) casing; 2) gas collector; 3) housing; 4) rubber ring; 5) bushing; 6) ring; 7) gas-collector housing; 8) chamber flange; 9) duct flange; 10) spherical ring; 11) detail A.

housing 1 backward it is not only possible to inspect the flame tubes 3, but where necessary to replace them without additional disassembly of the engine.

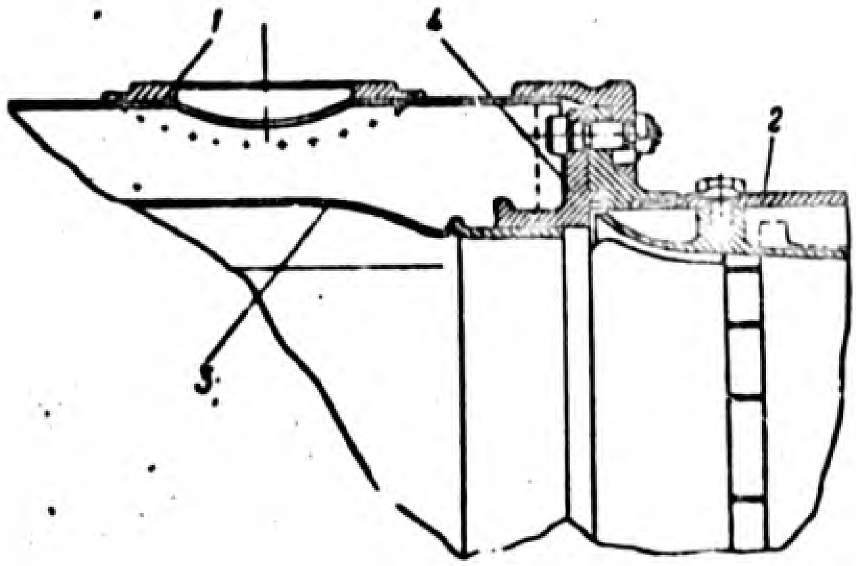

Fig. 112. Detail showing connection of combustion-chamber housing and turbine housing: 1) chamber housing; 2) turbine housing; 3) flame tube; 4) stress-absorbing ring.

Combustion-chamber casings and housings are made of St10 carbon steel where the wall temperatures do not exceed 300°C, while YalT (1Kh18N9T) chrome-nickel steel is used for higher temperatures.

3. LOADS ON CHAMBER ELEMENTS, STRENGTH OF CHAMBER ELEMENTS, AND VIBRATION

Combustion-chamber elements are stressed by gas and inertial forces, gravitational forces, and by forces due to nonuniform heating of material and vibration. In addition, where chamber elements are connected into the engine stress-carrying system, additional stresses appear due to forces and moments transmitted from other engine elements.

Flame tubes are subject to the action of a small air-pressure drop; their principal loads are of thermal origin and are caused by nonuniform wall heating. The prediction of wall-temperature distribution nonuniformity and determination of thermal stresses is a fairly complicated problem. Thus as a rule, flame tubes are not designed for strength. Reliable operation is ensured by a reasonable tube design

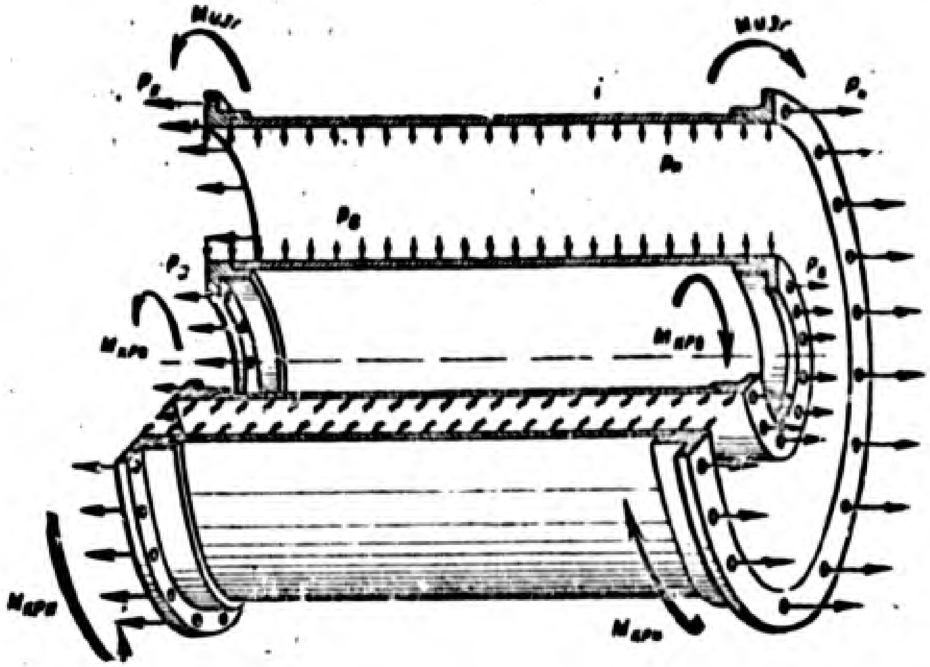

Fig. 113. Loading of outside and inside rings of annular and cannular combustion chambers.

based on actual tests either in an engine or on special setups.

Casings for can chambers are designed for strength, and the effect of the internal excess pressure and vibration experienced is taken into account.

When housings for annular and cannular chambers are designed, they are assumed to be loaded by a complex system of forces and moments. Figure 113 shows the loads on the outside and inside housings of an annular chamber. The outside housing experiences tension along the generatrices owing to the action of the internal excess pressure p_n, and tension through the cross section as a result of the axial force P_n transmitted from the turbine housing, and torsion from the twisting moment $M_{kr\ n}$. The inside housing is compressed by the external excess pressure p_v and is acted on by the axial force P_v and twisting moment $M_{kr\ v}$. In addition, gravitational forces and bending moments M_{iz} act on the housings.

Tensile stresses due to axial forces and bending moments are fairly small (less than 260-300 kg/cm^2), while tangential stresses due to the twisting moment amount to only 15-100 kg/cm^2 overall.

Thus for the outside housings, designing for strength reduces to determining the tensile stresses along the generatrices produced by the excess pressure, which reaches its maximum value in the flight mode in which there is maximum air pressure after the compressor. Thus, at maximum speed on a test stand, the tensile stresses along the generatrices of the outside housing are 1200-1400 kg/cm^2, while under maximum pressure in flight, the stresses in the combustion chamber reach 2000-2300 kg/cm^2.

Vibration in housings, casings, and flame tubes cause additional stresses and not infrequently lead to the formation of cracks and even to failure of chambers.

Shell vibrations are caused by periodic variation in the gas forces acting on the chamber walls; the forces are produced by the pressure drop. Maximum exciting-force amplitudes are observed when:

– the compressor operates unstably, and also when it approaches surge operation;

– spasmodic burning appears in the chamber;

– the fuel supply becomes uneven.

During vibration, the amplitudes and, consequently, the vibration stresses reach their maximum values at resonance, when the natural vibration frequency of the shell becomes equal to the frequency of the exciting forces. Since the exciting-force frequencies are multiples of the rpm, and the natural vibration frequencies depend on the type of oscillation, and the geometric dimensions and properties of the shell material, any factors raising the natural vibration frequency will move the resonance frequency toward higher speeds and vice versa. On this basis, it is very important to know how the natural frequency varies when structural changes are made in the combustion chamber or when its operating conditions vary.

Of all the diverse shell-vibration modes, we isolate two types – axisymmetric and bending.

<u>Axisymmetric vibration</u> is characterized by the fact that the shell cross sections remain round during the vibration process. The order of the vibration mode is determined by the number of half waves along the length of the generatrices. The first oscillation mode, with the lowest natural vibration frequency, will have one half wave (Fig. 114a), the second two (Fig. 114b), etc. As the number of half waves increases, the natural vibration frequency rises rapidly.

<u>Bending vibration</u> is characterized by the fact that the shape of the cross section changes during vibration: Vibration occurs with the formation of waves along the circumference with several nodes along generatrices. The lowest (first) mode (Fig. 115a) corresponds to two waves and four generatrix nodes, the next mode to three waves, and then four waves (Fig. 115b), etc.

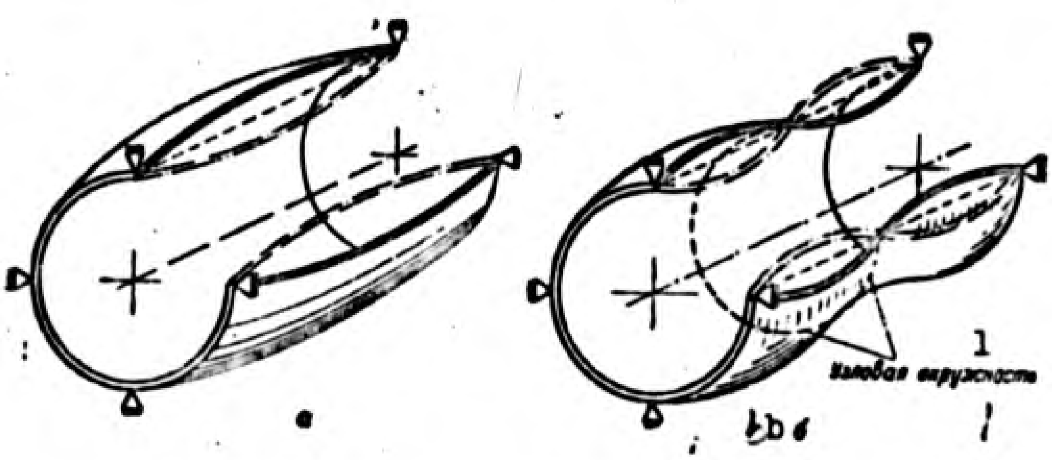

Fig. 114. Axisymmetric shell-vibration modes: a) vibration with one half wave along generatrices; b) vibration with two half waves along generatrices. 1) Circumferential node.

As in the first vibration, each vibration mode corresponds to a particular frequency. As the vibration mode becomes more complicated with increasing number of waves, there is an increase in the natural vibration frequency. Chamber rupture occurs most frequently at resonance for the first two modes of axisymmetric or bending vibration.

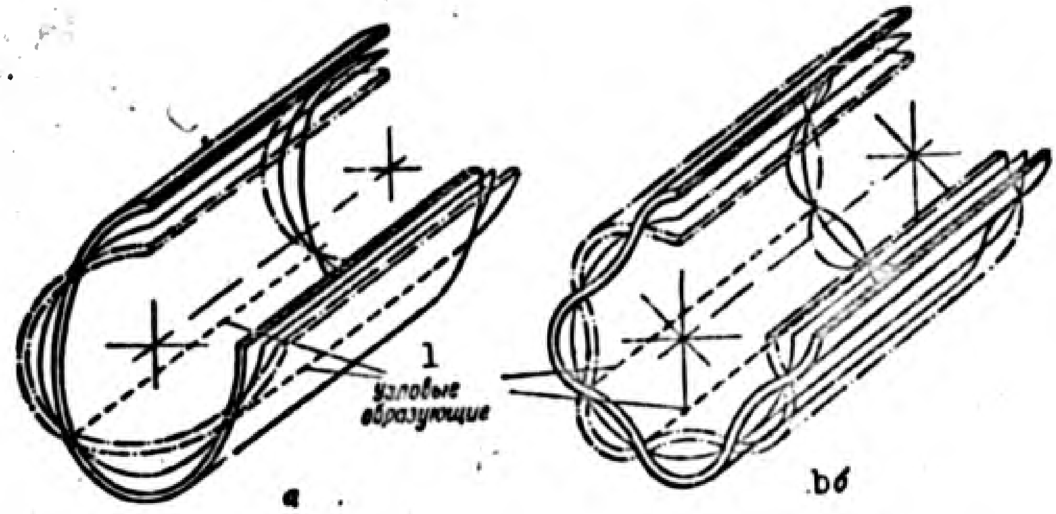

Fig. 115. Shell bending-vibration modes: a) vibration with two circumferential waves; b) vibration with four circumferential waves. 1) Generatrix nodes.

Shell geometric dimensions frequently affect the natural vibration frequency. Thus, for example, as the radius or length of the shell increase, the natural vibration frequency in any mode will decrease, while as wall thickness increases, the frequency will rise. An increase in temperature, which reduces shell stiffness, will reduce the natural vibration frequency. An increase in the internal excess pressure or in axial tensile forces will raise the natural vibration frequency, just as the natural frequency of a string increases when it is placed under tension; finally, an increase in external excess pressure and compression of the shell by axial forces will, on the other hand, reduce the frequency of vibration.

Thus, resonance regimes, which are very dangerous owing to the appearance of vibration stresses, may appear as a result of variation in operating conditions, and especially when the rotor speed, flight altitude or speed, and ambient temperature vary.

Various methods of varying the natural vibration frequencies or reducing the exciting-force amplitudes may be used to suppress dangerous resonance modes.

If during the final steps of chamber manufacture, the wall thicknesses or other shell dimensions (radius, length) are changed or longitudinal and transverse stiffeners are installed, the natural vibration frequencies will vary. To reduce the exciting-force amplitudes, we recommend that the engine not be allowed to operate for long periods of time near the surge limit, nor should the fuel system be permitted to operate with large pressure fluctuations in the fuel ahead of the nozzles, so that the appearance of so-called spasmodic burning can be avoided.

Chapter 8

EXHAUST SECTIONS. AFTERBURNERS

1. GENERAL INFORMATION

Exhaust sections of gas-turbine engines are most frequently designed to exhaust the gases behind the open end of the aircraft fuselage or motor nacelles in a nearly axial direction, and to create the required back pressure for the turbine with subsequent expansion of the gases at the end of the exhaust section. The exhaust gases should be directed axially to provide maximum engine thrust, since thrust depends on the axial component of the gas velocity. The back pressure behind the turbine determines the state of the gas in the combustion chamber and turbine efficiency.

Depending on function, all types of GTD [gas-turbine engine] exhaust sections may be classified into the exhaust sections themselves, afterburners, and thrust reversers.

The basic elements of GTD exhaust sections are the nozzles installed at the rear.

If in addition to directing the flow and expanding the gases, the exhaust section provides additional fuel combustion following the turbine in order to augment the engine thrust, the exhaust section is called an afterburner.

Negative engine thrust, opposing the direction of flight, may be used to brake the aircraft. This is done by turning the stream of gases leaving the engine. A 180° change in the thrust direction is called thrust reversal, and thus the exhaust devices that turn the gas jet in order to vary the direction of thrust have come to be called <u>thrust reversers</u>.

GTD exhaust sections as a rule consist of an <u>exhaust pipe</u> 1 (Fig. 116) and <u>nozzle</u> (orifice) 3. The length of the exhaust section is determined by the location of the engine on the aircraft. Where the exhaust pipe is short and the nozzle is far from the turbine, the necessary length of the exhaust section may be obtained with the aid of an <u>extension pipe</u> 2, located between the exhaust pipe and the nozzle.

The way in which the exhaust-section duct area varies with length and the shape of the individual elements are dictated by the conditions that ensure minimum hydraulic losses.

To prevent abrupt expansion of gas behind the turbine in the exhaust pipe, an inner cone 2 (Fig. 117) which is the fairing for the back of the turbine wheel is installed. The profile of the exhaust-pipe duct depends on its length and the velocity of the gas stream at the turbine exhaust. Where the gas-stream velocity is low, minimum losses under design conditions are achieved in a short exhaust section when the exhaust-pipe duct profile is governed by a law that ensures constant cross-sectional area throughout the length of the pipe.

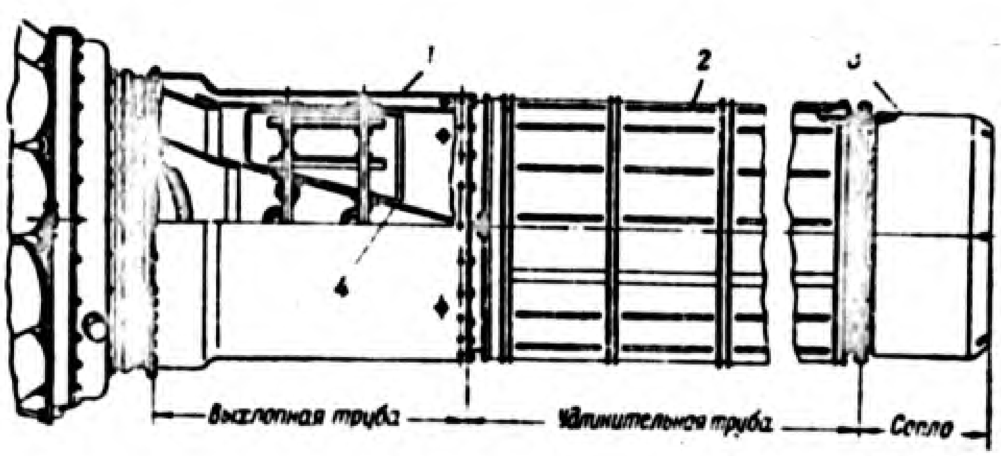

Fig. 116. Exhaust section for TRD [turbojet engine]: 1) exhaust pipe; 2) extension pipe; 3) nozzle; 4) inside cone; 5) exhaust pipe; 6) extension pipe; 7) nozzle.

For high-speed gas flows at the turbine outlet, where the Mach

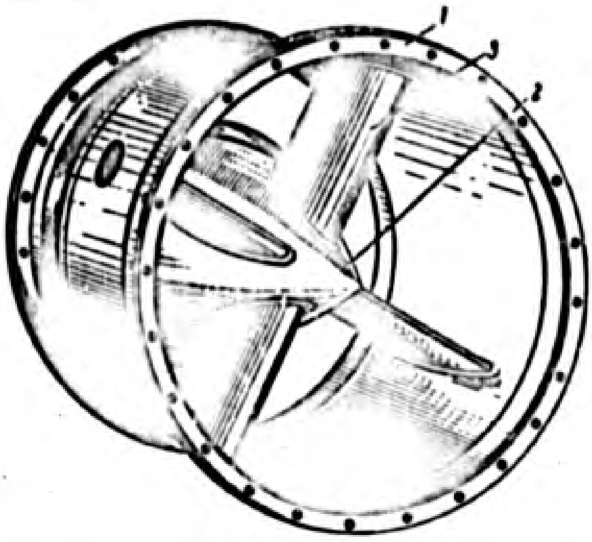

Fig. 117. Outside view of exhaust pipe: 1) outside pipe; 2) inside cone; 3) strut.

number for the flow exceeds 0.6, and where there is an extension pipe following the exhaust pipe, it is desirable for the exhaust-pipe duct to be divergent, since a reduction of gas velocity at the end of the exhaust pipe will lead to a reduction in the total losses in the exhaust section. The diffusion efficiency of the duct is usually small. The gas velocity at the extension-pipe entrance is reduced to 100-150 m/sec.

The radial struts 3 (Fig. 117), connecting the inside cone to the outside pipe 1 are streamlined. Where the gases leaving the turbine are turned through large angles, struts of nonsymmetric profile are used to turn the flow with minimum loss. As a rule, few struts are used (4 to 6), but it may be necessary to install a straightening assembly following the turbine. In this case, 10 to 16 or more radial struts may be used.

Depending on the amount of expansion required in the exhaust section, which is determined by the ratio of the pressure ahead of the nozzle to atmospheric pressure, convergent (subsonic) or convergent-divergent nozzles are used. Both convergent and convergent-divergent nozzles may be of the fixed-area or variable-area types.

Subsonic convergent nozzles are used for the most part on modern GTD with flight speeds not exceeding Mach 1.5-2.0.

Convergent-divergent fixed-area nozzles (Fig. 118), are also called Laval nozzles, may be employed in engines designed for supersonic aircraft.

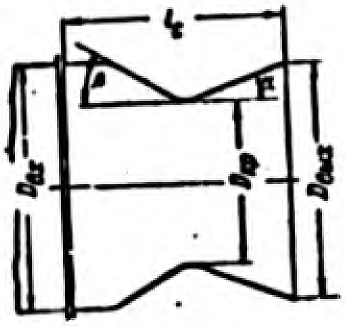

Fig. 118. Diagram of convergent-divergent nozzle.

Variable-area nozzles in which the dimensions of the exit and throat sections vary improve engine characteristics and pickup, and also make engine starting easier.

Variable-area nozzles have the following drawbacks: structural complexity and increased weight. Variable-area nozzles are thus used only when they permit a substantial improvement in engine characteristics. Variable-area nozzles are always used in engines with afterburners.

Structural shapes and materials used in fabricating exhaust-section elements are chosen chiefly on the basis of operating conditions and the nature of the loads carried. Here it must be remembered that elements of the exhaust and extension pipes as well as those of the nozzle may be heated to temperatures of 600-650°C by the gas jet. A nonuniform temperature field at the tube-wall surfaces will produce thermal stresses, while periodic variation in gas pressure will excite vibration in thin-walled exhaust-section shells.

2. DESIGN OF EXHAUST AND EXTENSION PIPES

Exhaust pipes of TRD consist of an outside pipe, inside cone, and connecting elements.

The outside pipe 1 (Fig. 119) is a cylindrical or conical shell made from heat-resistant sheet material with welded flanges used to attach the exhaust pipe to the turbine housing on one end and to the extension pipe or nozzle on the other. If the extension pipe is long,

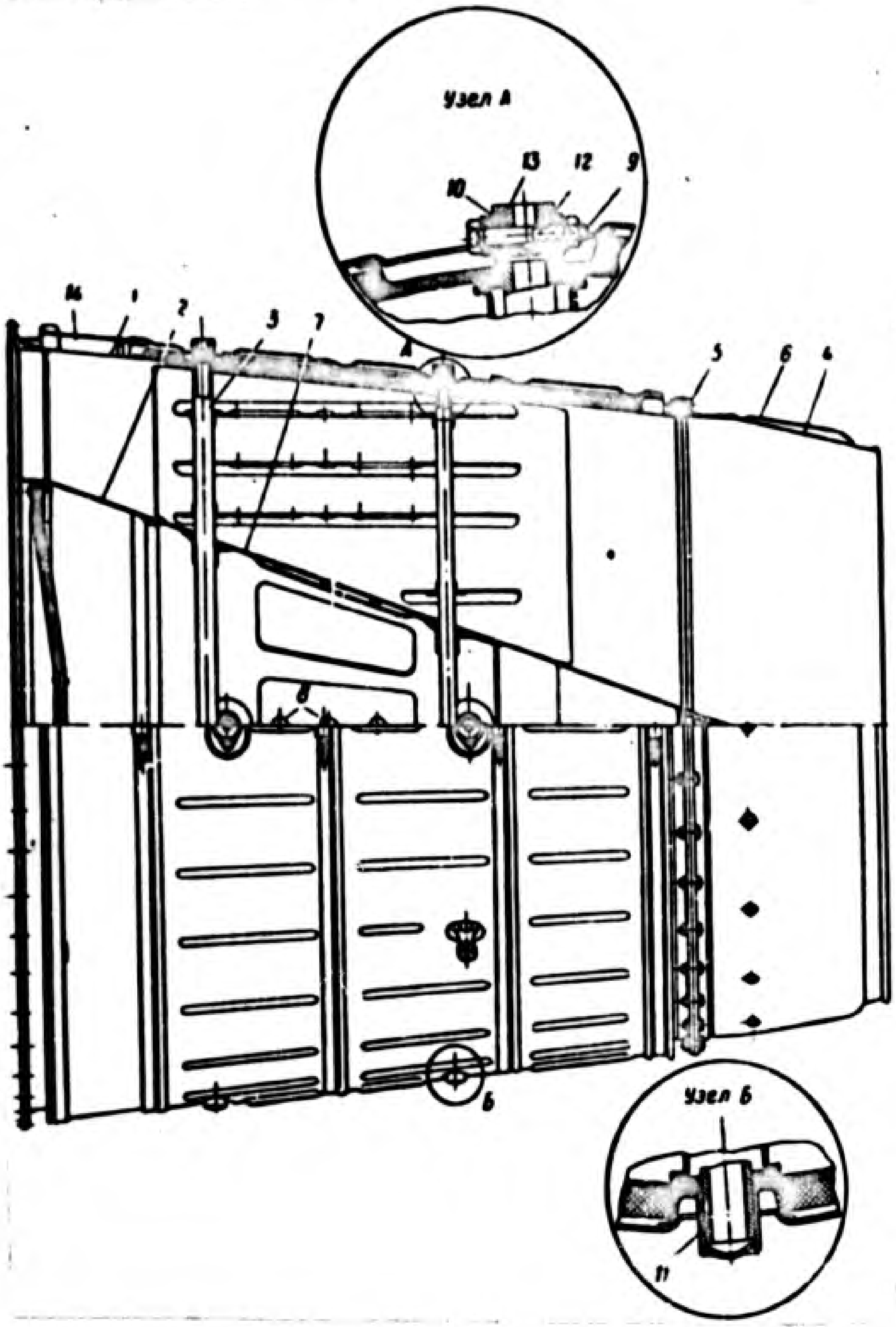

Fig. 119. TRD exhaust section without extension pipe: 1) outside pipe; 2) inside cone; 3) stress-absorbing rod; 4) nozzle; 5) nozzle flange; 6) nozzle casing; 7) reinforcing cone; 8) vents; 9) rods; 10 and 11) bushings; 12) bolt; 13) plug; 14) cooling-air chamber; 15) detail A; 16) detail B.

it is attached to the aircraft independently of the engine. Here the extension pipe is attached to the exhaust pipe by a telescoping joint (Fig. 120). A telescoping flange joint between individual parts of the exhaust section permits relative axial displacements and small angular displacements. To prevent a gas blowout at a flanged connection, a special reflecting ring 2 is sometimes installed.

Thermocouple mounting tubes are installed on the outside tube of the exhaust system to permit the engine thermal regime to be monitored.

Increased shell strength and stiffness may be gained by installing ring shrouds and reinforcing stringers. Of extreme importance for pipe strength is the location of welded seams. Shells with oblique weld seams are better. Shells produced by this welding method are stronger than where welds are made along generatrices.

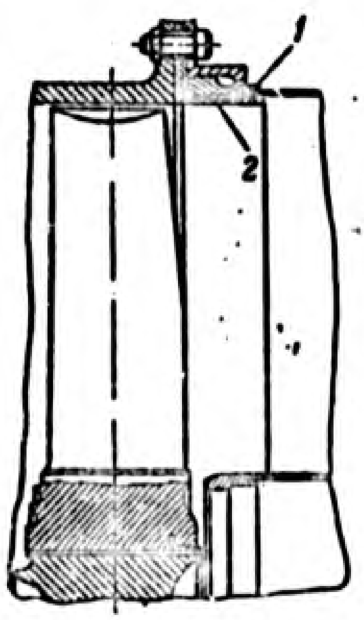

Fig. 120. Telescoping joint between outside pipe of exhaust section and turbine housing: 1) outside pipe; 2) reflecting ring.

The inside cone 2 (see Fig. 119) of the exhaust pipe is welded from heat-resistant sheet material. A bottom is attached to the cone base. Sometimes it has two walls containing a layer of asbestos. The bottom of the cone acts as a heat-reflecting shield that reduces the temperature rise at the rear of the turbine wheel due to thermal radiation from the hot parts of the inside cone. Thus there is a closed cavity within inside cones with bottoms. In order to avoid a rise in gas pressure within this cavity as the engine heats up, vents 8 are provided in the bottom or side of the cone, as a rule, under the struts (see Fig. 119). The vents connect the cone interior with the flow passage of the exhaust system, and they should be large enough to prevent any possible increase in the pressure drop at the cone wall in the presence of transients during engine operation.

In engines where the turbine is cooled by supplying cooling air to the rear of the last-stage wheel, and in multistage TVD turbines where there is little temperature rise in the last-stage wheel, there is no need to install a cone bottom.

In exhaust sections using a variable-area nozzle, the cone is made in two sections – a fixed section 2 (Fig. 121) and a movable section 3. Displacement of the cone moving section varies the nozzle cross-sectional area. In such designs, vigorous air cooling is usually provided. The path of cooling-air travel is indicated by arrows in the figure.

The inside cone is connected to the outside pipe either by struts 1 (Fig. 121) or stress-bearing rods 3 (see Fig. 119), located in mutually perpendicular planes.

Stress-bearing rods are closed in fairings that are attached to neither the outside pipe or the inside cone.

In designing attachment fittings for struts and stress-bearing rods at the outside pipe or the cone, care should be taken to see that they are free to move under thermal expansion. Thus, for example (see detail A, Fig. 119), the top end of rod 9 is attached to the pipe with the aid of a bushing 10 and bolt 12. Then the lower end (detail A) is centered just in bushing 11, relative to which the rod can move. Gas blowoff between the outside-pipe and rod bushings is prevented by installation of the special plugs 13.

Extension pipes consist of one or several sections. It is easier to manufacture several fairly short sections than one long pipe.

Where the pipe is long, a gate valve 1 (Fig. 122) is sometimes installed to make engine starting easier; it bypasses part of the gases

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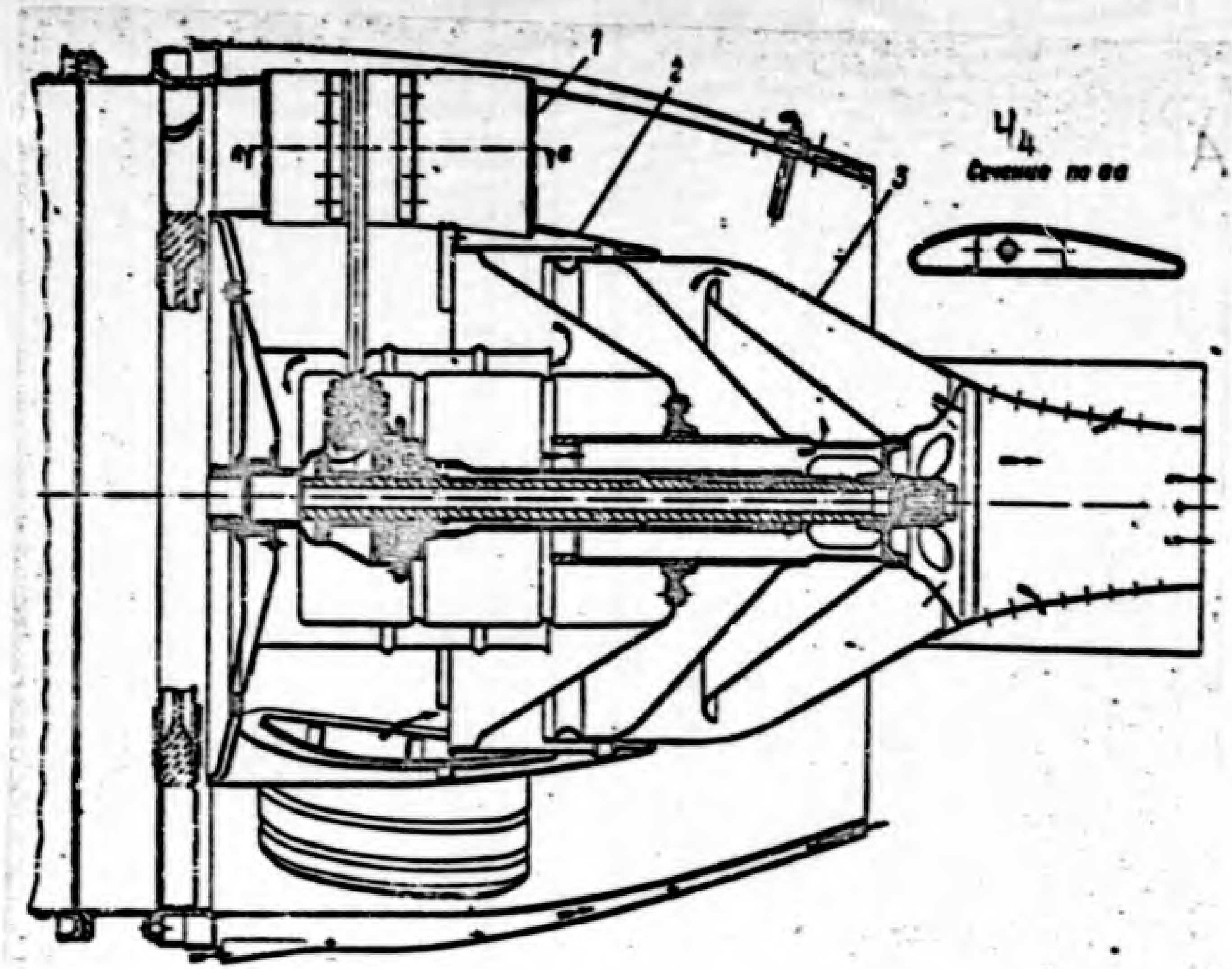

Fig. 121. TRD exhaust section: 1) cone attachment strut; 2) stationary section of cone; 3) moving section of cone; 4) section through aa.

to the atmosphere. The gate valve is located [illegible] pipe directly behind the turbine, and when it is open the hydraulic resistance of the exhaust system is reduced.

Extension-pipe elements are similar in design to outside-pipe elements. Extension pipes are provided with drains 2 to ensure that condensate runs out after the engine has been shut off or an unsuccessful attempt to start the engine has been made. The pipe is attached to the aircraft by means of fittings 3 on the extension pipe.

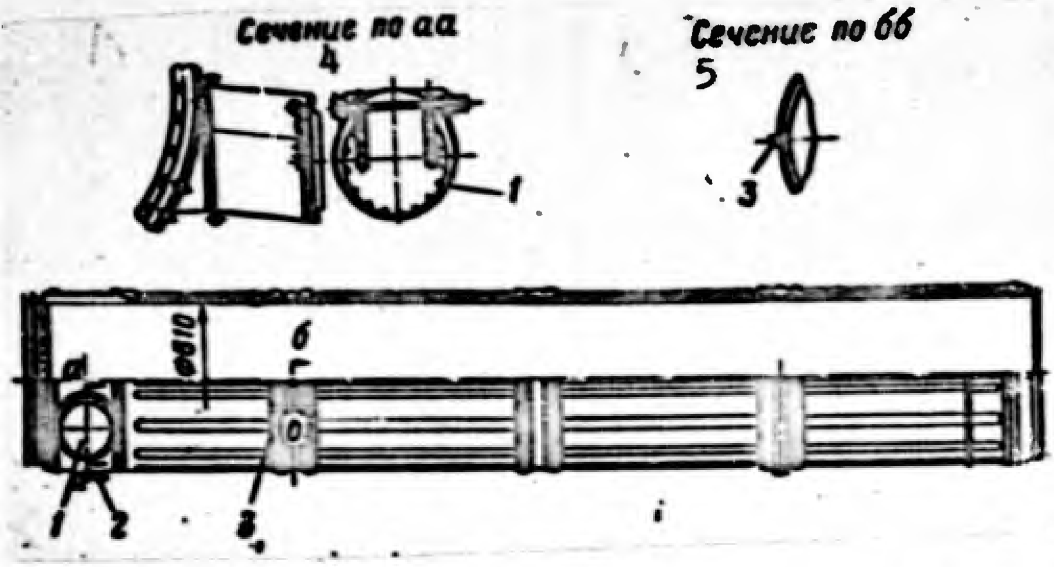

Fig. 122. Extension pipe: 1) gas-bypass gate valve; 2) drain; 3) pipe attachment fitting; 4) section through aa; 5) section through bb.

3. NOZZLE DESIGN

Fixed-area convergent nozzles are most frequently made as a conical shell welded to a machined flange 5 (see Fig. 119) which is used to attach the nozzle to the exhaust or extension pipe. The nozzle is cooled by the air passing through the annular slot between the nozzle wall 4 and the casing 6. The air is moved through the slot by the dynamic pressure and the entraining action of the gas leaving the nozzle.

Variable-area nozzles have moving elements. Hydraulic, mechanical, pneumatic, and electrical control systems are used to regulate the moving elements of variable-area nozzles.

Figure 123 shows a nozzle in which the exit cross section is ad-

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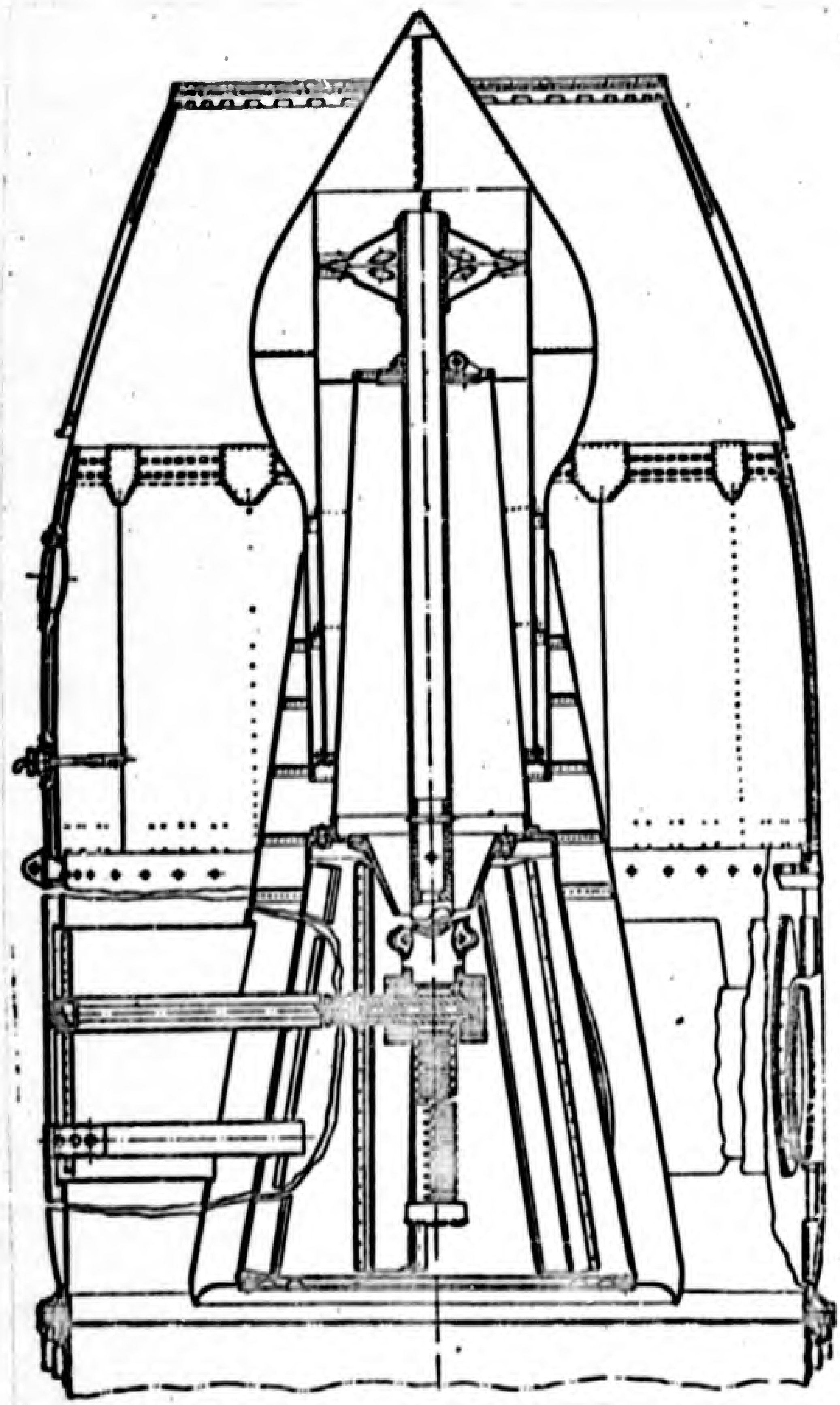

Fig. 123. Exhaust section with variable-area nozzle.

justed by means of a spindle which is moved by a hydraulic servomotor through a mechanical linkage. Failure-free operation of such a nozzle requires reliable cooling of the spindle and all elements controlling it.

Variable-area nozzles with external moving elements of the flap type are very promising. Depending on the number of flaps, nozzles are classified as twin-eyelid or multiflap nozzles.

A substantial drawback to twin-eyelid nozzles is the fact that it is impossible to keep the gas-jet cross section circular where it leaves the nozzle as the flaps take up various positions. In addition, it is difficult to obtain a reliable seal for the flaps where they join. Gas leakage through gaps in joints and variation in gas-jet cross-sectional shape lead to increased losses and reduced engine thrust. The forces acting on each flap are considerable, and they produce large loads on the flap attachment fittings, which increases the weight of the exhaust-section structure and, consequently, considerably increases the forces required to move the flaps.

Multiflap nozzles offer advantages over twin-eyelid nozzles. Where there is a large number of flaps, the gas-jet cross section remains nearly round no matter what position the nozzle flaps are in. The forces acting on each flap are less than those for a twin-eyelid nozzle. The flap attachment hinges are simplified in design. The flap loads are more uniformly distributed around the perimeter of the rear flange.

Figure 124 shows the construction of a multiflap nozzle. The flaps 1 are attached by hinges to the flange of the outside pipe, and are pressed against the flap control ring 2 by the pressure drop. An axial displacement of this ring causes the flap position to change in the following manner: when the ring moves forward, the nozzle exit area

Fig. 124. Variable-area multiflap nozzle: 1) nozzle flap; 2) flap control ring.

increases, while when the ring moves back, the exit area diminishes.

The basic difficulty in making a multiflap nozzle lies in ensuring that there is a reliable gas seal at the flap joints.

The seal may be made by filling the space at the joints with glass wool encased in an elastic material which is welded to the neighboring nozzle flaps.

A simpler joint seal involves a flap design in which the edge of one flap forms a tongue that fits into a longitudinal groove in the other flap (Fig. 125). In this case, the joints overlap owing to de-

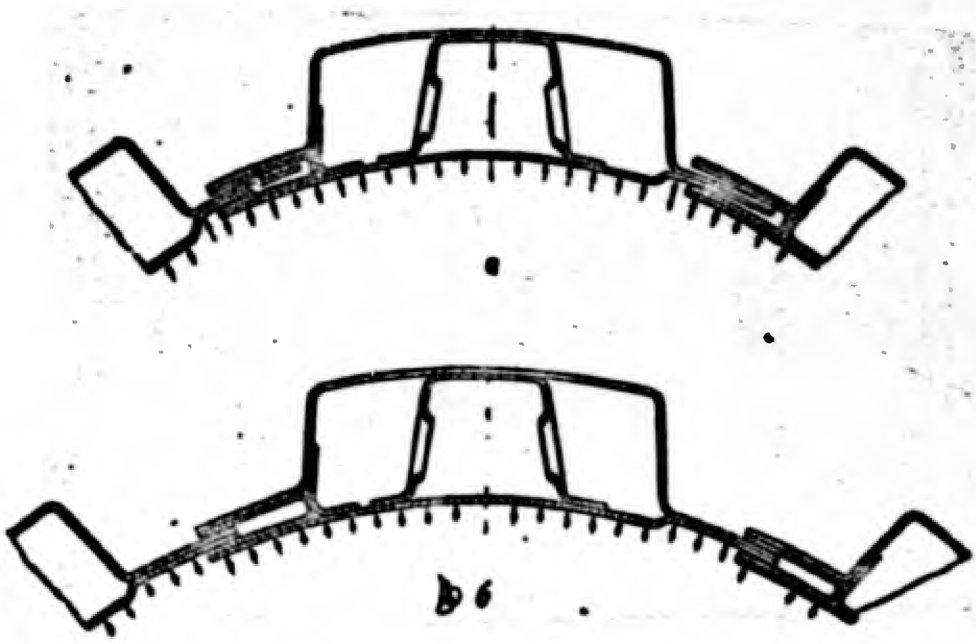

Fig. 125. Diagram showing way in which flap joints overlap for minimum and maximum nozzle cross-sectional areas: a) flap position for minimum nozzle cross section; b) flap position for maximum nozzle cross section.

formation of the tongues by the pressure of the gas, and seal reliability depends solely on the amount of overlap selected when the flaps are in the positions corresponding to minimum and maximum nozzle area.

For a given tongue thickness, an increase in the amount of overlap will reduce tongue stiffness. Thus, an increase in tongue length corresponds to a more reliable joint seal. On the other hand, an increase in the amount of overlap impairs tongue cooling conditions, and as a result slots may appear upon deformation where the flaps join.

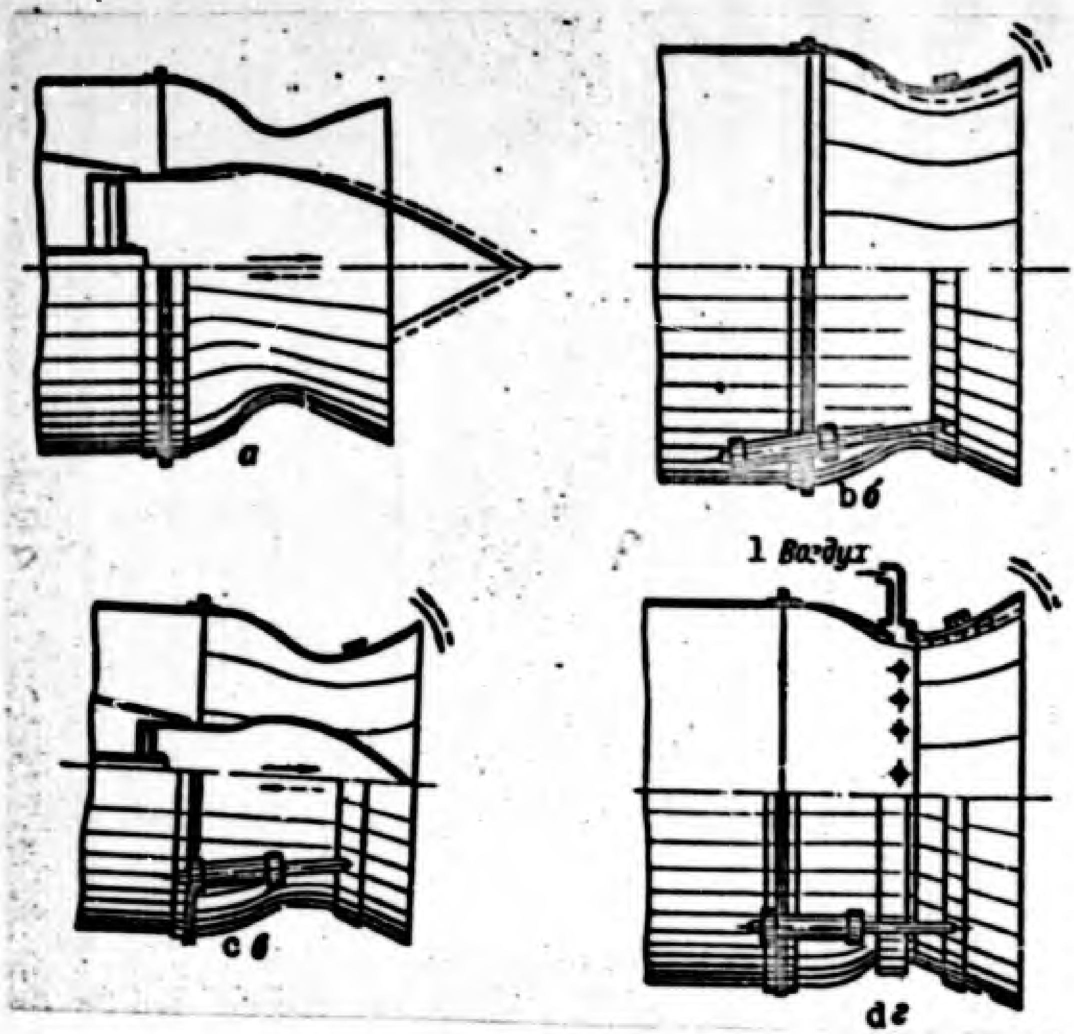

Fig. 126. Convergent-divergent nozzles [27]: a) with variable-position bullet; b) with variable-position flaps; c) with variable-position bullet and flaps; d) throat area varied by aerodynamic compression and exit area by flap rotation. 1) Air.

As a rule, the flaps are made with a box-like cross section, which ensures reliable cooling of the flaps and adequate stiffness.

Thus, while the engine is in operation, the hollow interior of a flap is cooled by the air stream.

The contacting surfaces of the flaps 1 (see Fig. 124) and ring 2 are so made as to provide the greatest contact area for any position of the ring. This is done by giving appropriate shape to the outside flap surfaces and to the ring.

The force needed to close the flaps depends on the ring cone angle. The force needed to move the ring is heavily affected by the friction forces appearing at the contact surface between ring and flaps.

Variable-area convergent-divergent nozzle designs have as yet not received wide application.

Figure 126 shows the basic arrangements of several variable-area convergent-divergent nozzles. Both the exit area and the minimum (throat) area may be regulated by changing the position of the bullet (Fig. 126a) or flaps (Fig. 126b), by simultaneously changing the position of bullet and flaps (Fig. 126c), by changing the throat area through aerodynamic compression (by a stream of air), and by changing the exit area by turning the flaps (Fig. 126d).

A variable-area nozzle produces the best effect when the throat and exit areas are varied independently, i.e., where nozzles are used that have the control systems shown in Fig. 126c and d; this, however, complicates the control system and nozzle construction. Thus a nozzle in which the throat area changes in proportion to the exit area may be used, i.e., a flap-type nozzle (Fig. 126b).

It is clearly possible to obtain a flap-type nozzle that also offers independent control of the exit and throat areas by introducing additional flap hinges at the throat together with a second control ring.

4. AFTERBURNERS

Exhaust sections for TRD that include the structural elements needed for additional combustion of fuel, which have thus come to be called afterburners, have a number of design peculiarities.

In such a TRD exhaust section, there is a divergent duct directly following the turbine – the afterburner diffuser (Fig. 127), within

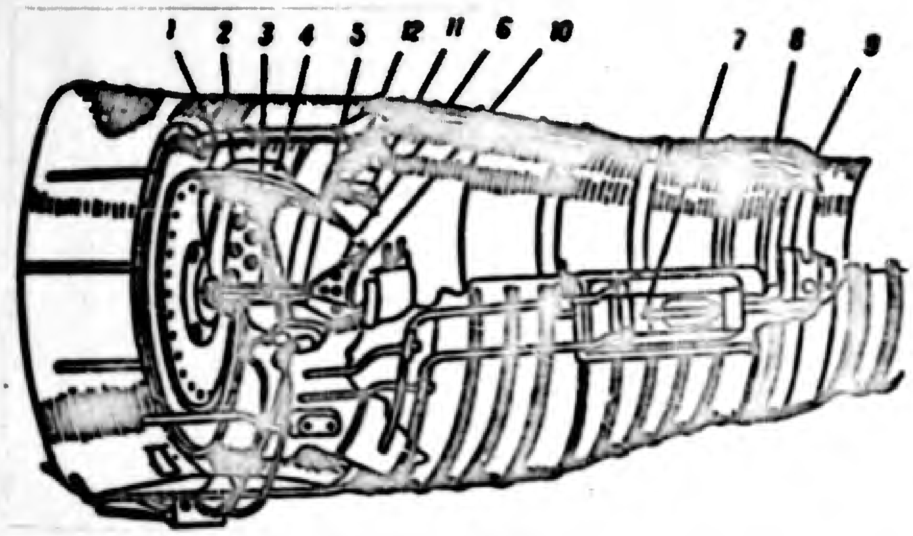

Fig. 127. Afterburner for TRD: 1) diffuser outside pipe; 2) inside cone; 3) stress-bearing rod; 4) streamlined strut; 5) starting (ignition) plug; 6) diaphragm; 7) hydraulic cylinder for flap control; 8) annular housing; 9) nozzle flap; 10) center flameholder; 11) annular flameholders; 12) nozzles.

which are located a fuel-supply device (fuel lines and burners), an ignition device (sparkplugs), and flameholders. The combustion chamber itself is located beyond the flameholders, and the rear flange of the chamber is connected to a variable-area nozzle.

Appropriate choice of afterburner element designs will ensure:

– stability of the fuel-combustion process over the entire range of flight altitudes and speeds;

– reliable starting of the afterburner at all altitudes;

– prevention of turbine-blade overheating when the afterburner is turned on and off, as well as during operation with the afterburner on;

– low hydraulic losses in the afterburner whether the device is on or off.

The stability of the combustion process, reliability of afterburner ignition, and the magnitude of hydraulic losses are determined to a considerable degree by the diffuser and flameholder designs, and by the atomization of the fuel supplied to the chamber. Here we must remember that an increase in the effectiveness of the elements that stabilize the combustion process will increase hydraulic losses and lead to a corresponding drop in engine thrust whether the afterburner is on or off.

The diffuser. The dimensions of the diffuser exit cross section are selected on the basis of the need to ensure the required drop in gas velocity at the combustion-chamber entrance. The velocity should not exceed 100-200 m/sec; higher velocities will make it difficult to ensure stability of the combustion process and sufficiently complete combustion, especially at high flight altitudes.

In order to reduce the weight and size of the afterburner, a short diffuser may be used, but for given entrance and exit areas, a reduction in diffuser length will increase the channel diffusion index, which is found in terms of the angle of a reference annular diffuser. As this angle increases, so will the hydraulic losses.

The basic elements of an afterburner diffuser are similar in design to the elements of a tailpipe. The diffuser consists of an outside pipe 1 (see Fig. 127), inside cone 2, and connecting elements. The inside cone is positioned and stresses transferred from the cone to the outside pipe with the aid of the stress-bearing rods 3, which are enclosed in the streamlined struts 4.

No blunt parts should be located in the afterburner-diffuser flow area (bolt heads, nuts, etc.), as they usually act as flameholders. A flame front is established at such elements, which helps to overheat and warp individual parts of the afterburner owing to the nonuniform distribution of temperatures at the walls.

Flameholders. Stable combustion of fuel in the jet can occur only at a low gas velocity not exceeding 15-25 m/sec. If the jet velocity were dropped to such a value by appropriate choice of duct diffusion index, a considerable increase in diffuser exit diameter would result. Thus combustion stability is provided by creating flow-reversal zones at the front of the chamber. This is done by installing one central flameholder and one or several ring flameholders in afterburners. Figure 128 shows stabilization of the combustion process with the aid of two ring flameholders and a central flameholder [27].

As the flow-reversal zone increases, the effectiveness of stabilization also rises. The dimensions of a flow-reversal zone depend on the midsection area of the flameholders. As this area increases, the flow-reversal zone will increase, and the stability of the combustion process will improve.

The relative locations of ring flameholders and the central flameholder, as well as the position of nozzles with respect to the flameholders will affect combustion stability, engine high-altitude performance, the nature of the temperature distribution over the chamber radius, and gas-pressure pulsations in the afterburner. The choice of flameholder dimensions and areas, their relative locations, and the positions of nozzles are determined empirically during development testing of the afterburner [17].

Fuel-supply device. Fuel is supplied to the afterburner with the aid of centrifugal-type injectors which are usually directed against the gas flow. The nozzles are located ahead of the flameholders. The point of fuel injection is so chosen that the process of vaporization of the atomized fuel is completed before it reaches the rear edge of the flameholder.

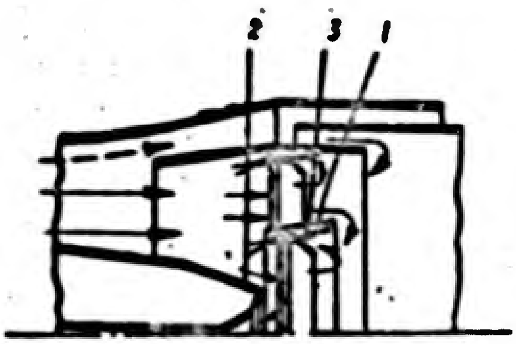

Fig. 128. Arrangement used to create flow-reversal zones by means of flameholders: 1) flameholder ring; 2) central flameholder; 3) nozzle.

The foreign literature recommends that the injector nozzles be placed at least 100 mm ahead of the flameholders. This nozzle arrangement is based on the assumption that the maximum dimension of

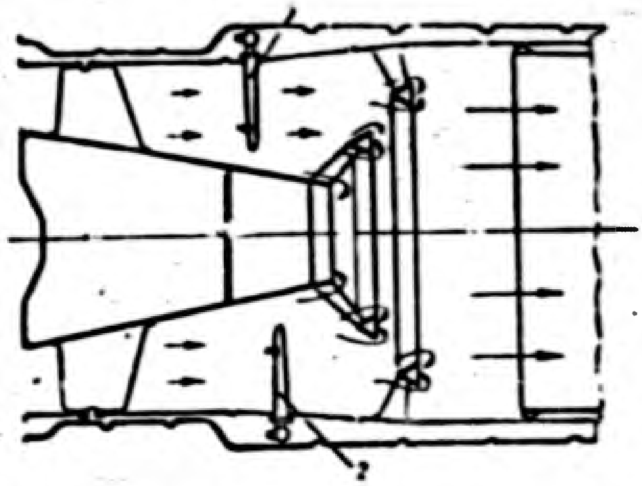

Fig. 129. Afterburner diffuser using two sets of nozzles (type ATAR engine): 1) front nozzle; 2) rear nozzles.

a fuel drop will not exceed 10 microns. It takes 0.0005 sec to vaporize such a drop. Then the process of vaporizing such drops in gas flowing at a velocity of 180 m/sec will be completed at a distance of 90 mm from the point at which the fuel is injected. Fuel that has not vaporized before reaching the flameholder strikes the walls of the ring flameholder, flows off in sheets from the rear edges of the ring, thus enriching the mixture beyond the flameholder. By increasing the distance from the point of fuel injection to the flameholder, we can obtain better drop vaporization and more uniform distribution of fuel over the entire combustion-chamber cross section, while at the same time, the mixture beyond the flameholder is made leaner. Thus, as the injectors are brought closer to the flameholder, the mixture at the center of the chamber will become richer.

In connection with the fact that the afterburner should operate stably both at high and low altitudes, as well as with a variable air-fuel ratio (especially where the degree of engine thrust augmentation varies), it is considered necessary to mount the injectors in several rows at varying distances from the flameholders (Fig. 129).

The injectors in one row are so arranged that the circles of maximum droplet-pattern cross section produced by neighboring injectors do not intersect but uniformly cover the cross section of the ring flameholder.

Combustion chambers (afterburners) are cylindrical or conical pipes made from heat-resistant sheet material. The geometric dimensions of the combustion chamber itself, beginning directly after the flameholder and ending where the nozzle is attached, are chosen with an allowance for the conditions that ensure the most complete combustion possible.

Ignition device. Ignition of the fuel when the afterburner is started on the ground at high gas temperatures (600-700°C) and high fuel pressures ahead of the injectors, which ensures good atomization, will occur with no need for additional starting equipment. Reliable ignition of the fuel in flight, especially at high altitudes, requires an additional heat source in the form of a starting flame. Such a pilot flame can be obtained by supplying fuel through several starting nozzles into the very turbulent flow behind the central flameholder. The mixture is ignited in this zone by the sparkplug 5 (see Fig. 127), located at the center of diaphragm 6.

Nozzles [18]. One of the fundamental problems in designing afterburners is the construction of a reliably operating nozzle with variable exit area. By varying the exit area of converging nozzles as well as the exit and throat areas of diverging nozzles, it is possible to hold the gas constants constant ahead of the afterburner when the afterburner is turned on and off.

Single-regime afterburners use two-position nozzles. When the afterburner is turned on, there is an increase in the nozzle flow area that corresponds to the degree of thrust augmentation used. Universal chambers (in which the degree of thrust augmentation is variable) are equipped with multiposition nozzles in which the exit area is subject to stepped or continuously variable control.

In principle, moving elements of the inside cone may be used to change afterburner-nozzle cross-sectional areas for independent regulation of exit and throat areas in convergent-divergent nozzles (see Fig. 126c). Here it is necessary to ensure reliable cooling of the inside-cone elements, which is difficult to do even with a low degree of thrust augmentation. Therefore, afterburner nozzles are most frequently designed with no central body, and make use of moving external elements alone (flaps).

5. THRUST-REVERSAL DEVICES [27]

Wheel brakes, drag flaps, and braking parachutes are widely used to decelerate aircraft during landings. To increase the efficiency of wheel brakes, we must increase the resistance to wear of the landing-gear wheel tires. An improvement in the landing performance of an aircraft by the use of flaps and parachutes involves structural complications and a substantial increase in aircraft weight. Thus, the use of TRD negative thrust, counter to the direction of flight, offers promising possibilities toward improvement of aircraft takeoff, landing, and maneuvering performance.

Engine negative thrust may be created by deflecting the jet of gases leaving the nozzle.

Devices designed to turn the gas jet are called thrust-reversal devices.

By employing thrust reversal during landing it is possible:

– to reduce landing-strip size and to permit aircraft to land at airports where the landing strip is completely covered with ice, as well as on aircraft-carrier decks;

– to reduce the time required for a landing approach owing to the possibility of increasing the glide angle;

– to increase the service lives of aircraft takeoff and landing equipment (brakes and wheel tires);

– to improve aircraft flight performance in maneuvers.

In design and development of thrust-reversal devices, the following specifications, usually applicable, should be kept in mind.

1. A negative thrust of at least 35-40% of the maximum thrust should be obtained under engine test-stand operating conditions.

2. It should be possible to change rapidly from negative thrust to maximum positive thrust, as is necessary in the case of an aborted landing of an aircraft.

3. When the thrust-reversal devices are brought into play, the temperature of the gases ahead of the turbine should not rise above the maximum permissible value.

4. The device used to turn the jet should add little weight.

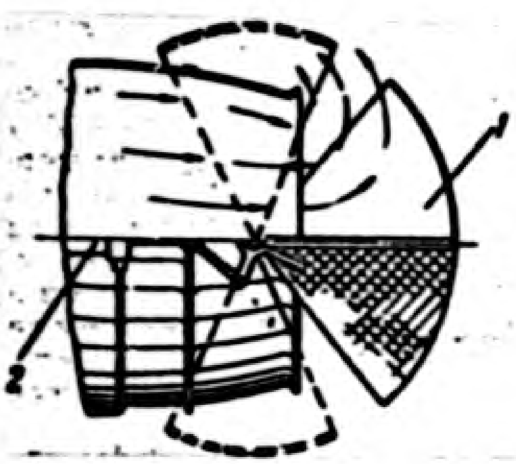

Fig. 130. Two-scoop thrust reverser: 1) scoop; 2) hydraulic cylinder.

5. Aircraft stability and response to controls should not be impaired when the thrust-reversal devices are brought into play.

6. Thrust-reversal facilities should be so designed and placed as to ensure that hot gases will not strike the skin of the fuselage, wings, and aircraft control surfaces.

Thrust-reversal devices may take quite different forms, both with respect to the principle used to deflect the stream, and with respect to the actual construction. The gas jet may be deflected before or after the nozzle. The gas jet may leave the deflecting elements either as two jets or as small jets uniformly distributed over the perimeter of the exhaust-system cross section.

Scoops that change the direction of motion of the gases are used to turn the jet into two large jets. Figure 130 shows a two-scoop device that deflects the gas jet after the nozzle. The scoops 1 are set in the position shown by the dashed lines when the thrust reverser is inoperative. In this position, the scoops have no effect on operatic of the exhaust system. The scoop positions are changed with the aid of the hydraulic cylinders 2.

Owing to the high gas exhaust velocity at the nozzle, the forces acting on the scoops and transmitted through the exhaust-system attachment fittings may reach considerable values. In addition, the design is complicated by the need to ensure reliable cooling of the scoops. As a result, equipment used for jet deflection following a nozzle is heavy. It is therefore better to deflect the gas jet ahead of the nozzle in order to reduce the loads on the scoops.

Designers have devoted a good deal of attention to thrust reversers using deflecting grids (Fig. 131a, b, and c), made in the form of stamped or shape-formed rings. The cross-sectional profiles of these rings are similar to the cross section used for turbine moving blades. In order to reduce drag when the reverser is inoperative, the grid may be retracted under the engine housing. When the thrust reverser is on, the grid is positioned after the nozzle. The gas stream leaving the nozzle is directed into the grid rings, which turn the jet.

The thrust reverser is placed in operation by directing the gas stream toward the grid. To do this, various devices acting on the stream may be used: air may be supplied to the center of the stream through vents 1 (Fig. 131a); the stream may be twisted with the aid of rotating blades 4 (Fig. 131c); and the positions of blades 2 (Fig. 131b) or special deflectors may be varied.

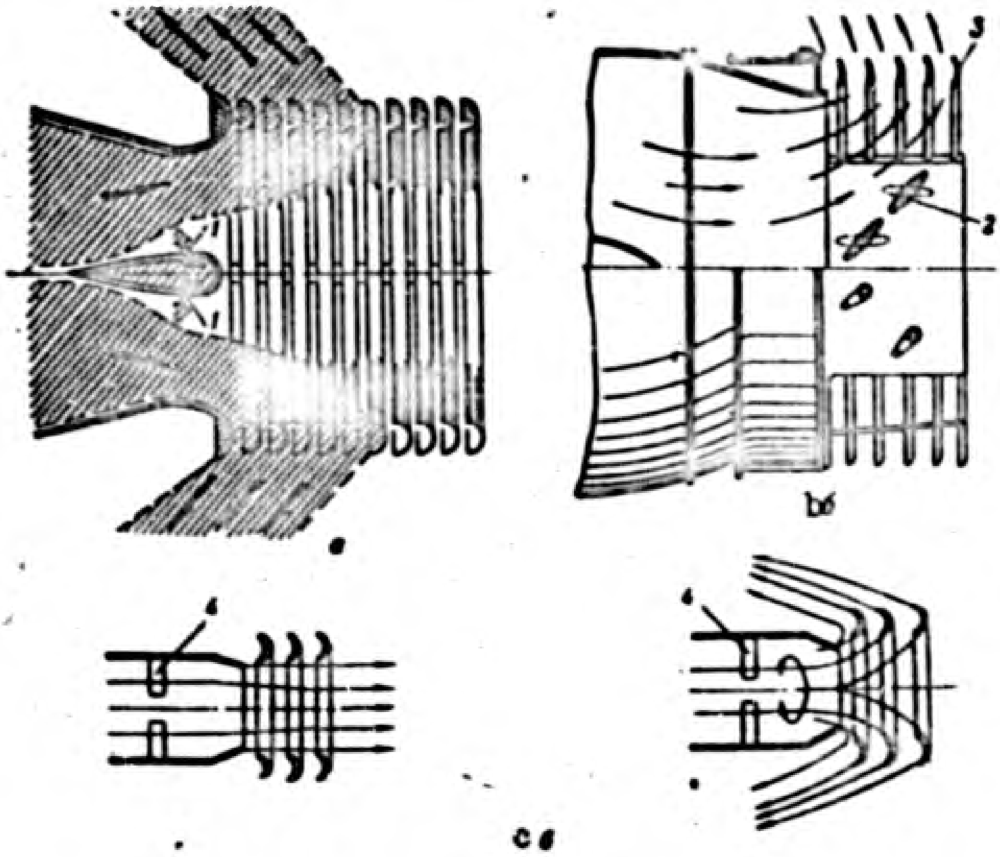

Fig. 131. Thrust reversers with deflecting grids: a) with air supplied to center through vents; b) with variable-position blades; c) stream twisted by rotatable blades (to the left, reverser inoperative; to the right, reverser operating); 1) vents; 2) blades; 3) guide grids; 4) rotatable blades.

A compressor supplies the air used to deflect the gas stream toward the grid. The amount of air needed to direct the gases into the grid is 2-3% of the total flow rate. In this case, the negative thrust may reach 50% of the maximum positive thrust.

Deflection of the jet owing to centrifugal forces when the gases are turned by rotating blades leads to a sharp change in thrust from the positive to the negative values. When this occurs, the turbine back pressure will also rise sharply. By controlling the nozzle exit area, it is possible to keep the turbine back pressure constant, but this requires exact coordination of changes in nozzle cross section and blade rotation.

6. THERMAL INSULATION AND COOLING OF EXHAUST SYSTEMS AND AFTERBURNERS

Gas temperatures in TRD exhaust systems reach 600-700°C. In present-day engines, exhaust system elements are usually made of heat-resistant alloys that operate reliably at wall temperatures of up to 600-650°C. In order to protect aircraft elements and engine control and supply elements against thermal radiation, thermal-insulating materials are installed between the outside pipe and the aircraft parts, or a system of shields is set up.

Where the material of the outside pipe possesses sufficient resistance to heat, measures are taken to reduce the heat lost to the surrounding medium. Asbestos cloth and aluminum foil are normally used as thermal-insulating materials. Here one layer of asbestos cloth 1 is placed at the outside pipe wall (Fig. 132), and up to five layers of

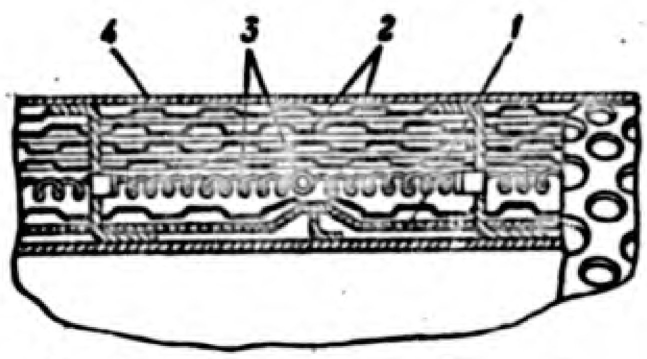

Fig. 132. Outside-pipe thermal insulation: 1) asbestos cloth; 2) aluminum foil; 3) spiral wires; 4) casing.

corrugated aluminum foil 2 are added. Between the asbestos cloth and aluminum foil there is a grid of spiral wires 3. Then the entire insulation pack is covered by a casing 4, which consists of two halves. The casings are attached to the pipe with the aid of band clamps. This way of attaching the casing ensures that it is free to move relative to the pipe under thermal deformation.

The thermal-insulating capability of multilayer aluminum foil is ensured by the presence of the closed air cavities, which have little thermal conductivity. In addition, the bright white surface of the foil acts as a heat reflector, which considerably reduces the amount of heat lost owing to thermal radiation.

The first domestic turbojet engines, models RD-10 and RD-20, had a fairly complicated exhaust-section cooling system that made use of air bled from the fourth compressor stage in the RD-10 and from the space under the engine hood in the RD-20. It was necessary to use such a cooling system to ensure failure-free operation of the control system for the inside-cone moving parts, and also owing to the fact that materials of inadequate heat resistance were used for the nozzle elements. In these engines, the exhaust-system parts were made from aluminum-coated type 10 steel, and the hottest parts from YaIT steel.

In present-day TRD, exhaust devices are made from the same materials as afterburners, so that the permissible temperature of parts should not exceed 600-650°C. The mean gas temperature within a chamber with the afterburner operating reaches 1100-1200°C with 20-25% thrust augmentation under engine test-stand operating conditions. A further increase in the degree of thrust augmentation will increase gas temperature still further. Thus afterburner operating reliability depends considerably on the efficiency of combustion-chamber cooling.

Combustion-chamber wall temperatures may be reduced in comparison with the gas-stream temperature by introducing boundary-layer (film) cooling, which essentially consists in creating a flowing layer of cold air or low-temperature gas (600-650°C), supplied through slots or vents within the chamber; external cooling may also be used, with the outside pipe surface cooled by a blast of cooling air which provides convective cooling of the walls.

Cooling air may be taken from a compressor or the space under the engine hood. Atmospheric air is most frequently used for this purpose. Here the cooling system is characterized by its simplicity while the absence of additional ducting or wiring leads to a reduction in the weight of the entire engine installation.

Atmospheric air is circulated either by drawing it from the space between the chamber wall and the casing, using the entraining effect of the gas jet leaving the nozzle, or by applying pressure to this space through the dynamic pressure, or by a combination of both these methods.

In order to increase the period over which the engine can be operated continuously with the afterburner on, and to increase the degree of TRD thrust augmentation, it is necessary to use the same methods for cooling and protecting chamber parts as are used in the cooling systems for main combustion chambers.

7. EVALUATING THE STRENGTHS OF EXHAUST-SYSTEM AND AFTERBURNER ELEMENTS

Outside pipes and nozzles for TRD exhaust systems are thin-walled cylindrical or conical shells subject to stresses in compression or tension produced by pressure drops as well as by torsion and bending.

The loads acting may be:

– radial and axial forces produced by pressure drops at the shell walls;

– twisting moments transmitted to the outside pipe by members that straighten the gas stream after the turbine;

– bending moments produced by exhaust-system gravitational forces and by inertial forces.

The largest stresses in shell walls and flange-connection elements are produced by pressure drops. The stresses appearing owing to twisting and bending moments are small by comparison.

The pressure within the pipe depends on engine operating conditions and the flight regime. As the speed of the engine rotor increases and flight speed goes up, the pressure within the exhaust system rises continuously. The air pressure in the engine nacelle is determined by the flight speed and the pressure of the external atmosphere.

The most dangerous conditions for exhaust systems and afterburners are flight near the ground at maximum speed and minimum ambient temperature, and sharp throttling down from maximum flight speed at low altitudes.

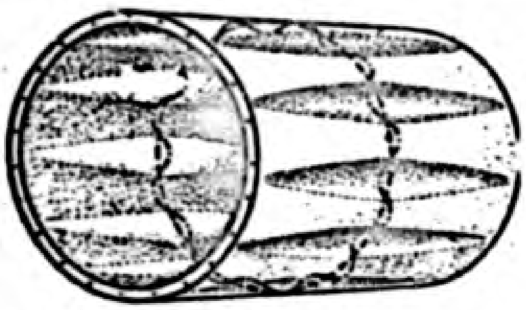

Fig. 133. Diagram showing formation of waves when a shell loses stability.

While the first regime is characterized by maximum shell bursting stresses owing to excess internal pressure, the second regime is characterized by maximum compression due to excess external pressure.

The axial forces loading the shell and flange-connection elements appear as a result of pressure drops at conical sections of the exhaust section, and are also transmitted from struts and other elements connecting the inside cone to the outside pipe.

The axial force acting on an exhaust section as a whole are directed to the rear under steady conditions, and it will equal the sum of the forces loading the outside pipe, inside cone, nozzle, and struts.

Shells can fail not only under internal pressure, but also as a result of the action of an external excess pressure. In the latter case, shell failure takes the form of a loss of stability appearing in the formation of deep hollows (buckling) along the shell length and perimeter.

In shells welded to rigid machined flanges that remain circular after loss of stability, a single half wave is formed along generatrices (Fig. 133), and several waves around the circumference. The shorter the shell, the larger the number of circumferential waves.

The pressure at which loss of stability occurs is called the critical pressure.

The value of the critical pressure depends on the initial buckling. The presence of depressions directed toward the center of curvature leads to a reduction in the pressure at which the shell loses stability. Axial forces compressing the shell also reduce the critical pressure.

In order to prevent loss of stability, the critical pressure should be higher than the maximum excess pressure acting.

During operation and in installation of outside pipes of exhaust systems and afterburners, it is necessary to check the condition of the outside pipe to see that there are no depressions to reduce the critical pressure.

Part Four

STRESS-BEARING SYSTEMS AND TRANSMISSION ELEMENTS OF GAS-TURBINE ENGINES

Chapter 9

STRESS-BEARING SYSTEMS OF GAS-TURBINE ENGINES

1. GENERAL INFORMATION.

The stress-bearing system of a gas-turbine engine [GTD] absorbs the loads acting on the structural elements and transmits the resultant load to the points at which it is attached to the aircraft.

The stress-bearing system of a gas-turbine [GTD] engine consists of the stress-bearing of the rotor (that part of a gas-turbine engine [GTD] that turns in bearings) and the stress-bearing system of the housing (the motionless part).

The stress-bearing system of a gas turbine engine [GTD] rotor is made up of parts of the compressor and turbine rotors and their points of attachment, and also of parts of the reduction gear in a turboprop [TVD] engine.

The housings of the compressor, combustion chamber, turbine, inlet and outlet sections, and also the housings of the bearings and reduction gear (for the turboprop engine) form the stress-bearing system of the engine housing.

The stress-bearing system of the rotor is connected through bearings with the stress-bearing system of the housing, comprising the stress-bearing system of the gas-turbine engine [GTD].

The parts of the stress-bearing housing of a gas-turbine [GTD] engine (on which the attachment points to the aircraft are situated) are the basic stress-bearing members. All the rest of the members of the stress-bearing system are joined to them.

The design of the parts of the stress-bearing system is subject to the following requirements.

1. The rigidity of the stress-bearing parts of the system should insure the constancy of their shape under load.

2. To insure miminum weight, the shapes of stress-bearing parts and their connections should be such that during operation of the gas-turbine [GTD] engine they should experience the lease possible loading.

3. To decrease thermal stresses and buckling, the design of stress-bearing housings and parts connections that are heated to different temperatures during operation should insure the possibility of free expansion of the parts when they are heated.

4. For reduction of gas-pressure losses, the stress-bearing parts situated in the gas flow should have a smooth surface and well streamlined shape.

In general, gas forces, gravitation, inertial forces, and also the forces arising as the result of nonuniform heating of attached parts act on the elements of the stress-bearing system of a gas-turbine [GTD] engine.

Figures 134 and 135 show examples of the loading of the elements of the stress-bearing system of a turbojet [TRD] engine with centrifugal and axial compressors by axial gas forces and torques. Gas-flow forces acting on the parts of a gas-turbine [GTD] engine may be presented in the form of three components, i.e., radial, axial, and circumferential.

The radial components of the gas-flow forces are balanced inside the engine, loading the component parts by internal stresses (they are not shown in the diagram). The forces arising as a result of nonuniform heating are also in equilibrium. The axial forces, however, are only partly compensated. The resultant axial force is equal to the thrust (taken in Figs. 134 and 135 as 100%) and it is transmitted through the attachment points of the engine to the aircraft.

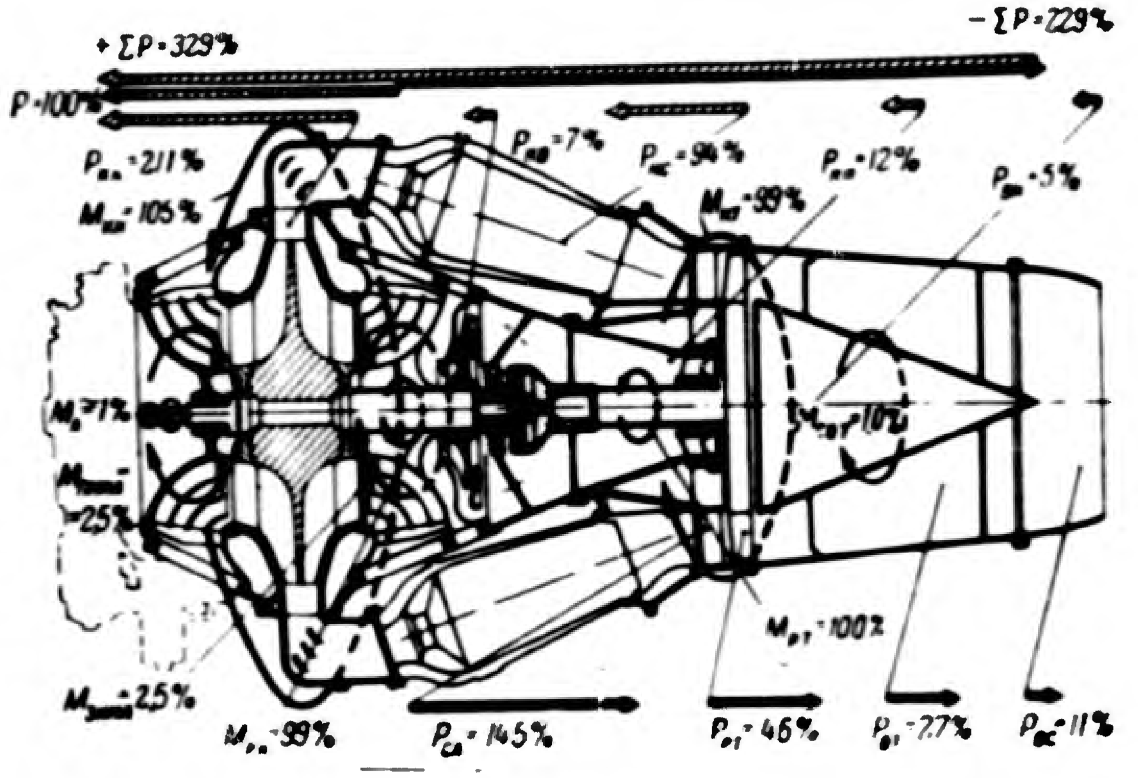

Fig. 134. The balance of axial forces and torques acting on the elements of the stress-bearing system of a turbojet [TRD] engine with a centrifugal compressor. (Based on percentage of the thrust force and torque on the turbine rotor for an engine with a centrifugal compressor, in the static operational regime with maximum rpm). P_{KK}) Axial force on compressor housing; P_{KV}) axial force on fan housing; P_{KS}) axial force on combustion chamber; P_{KP}) axial force on bearing housing; P_{VK}) axial force on inner cone; P_{SA}) axial force on nozzle diaphragm; P_{RT}) axial force on turbine rotor; P_{VT}) axial force on outlet tube; P_{VS}) axial force on outlet nozzle; M_A) torque on assembly drive; M_K) torque on compressor rotor; M_{RT}) torque on turbine rotor; M_{PNNA}) torque on forward fixed guide-vane assembly; M_{KK}) torque on compressor housing; M_{ZNNA}) torque on read fixed guide-vane assembly; M_{KT}) torque on turbine housing; M_{SVT}) torque on the outlet tube struts.

The circumferential components of the gas-flow forces, acting on the rotor parts, generate torques that turn the rotor, and the circumferential components acting on the parts of the gas-turbine [GTD] engine housing are the torques twisting the stress-bearing frame. The directions of the moments exerted by gas forces are shown in Figs. 134 and 135 by arrows. With axial inlet of air into and axial ejection of gases out of the engine, the torques acting on the rotor and compressor housing are balanced by the respective torques imposed on the rotor and housing of the turbine and struts of the exhaust system and, consequently, those loads are not transmitted to the points at which the engine is attached to the aircraft.

The torque arising from the action of gas forces is transmitted to the points of engine attachment to the aircraft only in the case in which the air is rotating at the inlet to the engine or at the gas exhaust from the nozzle. Moreover, the following forces are transmitted to the engine attachment points:

the resultant of the centrifugal inertial forces and their moments;

transient inertial forces due to the mass of the component parts of the entire engine as a result of aircraft maneuvers, and also gyroscopic moments due to rotation of the rotor and propellers;

loads varying in magnitude and direction due to vibrations of the gas flow inside the engine and vibrations of the engine parts;

resultant torque due to transient regimes of engine operation;

the weights of the structural elements of the engine.

In turboprop [TVD] engines we also encounter the additional forces of thrust and the reaction moment of the propeller (in the case of a single propeller). In the case of two coaxial propellers turning in opposite directions, a reaction moment is present and is equal to the

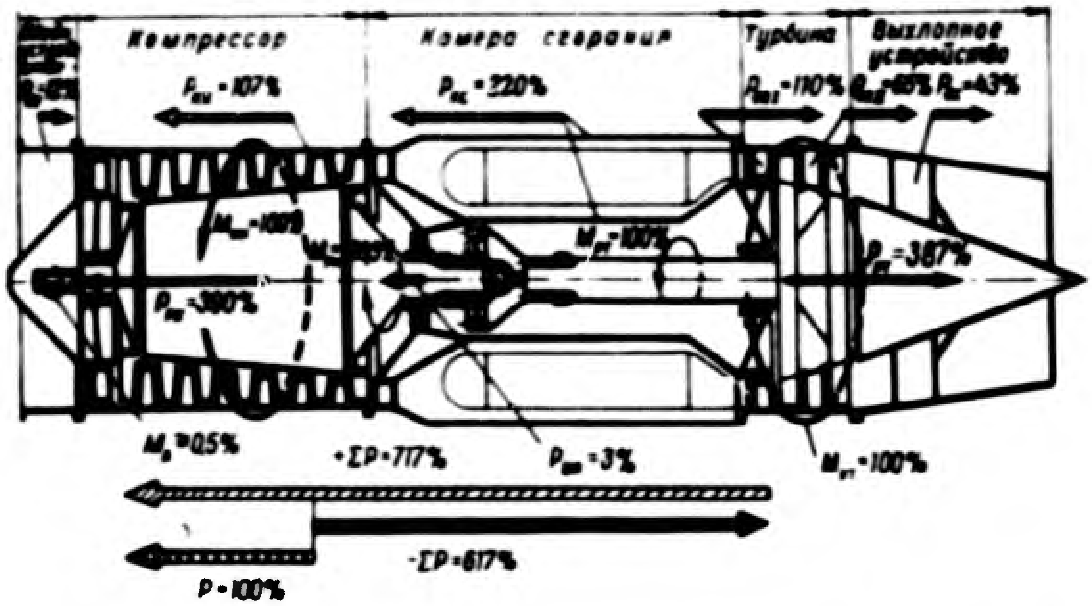

Fig. 135. The balance of axial forces and torques acting on the elements of the stress-bearing system of a turbojet [TRD] engine with axial compressor (ratios are given for the same regime as in Fig. 134). P_{VU}) axial force on inlet assembly housing; P_{KK}) axial force on compressor housing; P_{KS}) axial force on combustion chamber; P_{SAI}) axial force on nozzle diaphragm of first turbine stage; P_{SAII}) axial force on nozzle diaphragm of second turbine stage; P_{VS}) axial force on elements of the exhaust system; P_{RT}) axial force on turbine rotor; P_{UP}) axial force on thrust bearing; P_{RK}) axial force on the compressor rotor; M_{KK}) torque on compressor housing; M_{KT}) torque on turbine housing; M_{A}) torque on drive of the auxiliary assemblies; M_{RK}) torque on compressor rotor; M_{RT}) torque on turbine rotor.

difference of the moments on the forward and rear propellers.

2. STRESS-BEARING SYSTEMS OF ROTORS

The designs of gas-turbine [GTD] engine rotors are determined by the type of compressor, turbine, their points of attachment, and the number and location of bearings.

In turbojet [TRD] engines with a single-cascade compressor and in a turboprop [TVD] engine with compressor and propeller driven by a common turbine, a single-shaft stress-bearing system is used for the rotor.

In turbojets [TRD] with two-cascade, birotational compressors and in turboprop [TVD] engines with individual drives for the compressor and propeller, a two-shaft rotor stress-bearing system is used.

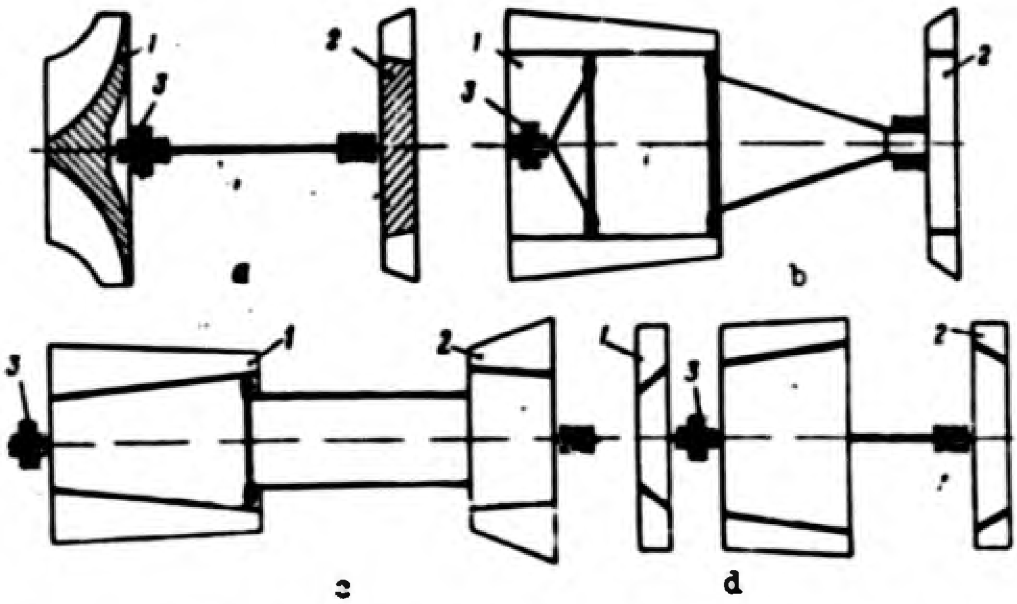

Fig. 136. Diagrams of stress-bearing systems of single-shaft two-support rotors. a) Cantilever arrangement of compressor and turbine; b) with compressor situated between bearings and the turbine cantilevered; c) with compressor and turbine situated between bearings; d) with cantilevered arrangement of the turbine and supersonic stages of the compressor; 1) compressor; 2) turbine; 3) support-thrust bearing.

In number of supports, rotors may have two, three, four, or many supports. The more rigid a rotor for given length, the smaller the number of supports for an allowed amount of bending.

Depending on the number of support-thrust bearings, single-shaft rotors may have one or two bearings, and this is a function of the type of connection between the compressor shaft and the turbine shaft.

The connections of compressor and turbine shafts may be rigid or flexible. The relative motion of the shafts is prevented with a rigid connection; with a flexible connection, the possibility of bending displacements of the shafts is provided. In this type of connection the shafts may transmit only torques; or torques and axial forces; or torques, radial, and axial forces.

Let us consider possible arrangements of stress-bearing systems for gas-turbine [GTD] engines.

Single shaft stress-bearing systems of two-support rotors may be made in one of the arrangements shown in Fig. 136.

In arrangement a, compressor 1 and turbine 2 are cantilevered on the shaft.

In arrangement b, compressor 1 is located between supports, and turbine 2 is cantilevered.

In arrangement c, compressor 1 and turbine 2 are located between supports.

In arrangement d, the supersonic stage of compressor 1 and turbine 2 are cantilevered on the shaft.

The arrangements of rotor stress-bearing systems shown here have only a rigid connection of the shafts in engines, both with centrifugal and with axial compressors.

Axial attachment of the two-support gas-turbine [GTD] engine rotor is provided by one of the bearings, called the support-thrust bearing, which absorbs the total axial and part of the radial load acting on the engine rotor.

When the axial loads acting on the compressor and turbine rotors are opposite in direction, support-thrust bearing 3 transmits only their difference, which makes possible a reduction in the size and weight of the bearing, and a simplification in the design of the support.

In a two-support rotor, the support-thrust bearing, being relatively more heavily loaded, is located near the compressor, i.e., in the region of lower temperatures.

The arrangement shown in Fig. 136, a, provides for ease in assembling and dismantling the engine, but it is used only in the case of short compressor and turbine length since in any other circumstance the cantilevered support results in unacceptable large bending deformations of the shafts at the bearing points due to the weight moments. Further, with such an arrangement, because of the small distance between the bearings, their loads are increased due to the gyroscopic moment.

With large compressor lengths, in order to reduce the bending of the gas-turbine [GTD] rotor, the shaft diameter is increased, making part of the compressor stages cantilevered, and the arrangement b (Fig. 136) is used.

In the case of a multistage turbine and a shaft with a large diameter, the arrangement c (Fig. 136) is used, i.e., the stress-bearing system in which the turbine is situated forward of the rear bearing. Increase in the distance between supports in this case leads to decrease in the load on the bearings due to the gyroscopic moment of the rotor.

A supersonic stage in an axial compressor is sometimes arranged to have a cantilever support, forward of the front bearing (Fig. 136, d), and this is convenient for gasdynamic reasons, because with such an arrangement the path of the air entering the compressor is not encumbered with the housing supports.

Bending deformations of the component parts of the stress-bearing system of a gas-turbine [GTD] engine, and also nonaxiality of the bearing sockets lead to noncongruency of the shaft journals and the bearing housings. Noncongruency of the axes may cause binding of the bearings and asymmetry of their loading, as a result of which the bearings quickly fail. To avoid this condition, in the design and construction of gas-turbine [GTD] engines adequate rigidity of the housing and shafts is provided; also the coaxiality of the bearing sockets is given very considerable attention. In a number of cases, self-adjusting bearings are used.

Stress-bearing systems of single-shaft two-support rotors are ordinarily used on auxiliary-service engines [27]. The over-all length of the rotor in such engines can be reduced because of the lower compression ratio of the compressor and, consequently, the small number of compressor stages.

Single-shaft stress-bearing systems with three-support rotors, which are most widely used in gas-turbine engines, have a rigid connection between the shafts of compressor and turbine (Fig. 137, a) or a flexible one (Fig. 137, b, c and d), achieved by the use of a splined coupling.

With a rigid connection of the compressor and turbine shafts, the stress-bearing system of the engine is statically indeterminate, and this requires a corresponding rigidity of the stress-bearing frame, and exacting coaxiality of the bearings in order to prevent overloading them.

A diagram of a stress-bearing system of a three-support rotor with flexible connection of the compressor and turbine shafts, transmitting torque and radial loads, is shown in Fig. 137, b. The torque and radial loading of the turbine are transmitted to the compressor by use of a coupling spline 2, permitting bending of the shafts by reason of the clearance in the grooves of the coupling. The clutch has a large diameter in order to insure the strength of the short grooves in the transmission of torque.

Attachment of the compressor and turbine rotors in the axial direction is individually accomplished, by use of support-thrust bearings 1.

In separate axial attachment of the compressor and turbine, the splined engagement should permit free axial elongation of the shafts when they are heated.

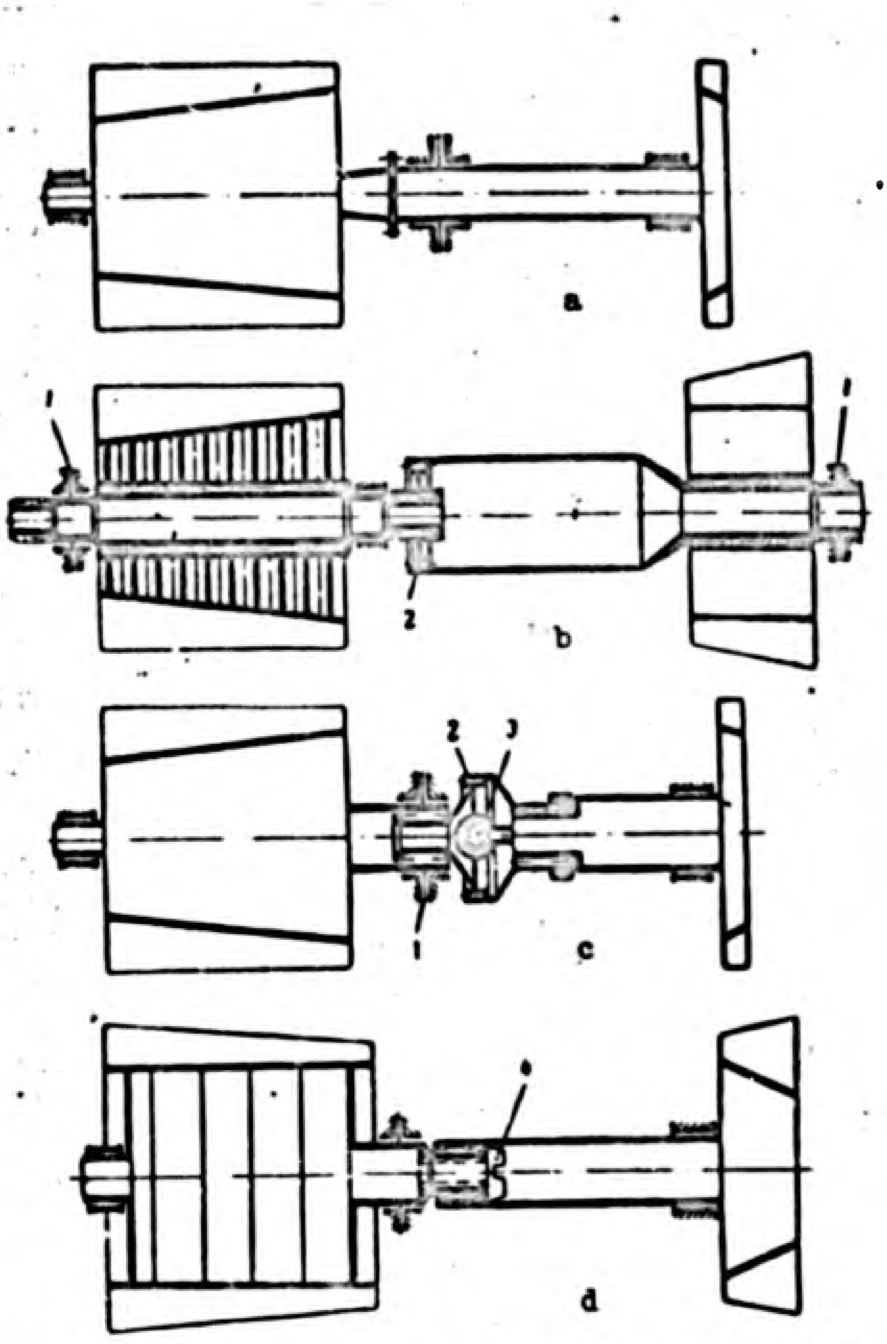

Fig. 137. Diagrams of stress-bearing systems of single-shaft three-support rotors. a) With rigid connection of the compressor and turbine shafts; b) with flexible connection of compressor and turbine shafts, transmitting torque and radial loads; c and d) with flexible connection of the compressor and turbine shafts, transmitting torque, axial, and radial loads; 1) support-thrust bearings; 2) splined coupling; 3) ball joint; 4) clamping screw.

Figure 137, c shows a diagram of the stress-bearing system of a three-support rotor with flexible connection of the shafts of compressor and turbine, transmitting torque, axial, and radial loading. The turbine torque is transmitted by a splined coupling 2. The axial link between the compressor and turbine is accomplished by use of spherical joint 3, being at the same time the second support of the turbine shaft. Only the difference of the axial forces acting on the compressor and turbine acts on support-thrues bearing 1.

From the structural standpoint, the simplest connection is that shown in Fig. 137, d. The axial load from the turbine is transmitted through the head of clamping screw 4, screwed into the compressor shaft. The radial load and torque of the turbine are transmitted by use of grooves.

The three-support gas-turbine [GTD] engine rotor having an axial link between compressor and turbine is held in the axial position by a single support-thrust bearing. In order to prevent significant change in the magnitude of the axial and radial clearances in the flow passage of the turbocompressor with changing operational regimes of the gas-turbine [GTD] engine (in heating and cooling), the support-thrust bearing is ordinarily located between the compressor and turbine.

Single shaft stress-bearing systems of four-support rotors mostly have flexible shaft connections. In this case either the compressor is situated between supports and the turbine is cantilevered, or the compressor and turbine are located between supports. The latter arrangement is used only in case the turbine has a large number of stages.

Figure 138, a shows a diagram of the stress-bearing system of a four-support rotor with cantilevered arrangement of turbine and flexible connection of the shafts, executed in the form of two splined couplings 3. The connection transmits torque and permits not only bending, but also parallel displacement of the connected shafts, which might be caused by noncoaxial bearings. In principle, such a connection might be executed in the form of a splined coupling either of large diameter with short grooves (Fig. 138, a) or of small diameter with long grooves (Fig. 138, b).

In the diagram shown in Fig. 138, a, the compressor and turbine rotors exhibit individual axial attachment in bearings 1 and 2 respectively. In such a design version of the engine rotors, the compressor and turbine rotors may be independently balanced, and they may be independently assembled in the stress-bearing frames.

The presence of four bearings, two of which are support-thrust bearings, increases the weight of the rotor and stress-bearing frames. The absence of an axial link between the rotors of the compressor and turbine leads to the necessity of axial individual relief of the support-thrust bearings.

A better version is the stress-bearing system of a four-support rotor, a diagram of which is shown in Fig. 138, b. The shafts of the compressor and turbine have flexible connections transmitting only torque. The axial bond between the compressor and turbine is achieved either by rod 4, or by a double spherical joint 6, permitting restriction of the number of support-thrust bearings 1 to one. Torque from the turbine to the compressor is transmitted through an intermediate long sleeve 5, permitting the use of long splines on a shaft of small diameter, because in this case, for the same bending of the shafts, the bending in the grooves is reduced.

Double shaft stress-bearing systems of gas-turbine [GTD] engine rotors are made up of the stress-bearing systems of single-shaft rotors considered above.

Figure 139, shows a diagram of the stress-bearing system in a four-support turboprop [TVD] engine with separate drives for compressor and propeller. The turbocompressor rotor has a splined coupling 3 with a tight fit. The last stage of the gas turbine 4 drives the pro-

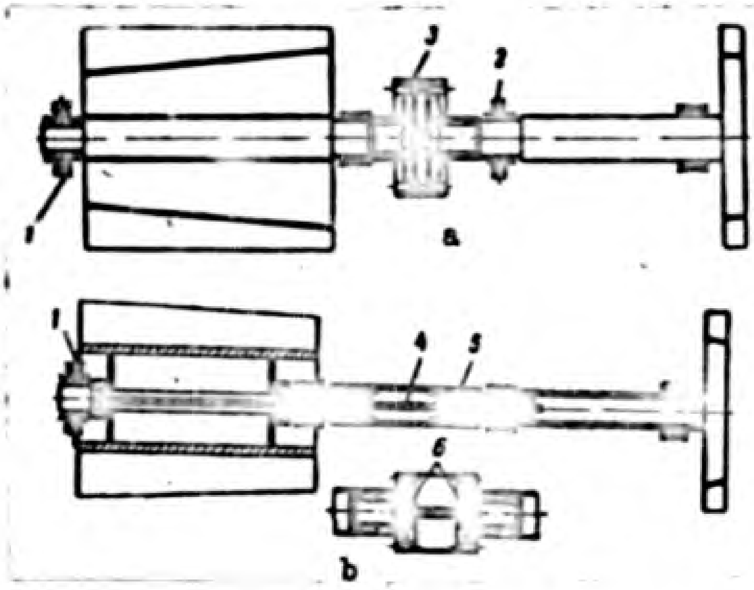

Fig. 138. Diagram of stress-bearing systems of single-shaft four-support rotors. a) connection of shafts by use of splined coupling of large diameter with short grooves; b) connection of shafts by use of a splined coupling of small diameter with long grooves; 1 and 2) support-thrust bearings; 3) splined coupling; 4) rod; 5) intermediate sleeve; 6) double spherical joint.

peller through a long shaft 2, a driving splined coupling 1, and a reduction gear. Such an arrangement for the stress-bearing system of a two-shaft rotor may be applied in gas-turbine [GTD] engines having a short turbocompressor rotor.

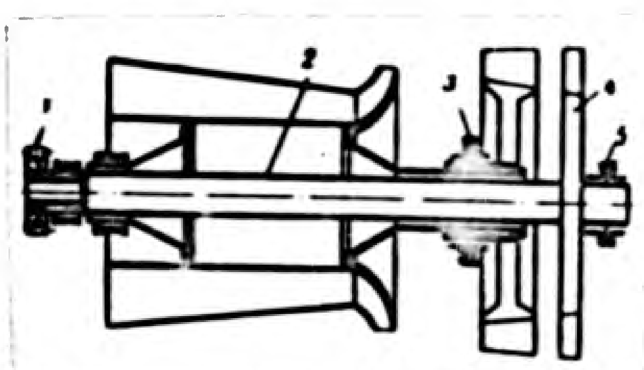

Fig. 139. Diagram of the stress-bearing system of a two-shaft four-support rotor. 1) Splined coupling; 2) shaft; 3) splined engagement; 4) propeller turbine; 5) support-thrust bearing.

Figure 140 shows an arrangement of a stress-bearing system of a two-shaft turboprop [TVD] engine rotor with five supports. The high-pressure rotor has two supports, with location of the support-thrust bearing 3 ahead of the turbine.

The low-pressure rotor has three supports, with separate axial attachment of compressor and turbine. For axial relief of bearings 1 and 4, the coupling 2 has spiral splines. For transmission of torque from the turbine to the compressor, the spiral splines when engaged generate axial forces just as occurs in case of attachment by a bolt and nut. The direction of the spiral splines and their pitch are chosen so that most of the axial forces arising in the turbine and compressor are compensated by the axial forces in the engagement of the spiral splines.

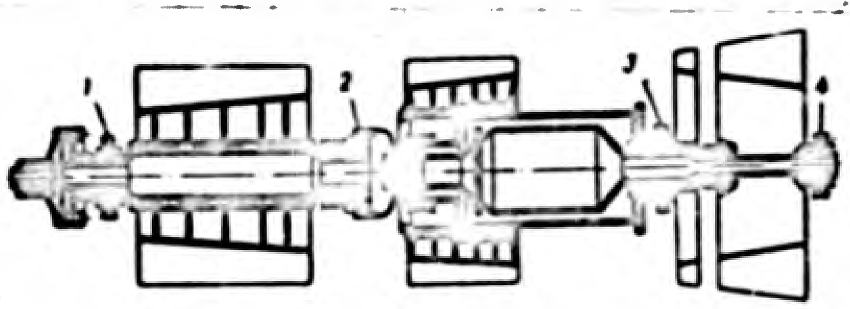

Fig. 140. Diagram of the stress-bearing system of a two-shaft five-support turboprop [TVD] engine rotor (the British "Orion" engine of the Bristol Company). 1, 3 and 4) Support-thrust bearings; 2) splined coupling.

The parts of the stress-bearing system of the gas-turbine [GTD] engine rotor (Fig. 141), operating in a steady regime, are loaded:

by torques due to the turbine M_t, the compressor M_k, and the accessory drive M_a (in the case of the turboprop [TVD] engine, by the torque of the propeller drive), causing torsional deformation;

by gravity forces G_t and G_k on the rotor parts;

by centrifugal forces P_j due to the inertia of the mass of the rotating members;

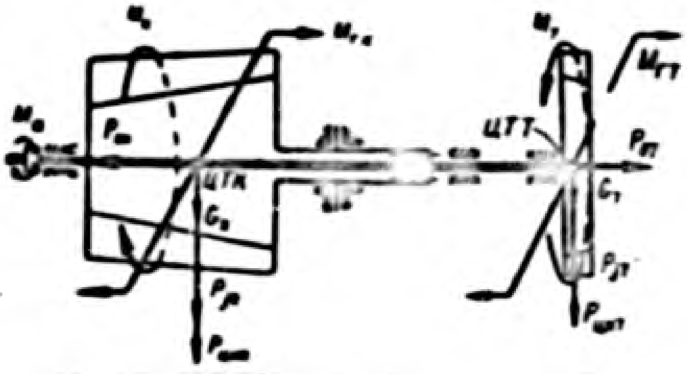

Fig. 141. Loads acting on the elements of the rotor stress-bearing system.

by gyroscopic moments M_g, due to aircraft maneuvering;

by centrifugal forces P_{tsnt} and P_{tsnk} of the unbalanced mass of the rotor, due to bending deformation;

by axial loads P_{ot} and P_{ok} due to the blades, and by loads due to gas pressure acting on the compressor and turbine disks (in the case of turboprop [TVD] engines, due to the thrust of the propeller), resulting in tensile deformation.

The largest bending deformations in the rotor parts arise when the aircraft is pulling out of a dive with the engine operating at maximum power, and when the aircraft is falling in a flat spin. In the former case, the bending moments caused by gravity forces, the unbalanced centrifugal forces of inertia, and centrifugal forces of the inertia of the rotor mass act in one plane (the vertical plane) and are additive. In the second case, the bending moments arising from the action of the force of gravity, the unbalanced centrifugal force of the inertia of the rotor, and the gyroscopic moment, maximum in magnitude, act in the same vertical plane and also are additive.

The gyroscopic moment of a gas-turbine [GTD] engine rotor depends on its weight, the geometric dimensions, and the angular rotational speeds of the rotor and aircraft. It has a maximum value when the axis of rotation of the aircraft is perpendicular to the rotational axis of the rotor.

The direction of the action of the gyroscopic moment, indicated by the arrows M_g, are illustrated in Fig. 142, in which two mutually perpendicular planes, the vertical x - z and horizontal x - y axis are drawn through the x - x axis of rotation of the rotor. The direction of rotor rotation is indicated by the arrow w, and the direction of the aircraft rotation, by the arrow Ω. The planes of aircraft rotation are cross-hatched by vertical and horizontal lines, respectively. The gyroscopic moment always acts in the plane passing through the axes of rotation of the rotor and aircraft, and has a direction such that with a rotor turning toward the left, when the aircraft is turning so as to dive (Fig. 142, a), the gyroscopic moment turns the aircraft to the right, and when the rotor is rotating to the right the gyroscopic moment (Fig. 142, b), turns it to the left. When the aircraft is turned to the left, with a left-turning rotor (Fig. 142, c) the gyroscopic moment tends to cause the aircraft to dive; and with a right-turning rotor (Fig. 142, d), it tends to cause it to pitch.

The unbalanced centrifugal forces of the rotating rotor mass, and the moments caused by these forces are variable in direction, and in addition to bending the rotor, also cause a load on the supports and points of attachment of the engine to the aircraft.

Along with the loads arising from vibration of the gas flow inside the engine and the vibration of its parts, the unbalanced centrifugal forces of the rotor also characterize the balance of the engine as a whole.

The engine is said to be balanced if the loads on the attachment points to the aircraft turn out to be constant in magnitude and direction. An unbalanced engine may cause undesirable vibrations of the power plant and aircraft.

In engine balancing, the loads on the attachment points (supports) are determined and means are sought for insuring, as much as possible, a uniform loading of those points (supports).

Static and dynamic unbalance of the rotor are differentiated. Static unbalance appears if the center of the rotor mass does not lie on the rotational axis, but is displaced at some distance from it. In that case, when the rotor turns, an unbalanced centrifugal force of inertia is developed, turning with the rotor, and loading its supports.

Dynamic unbalance appears in cases where an unbalanced moment is developed under the action of centrifugal forces of inertia.

The degree of static unbalance of the rotor is characterized by the so-called disbalance, by which is understood the product of the weight of a balancing weight G_b and the quantity r, describing the distance between the center of mass of the balancing weight and the rotational axis of the rotor. The quantity G_b is represented by a balancing weight 1, such that placing it at a distance r from its center of mass to the rotational axis of the rotor (Fig. 143) will achieve complete static balance.

For rotors of turbine and compressors of modern gas-turbine [GTD] engines, the maximum allowable magnitude of disbalance is small and amounts to 10 - 50 g-cm in all. However, because of the high rotational speeds of rotors, the unbalanced force in this case may amount to 30 - 80 kg.

The maximum magnitude of load on the rotor support under the action of the unbalanced moment is about the same as the force of unbalance.

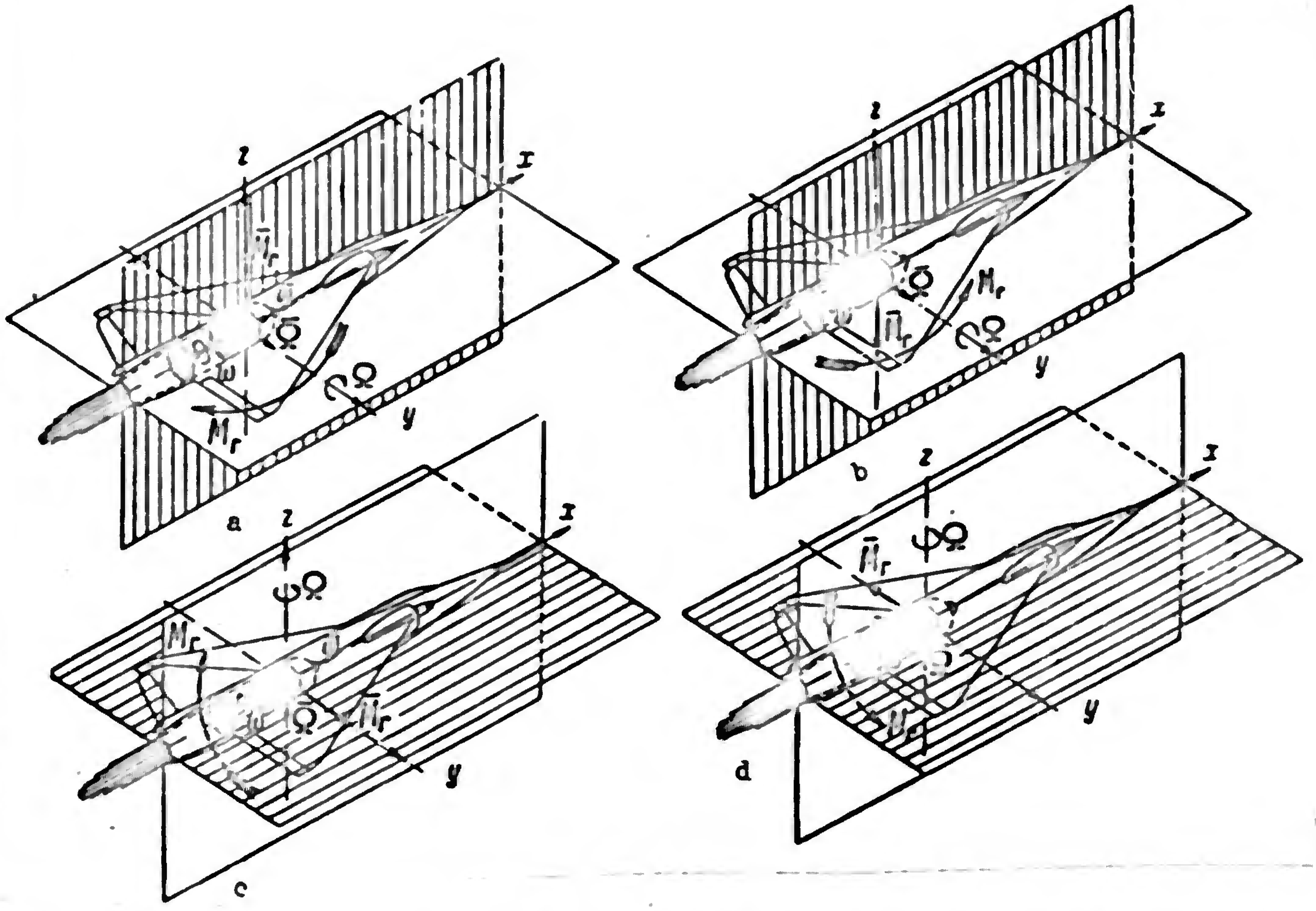

Fig. 142. For determination of the direction of action of the gyroscopic moment of a rotor. a) Aircraft going into a dive with left-turning rotor; b) aircraft going into a dive with right-turning rotor; c) aircraft turning left with left-turning rotor; d) aircraft turning left with right-turning rotor.

Under actual conditions static and dynamic unbalance are always simultaneously present.

Balancing of high-speed rotors of compressors, turbines, reduction-gear wheels in turboprop [TVD] engines is carried out on special stands for dynamic balancing, which make it possible to determine not the magnitude of the forces and moments of unbalance, but also the location of the plane in which they act.

In the connection of the shafts of a compressor and turbine by use of connecting couplings, at the time of engine assembly, the planes of action of the unbalance moments of the coupled rotors are arranged

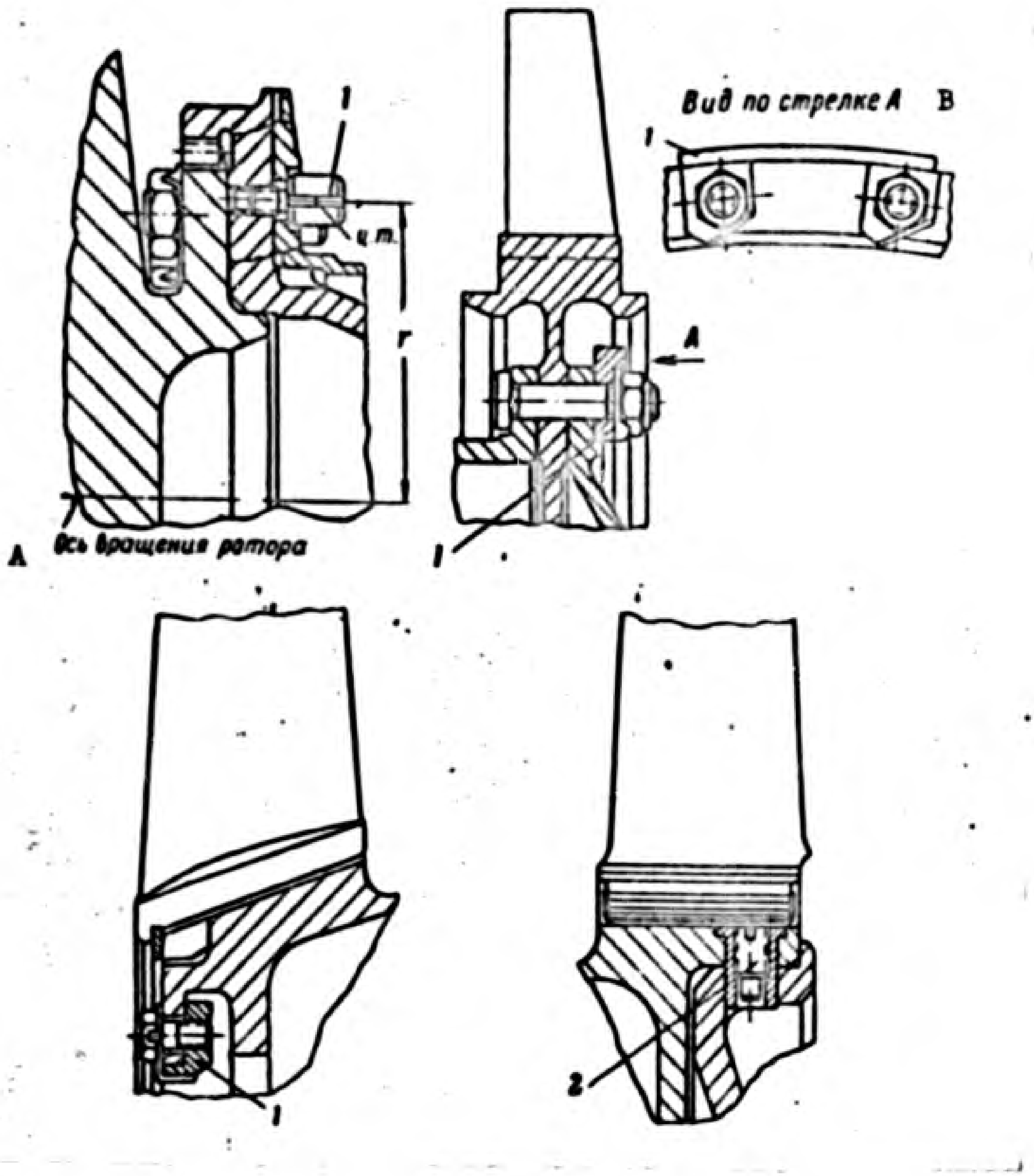

Fig. 143. Location of balancing weights on rotor parts. 1) Balancing weight G_b; 2) balancing plug; A) axis of rotor rotation; B) view along the arrow A.

so as to insure the minimum possible total moment of unbalance.

The precision of balance is insured by removal of unnessary material on the parts in especially designated places, rearrangement of compressor or turbine blades having different weights, and also installation of balancing weights 1 or plugs 2 (see Fig. 143) in planes located near the rotor supports. The precision of the balancing is evaluated by the magnitude of the <u>coefficient of vibrational over-</u>

load in the maximum operational engine regime. The coefficient of vibrational overload is defines as the ratio between the inertial force due to engine vibration and the gravitational force, or the ratio of acceleration due to vibration to the acceleration of gravity.

In the application of gas-turbine [GTD] engines with high-speed rotors, it is noted that upon attainment of the specified rotational speed, rotor bending begins to increase sharply, and may surpass the bending caused by the static load. In this case, the rotor bearings experience loads of alternating sign, causing shaking of the aircraft, and stresses arise in the shafts, exceeding permissable values in a number of cases.

As has been shown by a large number of investigations there phenomena can be attributed to the resonances in the forced bending vibrations of the rotor. Resonance is characterized by coincidence of the frequencies of natural vibrations with the frequencies of forced vibrations. The rotor speeds corresponding to resonance are called critical speeds.

Rotors of gas-turbine [GTD] engines comprise systems consisting of elastic shafts coupled to disks having mass. Such a system can vibrate under the action of periodically changing loads. In general, the rotor is subject to torques variable in magnitude, generating torsional vibrations, longitudinal loads, generating longitudinal vibrations, and transverse loads, generating transverse (bending) vibrations.

Experience in engine construction indicates that rotor bending vibrations in gas-turbine [GTD] engines represent the greatest danger to engine strength. For rotors of turboprop [TVD] engines, bending and torsional vibrations may be harmful. Longitudinal vibrations are not observed in modern designs.

The effort to lighten housing designs leads to the result that their mass and bending stiffness is commensurate with the mass and bending stiffness of rotors. On this account, the housings and the entire power plant are involved when the rotors vibrate.

In case of vibration in which inadmissibly large stresses or deformations are caused, the following design or constructional measures may be applied for decreasing or removing the harmful resonances beyond the operational range of engine rpm:

1) decreasing the magnitudes of the exciting forces and moments by evening out the gas flow around the periphery of the flow passage and using a more carefully balanced rotor; .

2) removal of harmful resonances from the operational range of gas-turbine [GTD] engine regimes by change in the frequency of natural vibrations or changing the frequency of the excitation.

If harmful speeds are displaced beyond the minimum operational rpm, the rotor is called flexible; if the harmful speeds are displaced beyond the maximum operational rpm, the rotor is called rigid. The required rigidity of such a rotor is achieved by increasing the rigidity of the shafts and their connections with the disks, and also by increasing the rigidity of the supports; all this ordinarily is achieved at the expense of increasing the weight of the design.

In case the gas-turbine [GTD] engine has a flexible rotor, the engine goes through the critical speeds during start-up.

If the rotor is flexible, the loads on the supports caused by imbalance of the rotor are less than are observed in a gas-turbine [GTD] engine with a rigid rotor at the same rpm. The deficiency of the flexible rotor is the possibility of the incidence of large bending deflections in aircraft maneuvers or landing, as well as the possibility of striking the housing. Engine vibrations are observed upon transition through critical rotor speeds. If rotor deflections upon transition through critical speeds exceed allowable values, deflection limiters or damping equipment are used.

3. STRESS-BEARING SYSTEMS OF HOUSINGS.

The stress-bearing systems of gas-turbine [GTD] engine housings may be classified according to the method used to connect the housings of the compressor and turbine, or the housings of the reduction gear and the turbo-compressor (in the case of the turboprop [TVD] engine). On the basis of such a criterion, all stress-bearing systems of housings may be divided into three types.

1. The stress-bearing system of housings with an inside (in relation to the combustion chamber) stress-bearing link between the compressor housing and the turbine housing. Such, for example, is the arrangement of the stress-bearing system of engine housings with a centrifugal compressor (Fig. 144), and also of an engine with an axial compressor and individual combustion chamber.

The linking of the elements 11, 12 and 13 of the turbine housing with the housing 4 of the compressor diffuser is accomplished by use of the housing 8, hood 9, the drum 10 of the gas collector, and also the housings 7 and 6 of the rear and middle rotor supports, and the rear stress-bearing frame 5.

2. The stress-bearing system of housings with an outside (in relation to the combustion chamber) stress-bearing link between the compressor and turbine housings is shown in Fig. 145. The forward housing 4 of the forward turbine rotor support and the housing 3 of the rear compressor rotor support are rigidly attached to the front of the cylindrical housing 5 of the combustion chamber. The housings 3 and 4 insure attachment of the housing 2 of the compressor with the shell 5 of the combustion chamber. The housing 6 of the rear turbine rotor

support with the exhaust assembly 7 is attached to the rear face of the combustion chamber. In this case the combustion chamber is situated inside shell 5, and the turbine rotor shaft is thermally insulated from the combustion chamber by a system of baffles and liners. Cooling air is ducted through the space between the walls of the liners or thermally insulating layers of asbestos are set between them.

The considered stress-bearing is superior to the previous one in its greater rigidity to bending and torsion for smaller wall thickness; but at the same time it exhibits some shortcomings. Of these we should mention the impossibility of access to a combustion chamber in use without disassembly of the engine, and the complexity of turbine rotor-support design when it is made from an aluminium alloy casting. Furthermore, it is necessary to protect the housing of such a support from the action of the stream of hot gases.

The double-contour stress-bearing system of housings is a combination of the first two arrangements. In a double-contour stress-bearing system, the outer and inner contours are closed off into one or two zones.

In the diagram of the stress-bearing system of the housing shown in Fig. 146, both the inside housing 5 and the outside housing 4 are welded fabricated of sheet steel. The combustion chamber is situated in the space between them.

The stress-bearing closure of the front contours is affected by blades 3 of the stator assembly in the last compressor stage.

The stress-bearing closure of the rear contours, near the turbine, is effected by a special stress-bearing frame consisting of two rings 6 and 7, connected by struts 8. The struts are arranged radially between the combustion-chamber flame tubes; they transmit part of the loading from the support to the outer contour, and increase the rigidity

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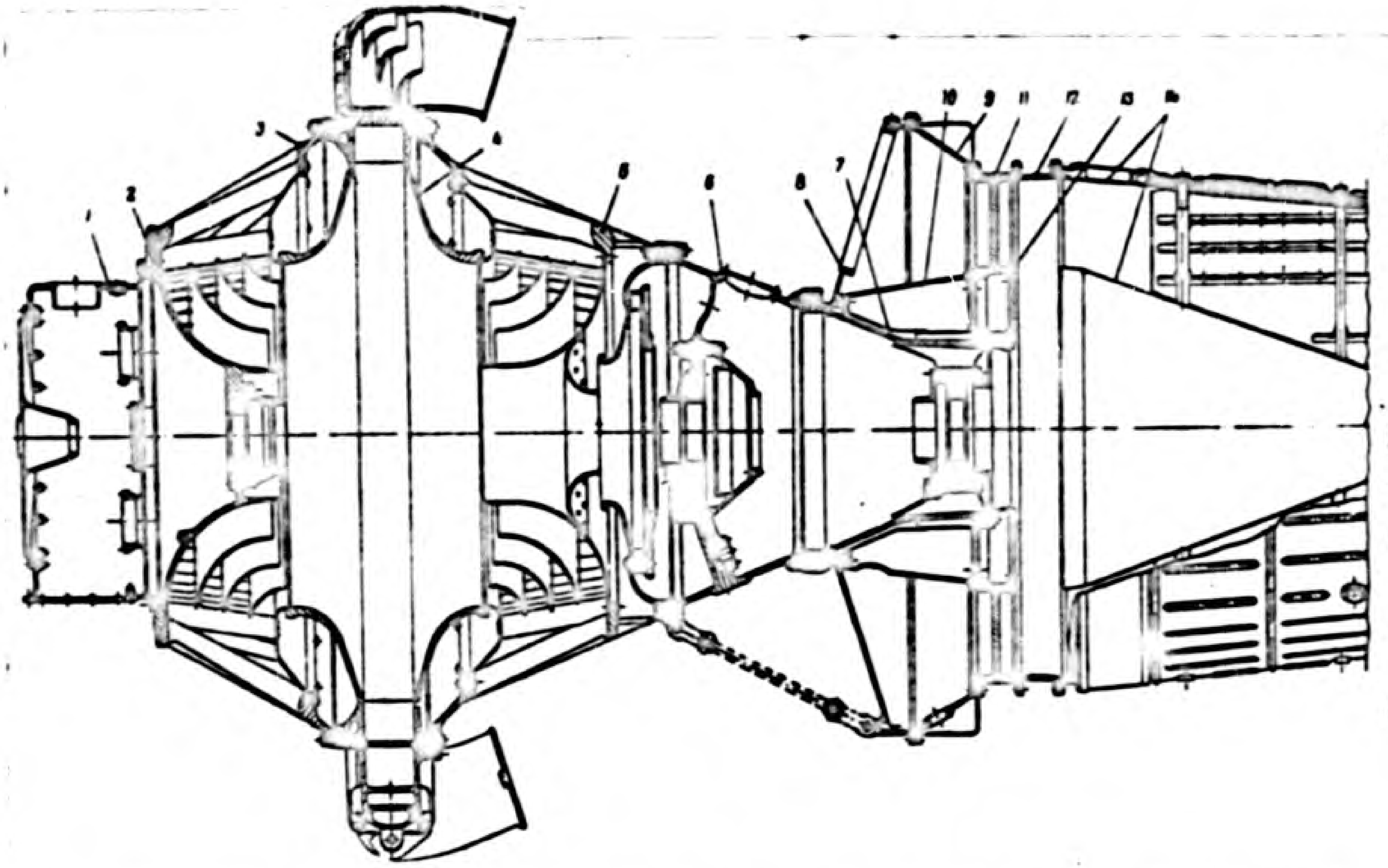

Fig. 144. Stress-bearing system of the housing of a turbojet [TRD] engine with centrifugal compressor and inside stress-bearing link between the compressor and turbine housings. 1) Drive compartment housing; 2) forward support housing; 3) forward stress-bearing frame; 4) diffuser housing with forward removable and rear hoods; 5) rear stress-bearing frame; 6) middle support housing; 7) rear support housing; 8) discharge-nozzle housing; 9) cover; 10) drum; 11) outside nozzle-diaphragm housing; 12) turbine-wheel housing; 13) inside nozzle-diaphragm housing; 14) outlet section.

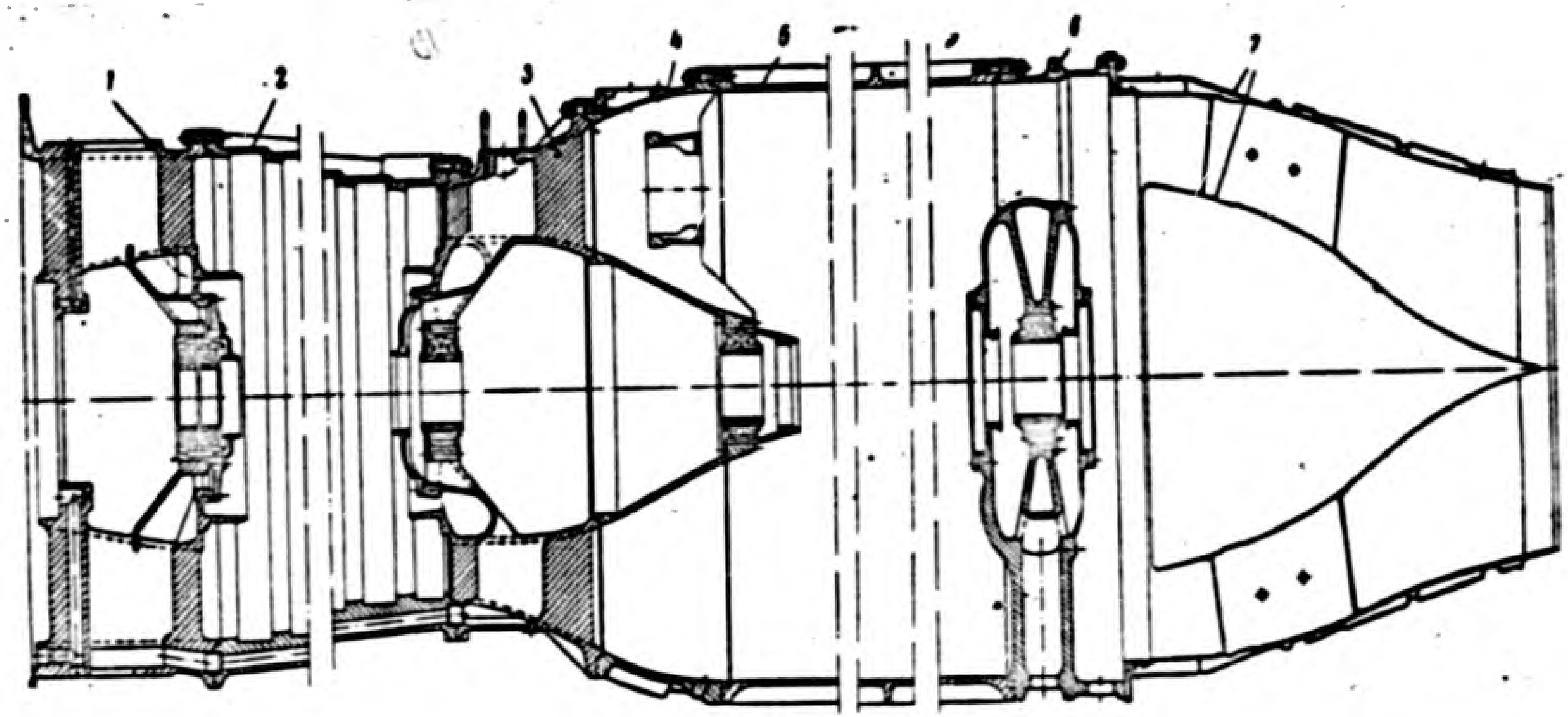

Fig. 145. Stress-bearing system of a turbojet [TRD] engine housing with axial compressor and outside stress-bearing link between the compressor and turbine housings. 1) Forward support housing; 2) compressor housing; 3) rear compressor rotor support housing; 4) forward turbine rotor support housing; 5) combustion chamber housing; 6) rear turbine rotor support housing; 7) outlet section.

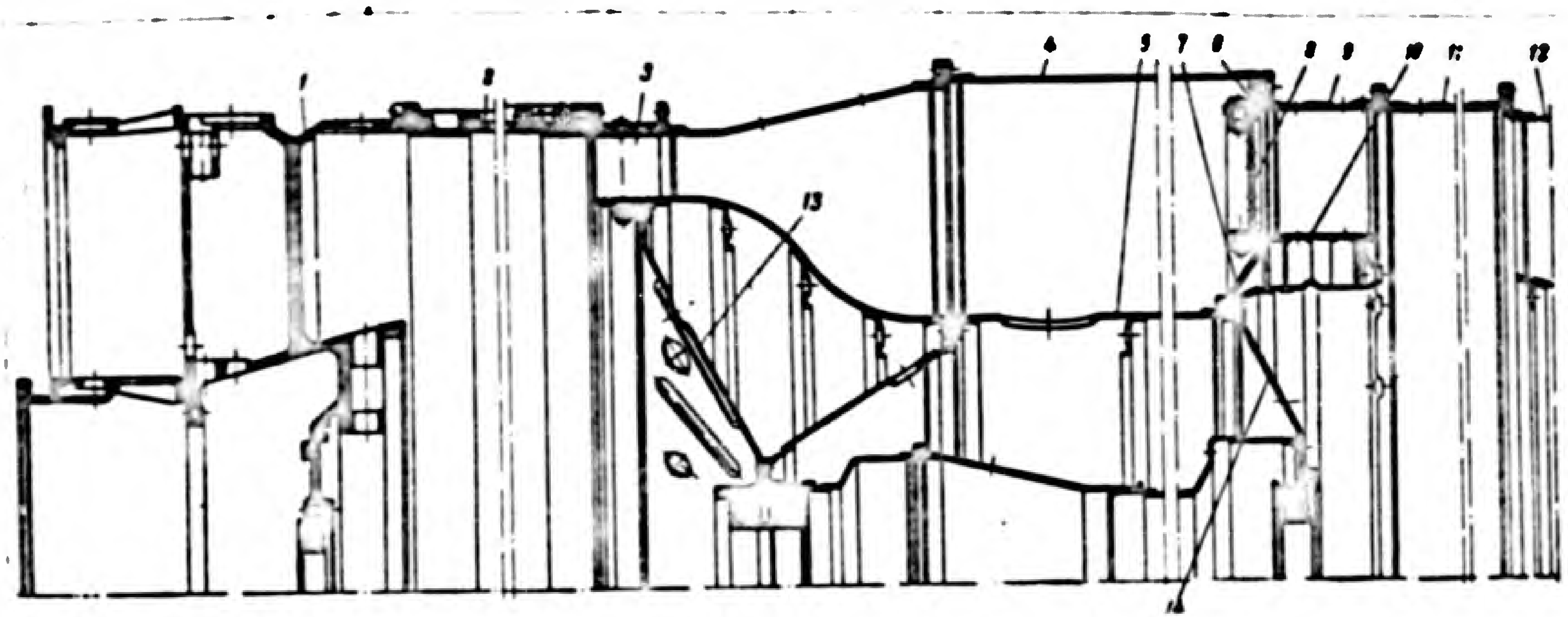

Fig. 146. Double contour stress-bearing system of the housing of a turbojet [TRD] engine with axial compressor. 1) Forward rotor support housing; 2) compressor housing; 3) vane of stator assembly; 4) outside combustion chamber housing; 5) inside combustion chamber housing; 6 and 7) stress-bearing frame rings; 8) stress-bearing frame strut; 9 and 11) outside turbine nozzle-diaphragm housing; 10) inside first stage nozzle-diaphragm housing; 12) outlet section; 13) middle support housing; 14) rear support housing.

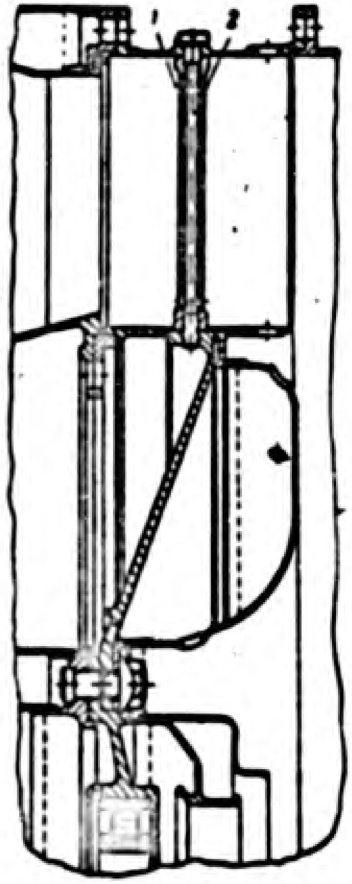

Fig. 147. Stress-bearing closure of the outside and inside contours by use of spokes. 1) thrust bushing; 2) screw.

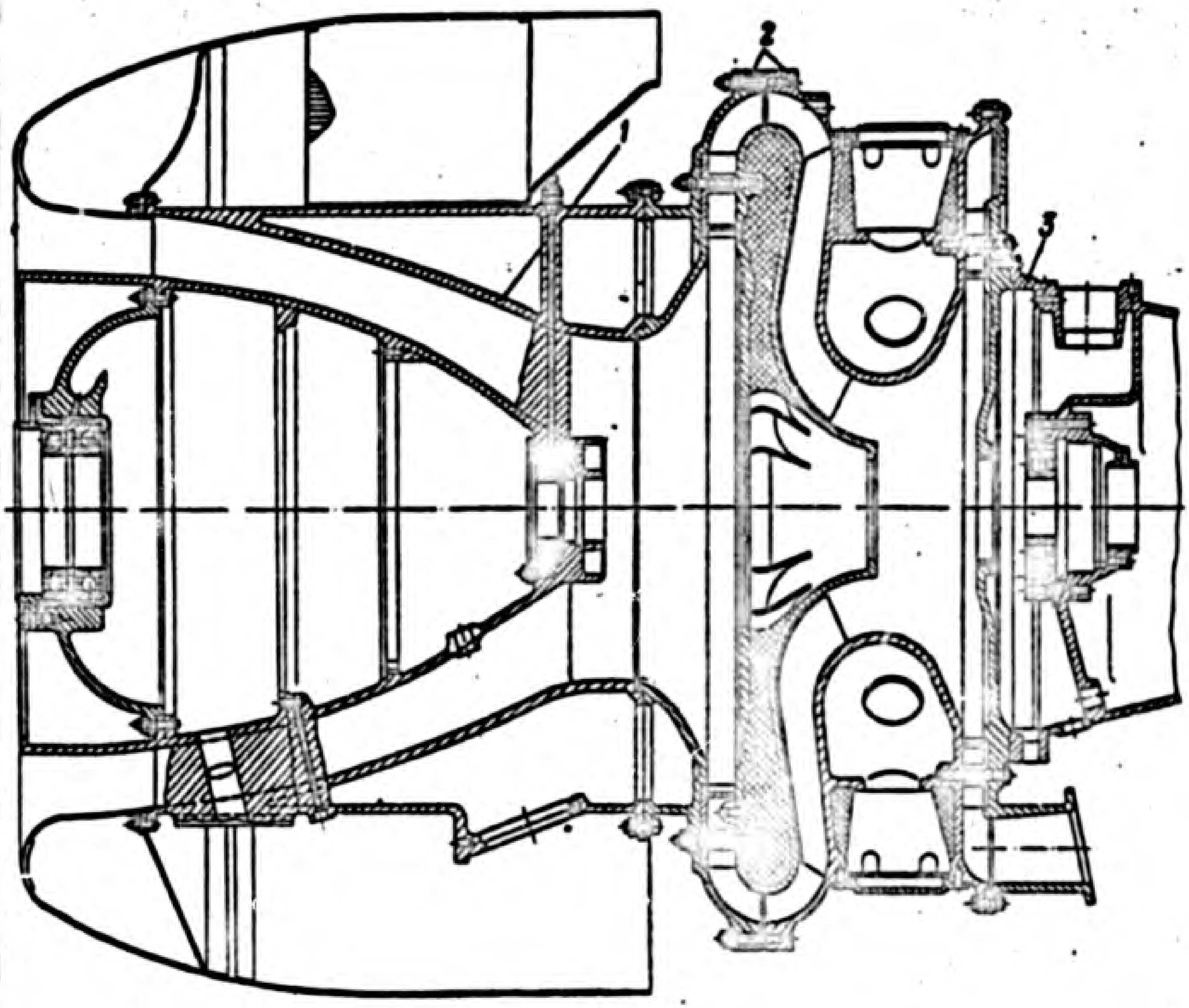

Fig. 148. Analysis of the stress-bearing system of the housings of the reduction gear and turbocompressor. 1) Reduction gear housing; 2) compressor housing; 3) turbocompressor rotor support housing.

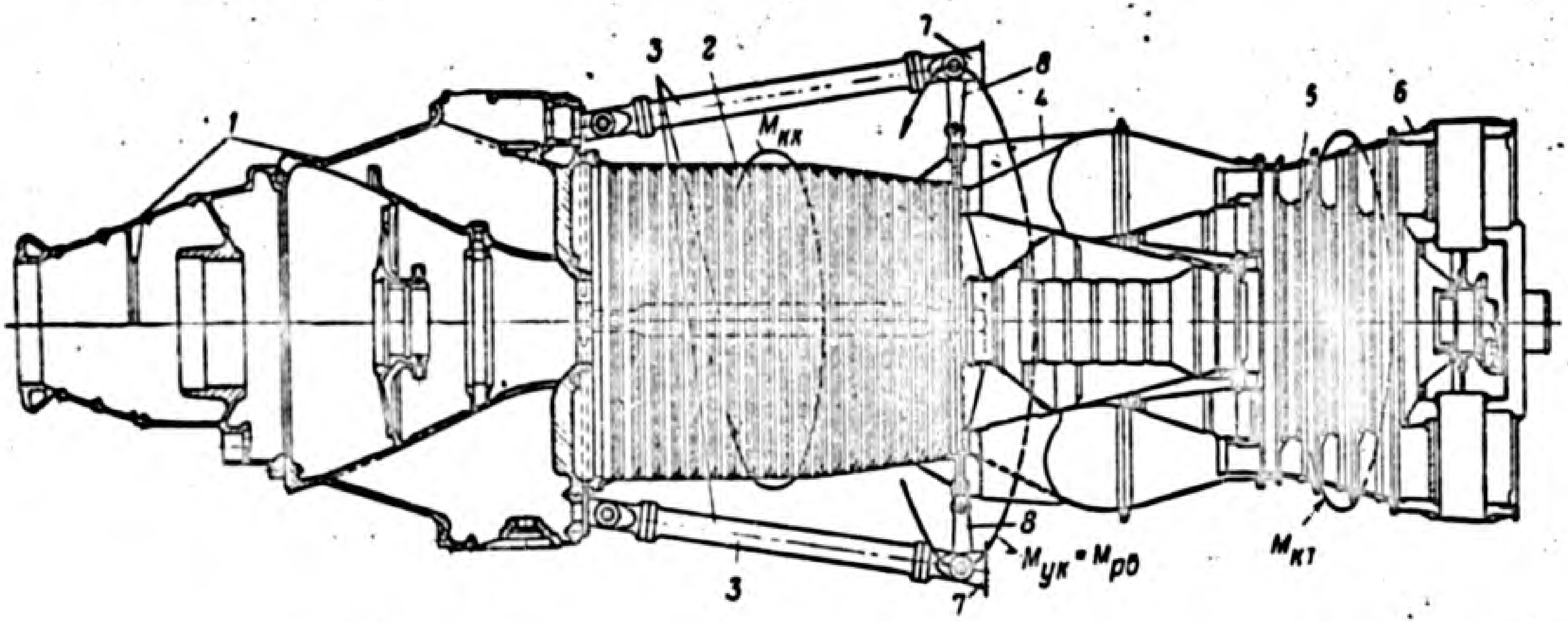

Fig. 149. Stress-bearing system of a turboprop engine housing. M_{kk}) Torque on compressor housing; M_{uk}) torque on points of engine attachment to the aircraft; m_{RV}) reaction moment from propeller; M_{kt}) torque on turbine housing; 1) reduction gear housing; 2) compressor housing; 3) stress-bearing rods; 4) combustion chamber housing; 5) outside turbine housing; 6) rear turbine support housing; 7) point of engine attachment to aircraft; 8) angle bracket.

of the system.

In order to relieve the housing, the frame may be replaced by spokes formed with thrust bushings 1 and screws 2 (Fig. 147), joining the outside and inside nozzle-diaphragm shrouds of the first turbine stage. The spokes pass through the hollow blades of the turbine nozzle diaphragm.

As a means of achieving a stress-bearing link between the reduction gear of a turboprop [TVD] engine and the basic stress-bearing parts of its turbocompressor housing, the following three arrangements are widely used.

1. A stress-bearing system in which housing 1 of the reduction gear is attached by the walls above directly to housing 2 of the compressor (Fig. 148).

2. A stress-bearing system in which housing 1 of the reduction gear is attached by the walls to the housing 2 of the compressor, and at the same time by stress-bearing tubular rods 3 with points 7 of engine attachment to the aircraft (Fig. 149).

3. In case power to the propeller is transmitted through a retractable reduction gear, it may have either independent points of attachment to the aircraft or a stress-bearing link with the engine through the use of a three-dimensional bracing girder.

Chapter 10

DESIGN OF TRANSMISSION ELEMENTS OF GAS-TURBINE ENGINES

1. ROTOR SHAFTS AND THEIR CONNECTION COUPLINGS

Shaft design is governed by the type of engine, the stress-bearing arrangement adopted, and also by the number and disposition of supports.

In a number of cases, integral shaft parts include the drum 2 of the axial compressor (Fig. 150, a) or the impeller 5 of the centrifugal compressor (Fig. 150, b). Sleeves 1 are attached to the impeller or drum.

In order to decrease shaft weight and to insure high strength and rigidity with small weight, shafts are made hollow, with as uniform strength as possible, and with as large outside diameters as possible.

If the shaft is made of several parts, the connection between parts may be made rigid (see Fig. 150, a and c) by use of flanges and bolts, tight fitting grooves and hold-down nuts, or they may be made flexible by use of connecting splined couplings 6 (see Fig. 150, b). In the latter case the part of shaft 7 which is rigidly attached to the turbine disk is called the turbine shaft, and the part of shaft 3 which is rigidly attached to the compressor is called the compressor shaft.

Fillets of as large a radius as possible are fitted between the various intermediate parts of the shaft in the couplings, or conical transitions are used. By such means high fatigue strength is insured and the danger of the appearance of cracks during heat treatment is reduced.

Since the shafts of gas-turbine [GTD] engines are very heavily loaded, they are fabricated out of alloy steels of type 18KhNVA, 40KNMA, 12Kh2N4A, etc.

In case each part of the rotor of a gas-turbine [GTD] engine is mounted on two supports with independent axial attachment, the connection of the shaft parts between them is accomplished by use of two splined couplings 1 and 2, and a sleeve 3 (Fig. 151, a).

If, however, the rotor of a gas-turbine [GTD] engine is mounted on three supports, connection of the shaft parts is most usually accomplished by use of a splined coupling, a diagram of which is shown in Fig. 151, b. In such a connection arrangement permits only relative bending displacements of the connected parts up to some angle, but parallel displacements of the parts are not possible.

Thus the splined couplings of the typical arrangements considered here provide the possibility of rotation in the connected parts, even in the presence of bending and displacement, without inducing additional bending deformations or loads on the supports.

Further, such a movable (flexible) connection of the shaft parts permits:

substantial simplification of the fabrication technology for units in the stress-bearing system and ease of engine assembly, because the necessity of precise centering and simultaneous machining of the separate parts are removed;

relief and simplification of the stress-bearing system of the engine;

insuring more precise balance of the rotors of the turbine and compressor, independently of each other.

In fabrication of connection units special attention must be devoted to fabrication precision with respect to the grooves (spacing, thickness of the tooth), which determines the number of teeth effective in operation, and good lubrication of the splined coupling is assured during operation. The grooves are, as a rule, evolutes.

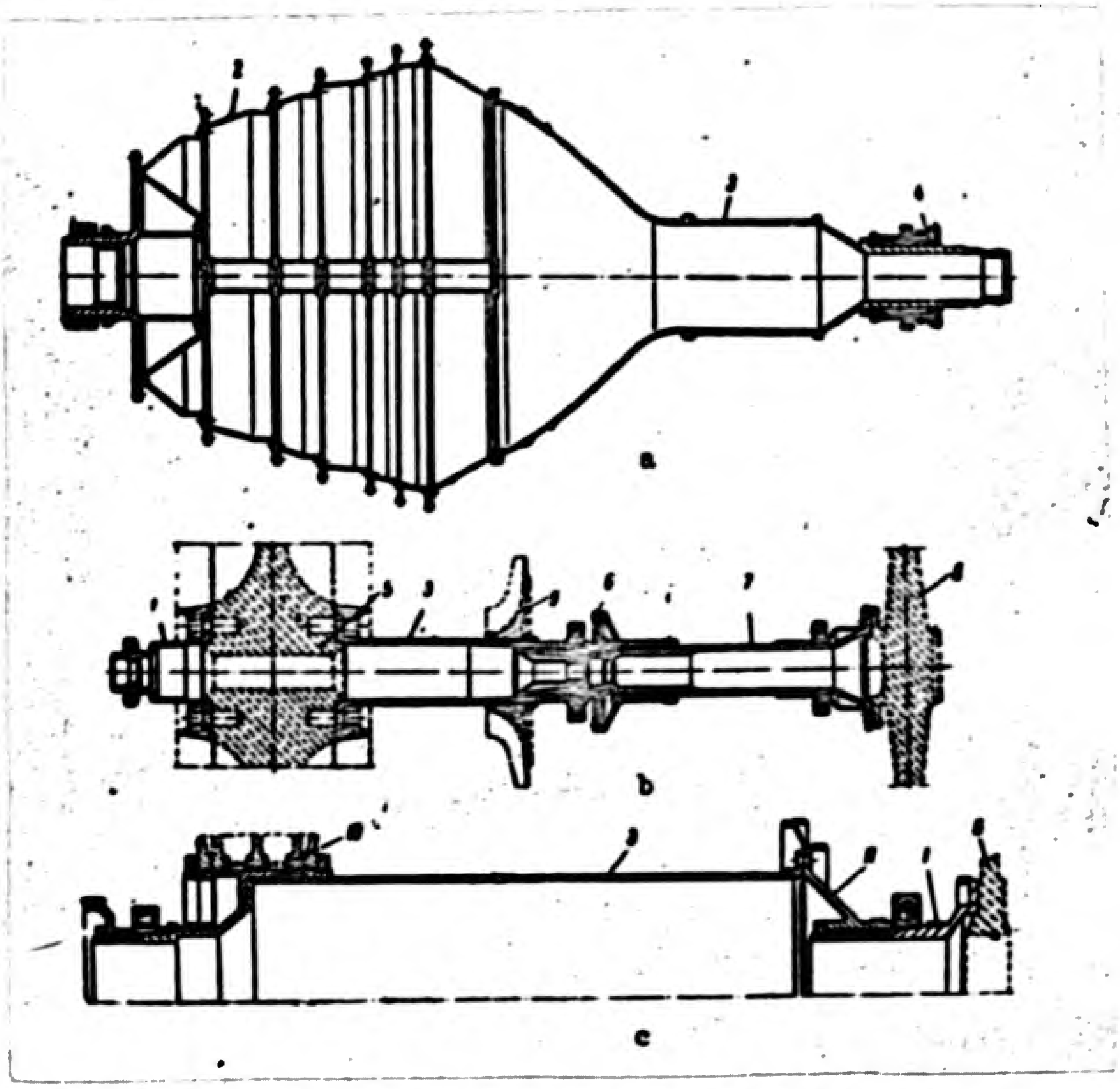

Fig. 150. Gas-turbine engine transmissions. a) with a two-support rotor and disk-drum compressor; 3) a centrifugal compressor; c) with two-support rotor and disk compressor; 1) journal; 2) drum; 3) compressor shaft; 4) sleeve for attachment of the turbine disk; 5) compressor impeller; 6) splined coupling; 7) turbine shaft; 8) turbine disk; 9) fan impeller; 10) compressor; 11) connecting coupling.

In relation to the forces transmitted, splined couplings are classified as couplings transmitting only torques, torques and axial loads, or torques, axial, and also radial loads.

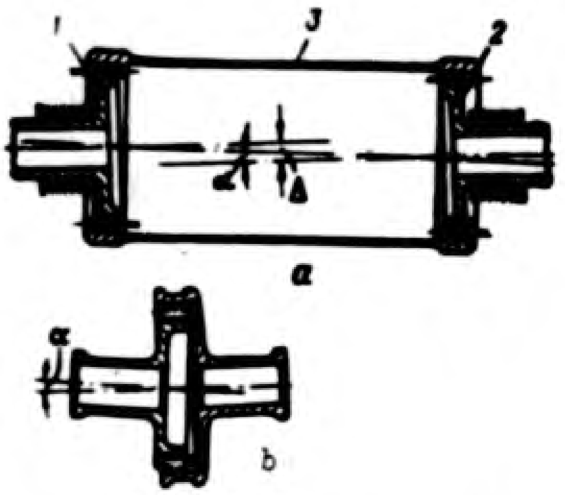

Fig. 151. Diagrams of shaft connections. a) Connection by use of two splined couplings and a sleeve; b) connection by use of one splined coupling; α) angle of bend; Δ) parallel displacement of the shafts; 1 and 2) coupling; 3) bushing.

Grooved couplings transmitting only torques from the turbine shaft to the compressor shaft may be built in several design variants. Figure 152 shows a connection coupling with intermediate sleeve 5, having inner evolute splines. The shafts of the turbines 8 and compressor 1 have rectangular splines, on which are mounted respectively the driving 6 and driven 2 parts of the coupling with outside evolute splines. The driving and driven parts of the coupling are centered on the shafts by cylindrical collar 10. Positioning of the intermediate splined bushing in the axial direction relative to one of the parts of the coupling is accomplished by an opening spring ring 3 and a stop in a projection 4 on the splines. On the hub of the driving part of the coupling there is a toothed rim 8 for driving an oil pump. Lubrication of the splined connection is accomplished by splashing oil fed for lubrication of the rotor bearings through the tube 9. For prevention of oil falling inside the shafts, the cavities are closed by plugs 11.

Figure 153 shows splined coupling in which the outside evolute splines are made directly on shafts 4 and 3 of the turbine and compressor, respectively, and the inside splines are made on the intermediate bushing 1. Positioning of this bushing in the axial direction is accomplished by locking collars 2 at the ends of the splines in the shafts. The axial load from the turbine rotor is transmitted to the forward support-thrust bearing 5 by rod 6.

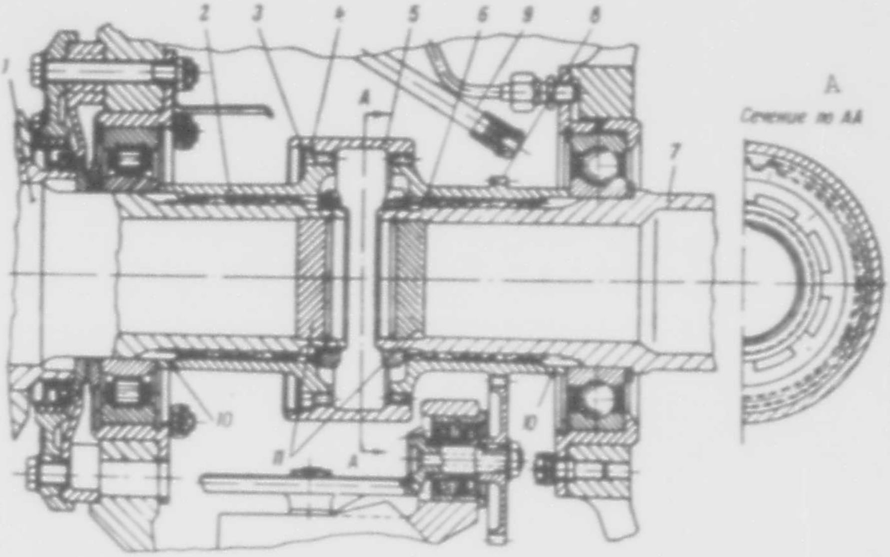

Fig. 152. Connection coupling with intermediate sleeve, transmitting only torque. 1) Compressor shaft; 2) driven part of the coupling; 3) opening spring ring; 4) spline projection; 5) splined bushing; 6) driving part of the coupling; 7) turbine shaft; 8) toothed rim; 9) tube with injector [spray nozzle]; 10) cylindrical collar; 11) plugs; A) section through AA.

The splined coupling is lubricated by splashing the oil fed for lubrication of the bearings.

A splined coupling transmitting torque and axial loading from the turbine shaft to the compressor shaft is shown in Fig. 154. Turbine shaft 1 is connected by evolute splines directly to rear journal 3 of the compressor. Setting of the turbine shaft for axial displacement and the transmission of axial loading is accomplished by screw 2.

Clearances in the splines of the connection and the axial clearance between the spacing bushing 4 and the turbine shaft insures the possibility of some bending of the shaft.

The splined couplings transmitting torque, radial, and axial loads, used in stress-bearing systems of three-support rotors, are shown in Fig. 155, a and b.

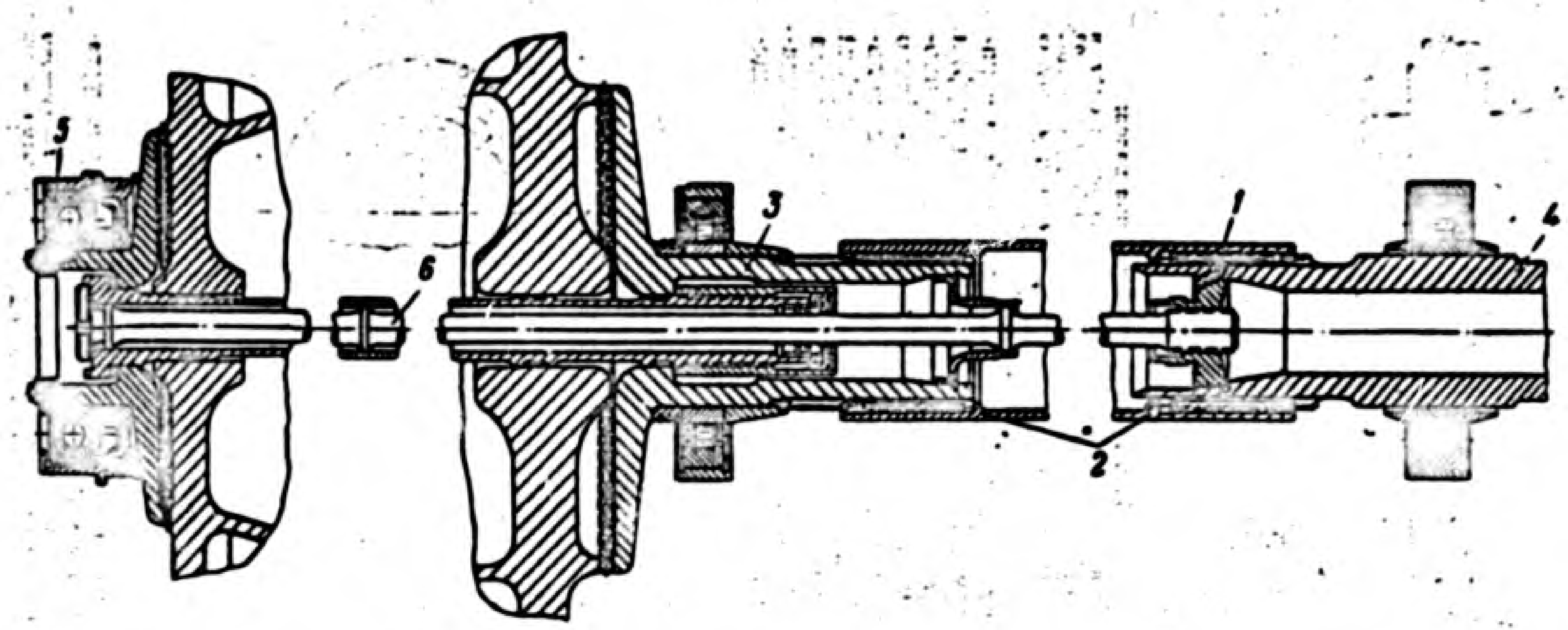

Fig. 153. Connecting units of shafts with elongated splined bushing, transmitting only torque. 1) Intermediate bushing; 2) shoulder; 3) compressor shaft; 4) turbine shaft; 5) support-thrust bearing; 6) rod.

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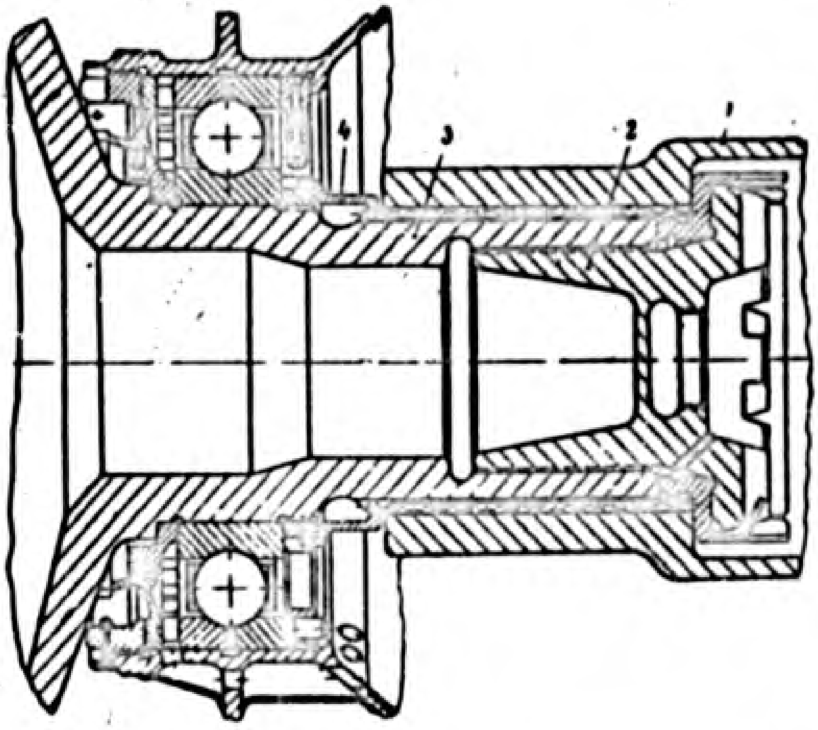


Fig. 154. Connecting unit of shafts transmitting torque and axial loading. 1) Turbine shaft; 2) screw; 3) rear compressor journal; 4) spacing bushing.

The driving part 6 of the coupling (Fig. 155, a) on turbine shaft 1 is fixed in the axial direction by retaining ring 2 and is centered by two cylindrical flanges 17.

The driven part 7 of the coupling is attached rigidly to compressor shaft 10 by use of rectangular splines and a hollow tie bolt 9, and is centered by two cylindrical flanges 18. The splined coupling splines 8 transmit torque. The spherical support 12 transmits radial and axial loads. The center of the support sphere is located in the plane of symmetry of grooves 8, which facilitates their work upon bending of the shafts.

Assembly of the spherical connection is accomplished through slots 14 on the rear of the turbine shaft and cover 4 of the socket, with subsequent turning of shaft 1 relative to shaft 10 through 60°.

For assembly and disassembly of the coupling considered, it is necessary to depress the radial pin 3, turn the retaining ring 2 through the width of the spline on the turbine shaft (in this case, the projections of the splines on the ring are situated opposite the indentations of the splines on the shaft) and move the driving part of the coupling in the direction of the turbine disk, disconnecting the splined coupline (splines) 8. Further, rotation of shaft 1 connects or disconnects the spherical connection. In order to insure the matching of the teeth, the relative positions of the coupling parts are kept constant during assembly by use of three locator pins 5 which are arranged irregularly around the circumference; the pins fit into holes on the driving part of the coupling. A constant position of the coupling parts relative to the shafts is insured by one broad spline 13.

The splined coupling is lubricated by injector 11, feeding oil from channel 15 of the main engine manifold into the splined coupling.

Figure 155, b shows another simpler shaft connection. The turbine shaft 1 is connected by splines with the rear compressor journal 19. Positioning of the turbine shaft in the axial direction is accomplished by use of connecting bushing 25 and bushing ring-holder 27. Bushing 25 has four lugs 29, evenly distributed around the periphery, a spherical supporting surface, and toothed sector 22 with two rectangular channels 28. Positioning of the connecting bushing in the axial direction relative to journal 19 is accomplished by four pins 23, which at the same time limit the rotation of the bushing.

At the end of thr turbine shaft there are a ring channel 26 and four lugs 24. An elastic plate lock 21 is attached to the journal 19.

Before assembly, bushing 25 is set up in the nonoperational mounting position, in which one of the lugs 29 is set opposite the wide splines on journal 19. In this position it is fixed by lock 21.

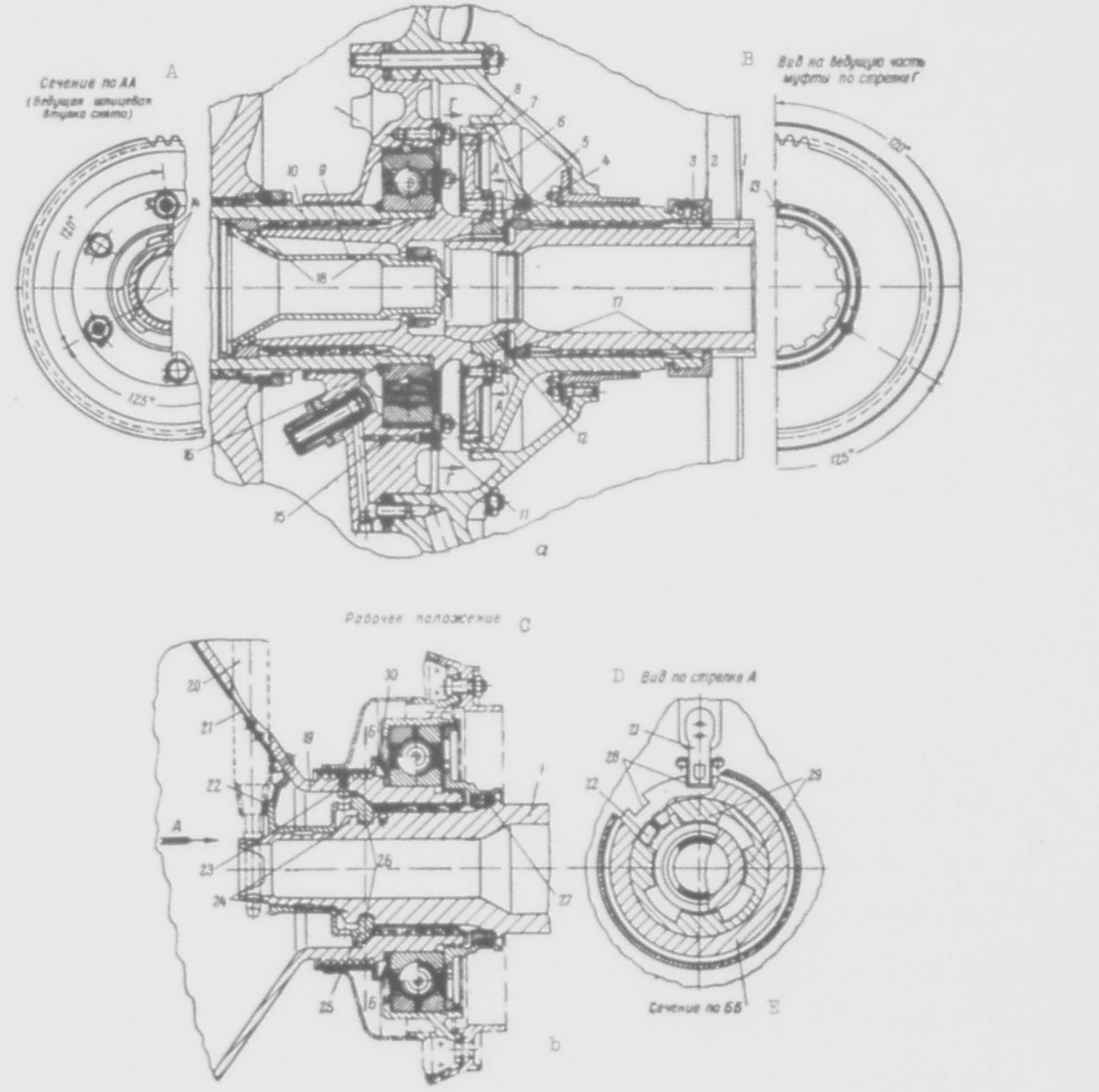


Fig. 155. Shaft connections transmitting torque, axial, and radial loads. a and b) Types of shaft connections; 1) turbine shaft; 2) retaining ring; 3) radial pin; 4) socket cover; 5) locator pin; 6) driving part of the coupling; 7) driven part part of the coupling; 8) splines; 9) tie bolt; 10) compressor shaft; 11) injector; 12) spherical support; 13) wide spline; 14) channels; 15) channel; 16) adjusting ring; 17 and 18) cylindrical flanges; 19) rear compressor journal; 20) key; 21) plate lock; 22) toothed sector; 23) pin; 24) lug; 25) bushing; 26) ring channel; 27) ring holder; 28) rectangular channels; 29) lugs; 30) regulating ring with radial vanes; A) section through AA (driving toothed hub removed); B) view on the driving part of the coupling along the arrow; C) operating position; D) view along arrow A; E) section through BB.

After shaft 1 is set up, the bushing is turned by key 20 into the operational position, in which its lugs are turned behind lugs 24 of shaft 1. Upon removal of the key, lock 23 enters into the second rectangular channel, fixing the bushing in the operational position.

Alloy steels 12Kh2N4A, 18KhNVA and 40KhNMA are used in the fabrication of the parts of the splined coupling.

2. ACCESSORIES, THEIR DRIVES AND ARRANGEMENT.

For the operation of a gas-turbine [GTD] engine, various accessories are mounted on it, i.e., fuel pumps and regulators, oil pumps, centrifugal separators and breathers, starting equipment, apparatus for regulating the compressor, transducers for indicating the rotor speed, etc.

Gas-turbine [GTD] engines are also provided with the following aircraft accessories: electrical generators, air vacuum pumps; compressed air compressors, hydraulic pumps, booster pumps, etc.

Accessories serving the engine are situated in special compartments directly mounted on the engine. The drives are located in the housings of the compartments. Accessories serving the aircraft may be mounted both on the engine and on special separate compartments mounted on the aircraft. In the latter case, only the drives for these compartments are provided for on the engine.

Modern turbojet [TRD] engines expend about 0.2 to 0.5% of the turbine power on driving the engine accessories, and 0.3 to 0.6% on driving the aircraft accessories. In turboprop [TVD] engines the power consumption is respectively, 0.1 to 0.17% and 0.15 to 0.25% of the turbine power.

Power for accessory drives is taken from the engine rotor shaft in front of the compressor, behind the compressor, or behind the turbine (in the case of rotor supports being present to the rear of the turbine).

The design of a central drive should insure minimum influence, elastic and thermal housing deformations on the magnitude of clearances in the meshing of the drive gears.

In turboprop [TVD] engines the power may be taken from the propeller reduction gear.

Drive compartments on which accessories are mounted can be positioned at any especially convenient place, depending on the geometry of the gas turbine [GTD] engine design.

In engines with centrifugal compressors the drive compartment is put at the forward part of the engine and is attached to the forward rotor-bearing housing.

Figure 156 shows a kinematic diegram of the accessory drives of an engine with centrifugal compressor having a double-sided inlet.

In an engine with an axial compressor, part of the accessories (starting equipment, booster oil pumps, centrifugal air separators, breather, etc.) may be mounted forward of the compressor on the bearing housing. However, for small diameter of the compressor hub, such a position is difficult. Furthermore, in this case access to the accessories is difficult in the process of their use.

Especially widespread are the following arrangements for location of accessory drives on the outside engine contour:

location of drives on the forward rotor bearing housing;

location of drives on the compressor housing with power takeoff from the rear compressor rotor journal;

location of drives on the compressor housing with power takeoff from the forward compressor rotor journal.

In the first and second arrangements for location of the accessory drives, the transmission of power to the drive compartment

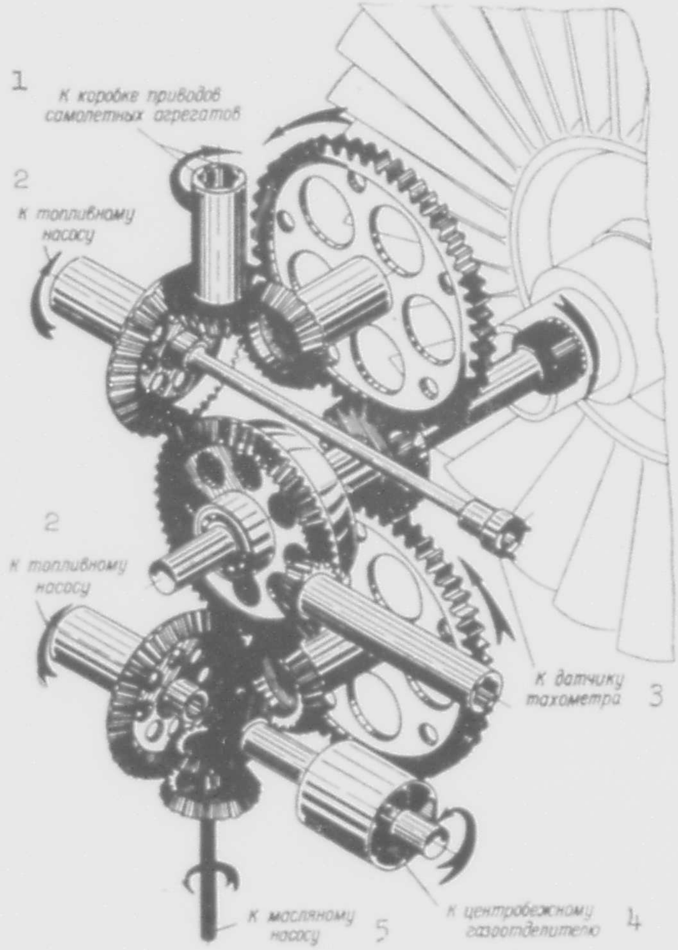


Fig. 156. Kinematic diagram of the assembly drives of an engine with a centrifugal compressor. 1) To the drive compartment of the aircraft accessories; 2) to the fuel pump; 3) to the tachometer transducer; 4) to the centrifugal separator; 5) to the oil pump.

(Fig. 157) is achieved by use of a radially located shaft (spring) with conical and cylindrical pinions. The second arrangement is used ordinarily in those cases when the first compressor stage is supersonic, with axial intake of air, in front of which there is no rigid forward bearing housing, or when the engine has a two-cascade compressor. In the latter case, the power take-off for the accessory drives is effected from the high pressure rotor, having relatively high speed in the low-power regime.

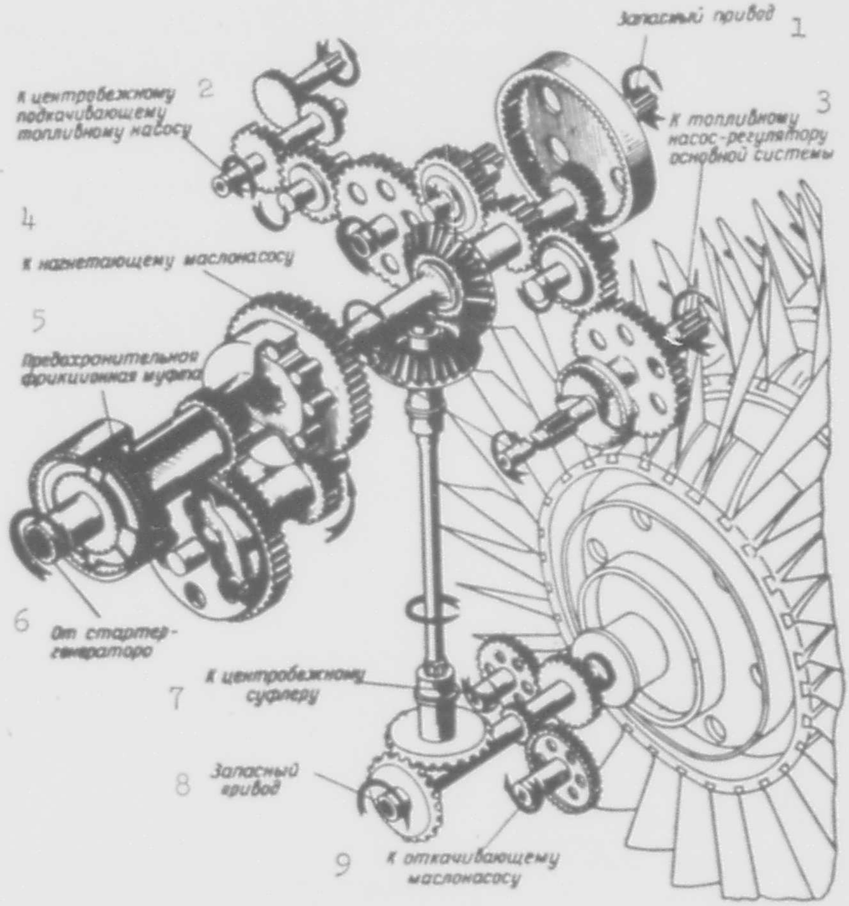


Fig. 157. Kinematic diagram of assembly drives in an engine with an axial compressor. 1) Spare drive; 2) to the centrifugal booster fuel pump; 3) to the fuel pump-regulator of the basic system; 4) to the pressure oil pump; 5) safety friction clutch; 6) from the starter generator; 7) to the centrifugal breather; 8) spare drive; 9)to the vacuum oil pump.

As for arrangement in the third manner (Fig. 158), such a disposition is resorted to on aircraft engines when access to the forward part is difficult. The rotation of the intermediate drives considered in this case is transmitted to drive the accessories by use of horizontal shafts.

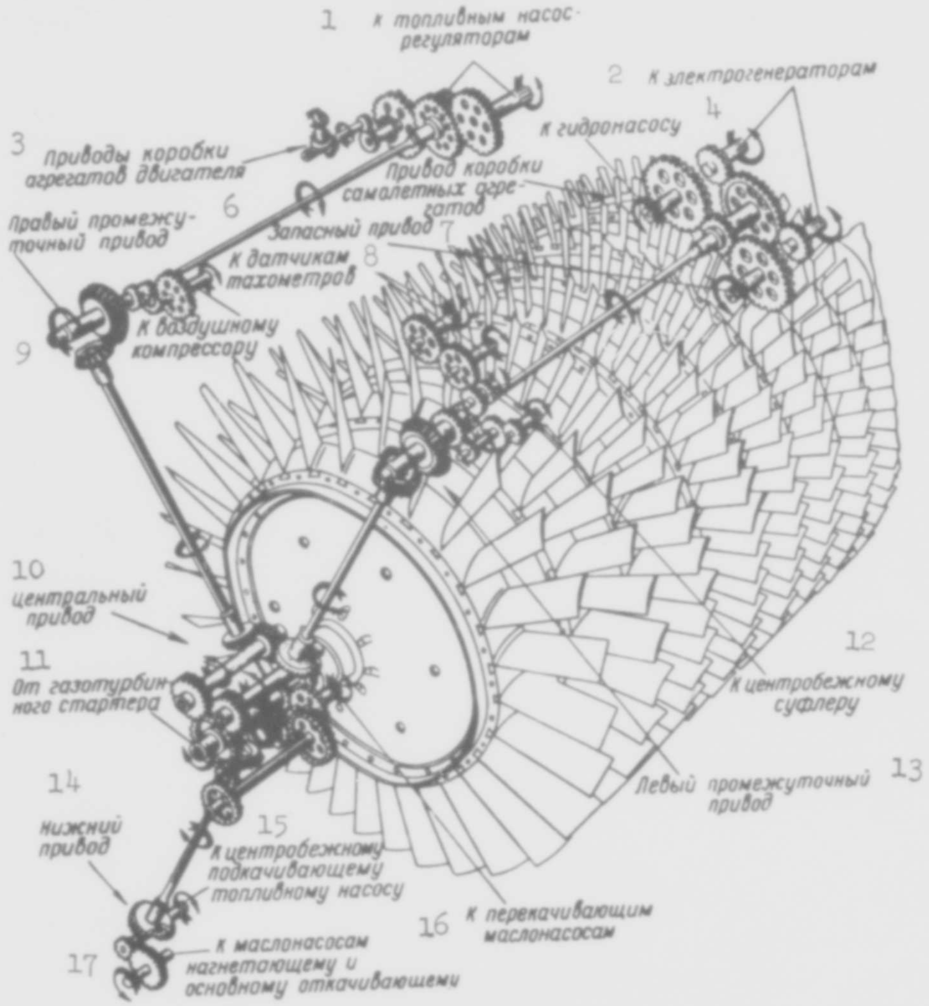


Fig. 158. Kinematic diagram of the accessory drives of an engine with axial compressor with intermediate drive. 1) To fuel pump-regulator; 2) to the electrical generators; 3) drives of the engine accessory compartment; 4) to the hydraulic pump; 5) drives of the aircraft accessory compartment; 6) right intermediate drive; 7) spare drive; 8) to tachometer transducer; 9) to air compressor; 10) central drive; 11) from the gas turbine starter; 12) to the centrifugal breather; 13) first intermediate drive; 14) lower drive; 15) to the centrifugal booster fuel pump; 16) to the oil tranfer pump; 17) to the oil pumps, pressure and main vacuum pumps.

Part Five

ROTOR BEARINGS AND LUBRICATION SYSTEMS

OF GTD [GAS-TURBINE ENGINE (S)]

Chapter 11

GTD [GAS-TURBINE ENGINE (S)] ROTOR BEARINGS

1. OPERATIONAL AND DESIGN FEATURES OF GTD [GAS-TURBINE ENGINE (S)] ROTOR BEARINGS.

Especially widespread in GTD [gas-turbine engines] is the use of roller bearings. Friction bearings may be encountered only in low-power, starting GTD [gas-turbine engines] and in accessory drives.

The limited occurrence of friction bearings is due to their large friction, increasing with increase in rotational speed, their large longitudinal dimensions, and also their increased requirement for purity in the lubricant, and their great sensitivity to the temperature of the latter.

The considerable heat generation in friction bearings at high rotational speeds requires increase in the oil circulation through the engine, and also the presence of powerful oil pumps, oil filters, and radiators. The increase in the clearance in bearings, necessary for insuring the required oil flow, impairs the balance of GTD [gas-turbine engine] rotors.

Roller bearings have smaller frictional forces, almost independent of the rotational velocity, a higher work capacity at high operational temperatures, and smaller longitudinal dimensions.

The small frictional forces are accompanied by reduced power loss, resulting in reduced heat generation and oil circulation which substantially simplifies the design of the lubrication system.

In modern GTD [gas-turbine engines, rotor bearings work at speeds of 5000 - 18,000 rpm. In auxiliary engines and in turbocooling installations the rotational speeds of the bearings reach 35,000 to 120,000 rpm [27].

The rotational speed determines the quantity of heat developed in the bearing and the strength of its separator. The strength of a separator is determined by the peripheral speed, depending on the diameter of the bearing and the rpm. Therefore, it is possible to render a judgement as to the separator loads on the basis of the magnitude of the peripheral speed at the circumference of the centers of the balls (rollers), which [the magnitude] in high-speed bearings does not exceed 50 - 80 m/sec, or on the basis of the magnitude of the product of the rpm and the inside diameter of the bearing. This product varies in the limits $0.5 \cdot 10^6$ to $1.6 \cdot 10^6$ (sometimes reaching $2.0 \cdot 10^6$) mm rpm.

The operational temperature reaches 180° C for compressor rotor bearings and 250° C for turbine rotor bearings.

With increasing flight speed in aircraft with GTD [gas-turbine engines], a tendency is encountered for the operational temperatures of the bearings to increase due to the increase in the deceleration [stagnation] temperature of the air.

High requirements are imposed on high-speed, strongly heated aviation bearings with respect to the geometrical precision of the roller shapes and the finish of their surfaces, as well as with respect to the quality of the materials used, the limits of permissible variation in the axial and radial clearances, the assembly clearances, and the quantity and manner of oil feed.

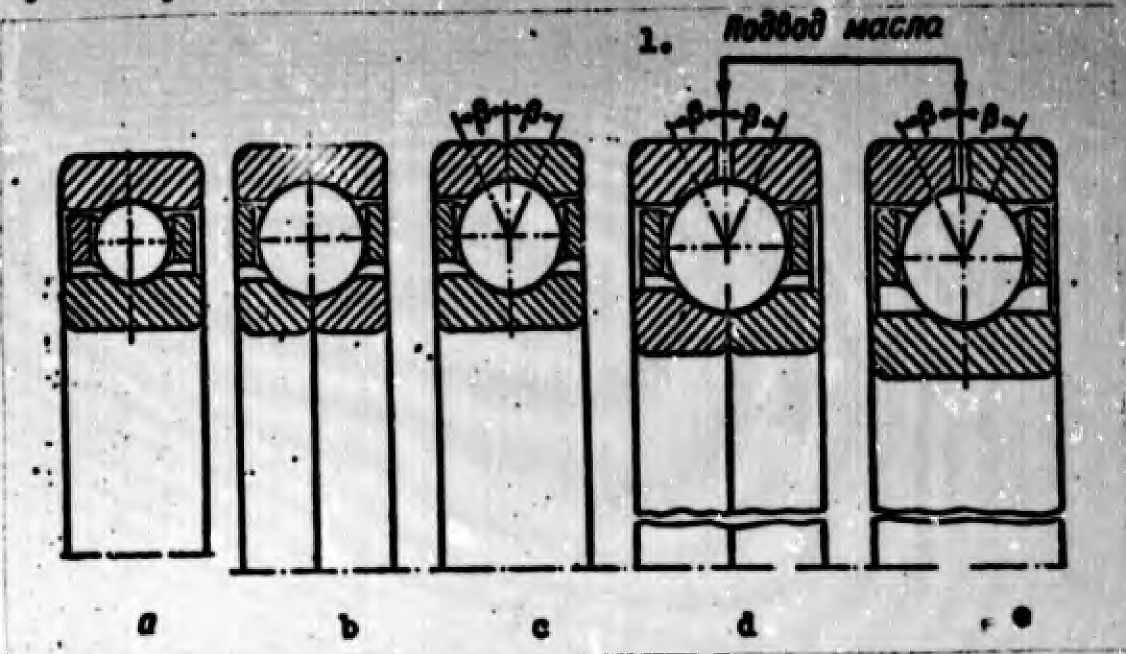


Fig. 159. Single-row ball bearings. a) Two-point; b) three-point; c, d and e) four-point; β) contact angle between ball and race; 1) oil feed.

Examples of ball-bearing design and roller-bearing used in aviation engines are shown, respectively in Figs. 159 and 160.

Single-row two-point ball bearings (Fig. 159, a) (two points of contact between the ball and the races) can absorb radial and small axial loads, and are used most usually in rotor supports of compressors and turbines.

Single-row three-point (Fig. 159, b) and four-point (Fig. 159, c, d and e) ball bearings can absorb radial and considerable axial loads, and are used ordinarily in rotor supports of compressors, turbines, and reduction gears of TVD [turboprop engines] for axial location of the shafts. The races for the balls in these bearings have larger depth. In order to facilate the assembly of such bearings, one of the races is split.

Roller bearings are used to absorb radial loads. Collars for containing the rollers are found on the inside or outside races, depending on the conditions of assembly, disassembly, and lubrication of the bearing.

A widely used steel for fabrication of the bearing races in production GTD [gas-turbine engines] is the ball-bearing chrome steel ShKh15; for roller shapes, ball-bearing chrome steels ShKh6 and ShKh9 are used, and for separators, bronze BraZhMts - 10 - 3 - 1.5 and aluminum alloys AK4 and D1T are used.

Chrome-nickel-molybdenum or tungsten tool steels are used for fabrication of races and roller shapes of bearings operating under conditions of elevated temperatures. The first steel retains high hardness up to temperatures of 370 - 400° C, and the second, up to 590° C. Nickel alloys may also be used [27] for separators working under conditions of high temperatures and inadequate lubrication.

In connection with the high-temperature operational regimes, GTD [gas-turbine engine] rotor bearings have larger radial and axial clearances to prevent the jamming of the rollers (balls) and the separators, which is possible due to expansion as a result of heating and deformation under the action of centrifugal forces.

Only bearings with separators are used for GTD [gas-turbine engine] rotors. Bearings without separators, in spite of their higher load-bearing capability, are not used at all because of the rapid incidence in them of the so-called "ring wear" effect, caused by contact between the balls during rolling.

Separators may be riveted or one-piece. They may be stamped or machined; the separators are centered with respect to the outside or inside race.

If the separator is centered with respect to the outside race, the lubrications of the inner race and the centered surface is improved, as is the balance of the separator during the process of operation and the removal of heat from the separator through the cooler outside race. Centering deficiencies with respect to the out-

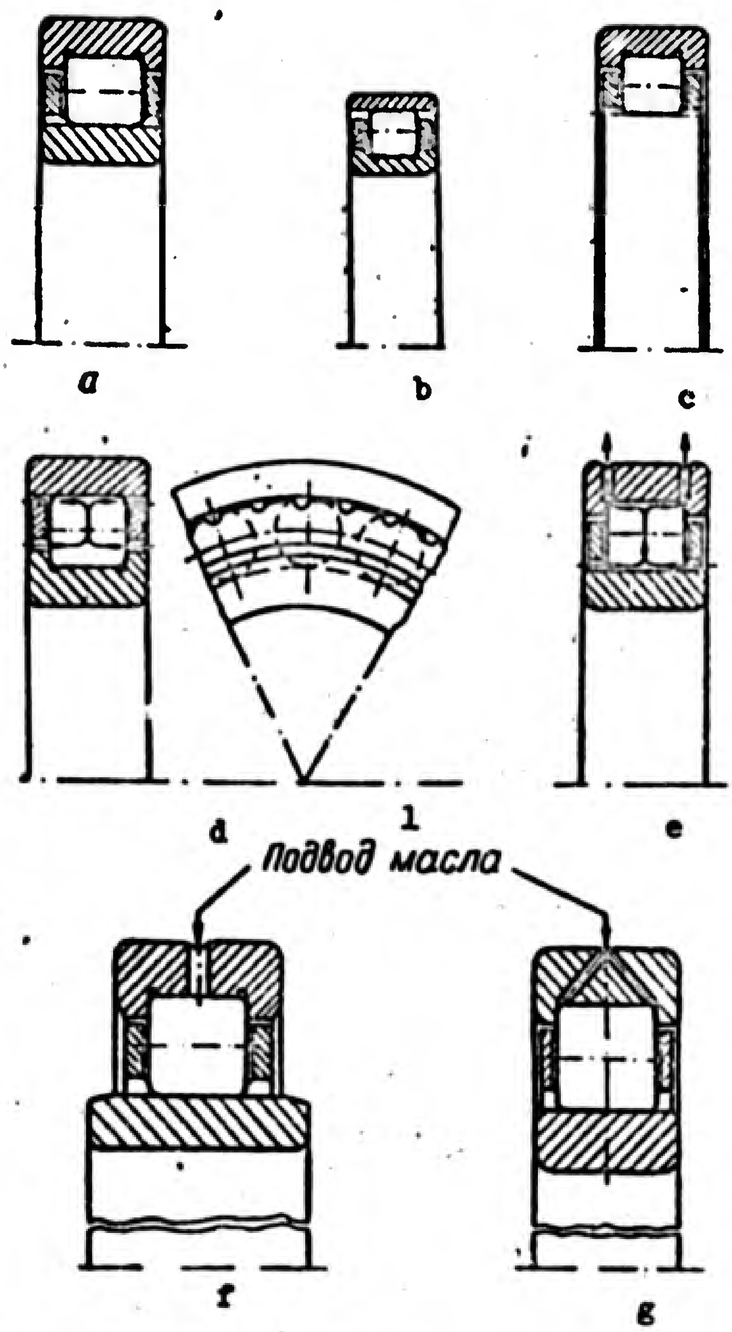


Fig. 160. Roller bearings. a, b, c, d, e, f and g) Types of design; 1) oil feed.

race include an increase in the weight of the separator, the necessity for precise choice of clearance between the separator and outside race in order to prevent seizing upon heating due to a difference between the coefficients of linear expansion for the materials and due to deformation of the parts under the action of centrifugal forces.

To reduce bearing temperature, oil circulation should be improved by machining holes or slits into the outer ring at the points of the race undercuts (Fig. 160, e), special milling of the separator (Fig. 160, d), making the outside ring without flanges (Fig. 160, b) or by taking other measures.

2. DESIGN OF BEARING SUPPORTS

Bearing supports are designed in conformity with the chosen arrangement of the stress-bearing system of the GTD [gas-turbine engine] rotor. Supports intended only for radial loading, depending on the magnitude of the latter and also on the temperature conditions, have one support roller bearing, a support ball bearing, or a combination of one roller bearing and one ball bearing.

Supports intended for radial and axial loads have from one to three support-thrust ball bearings.

The bearing race mated to the part that rotates relative to the load-bearing part (the shaft) is installed with a tight fit in order to prevent its rotation relative to the part; the race mated to the part that is stationary relative to the load-bearing part (the housing) is installed with a clearance so as to insure the possibility of rotation of the race relative to the housing.

In the fabrication of a bearing housing made of aluminum alloy, steel cups 4 (Fig. 161) are pressed into the sockets for the bearing, and this increases the rigidity of the support and permits retention of the initial clearances of the outside race. In a steel housing 3 (Fig. 162) the bearing is mounted directly.

In order to prevent wear as a result of the rotation of the outside race, the seated and supporting surfaces are carburized, chromized, or nitrided depending on the material of the housing or the cup.

Machining of the seated and supporting surfaces is carried out with a high order of precision, insuring proper installation of the

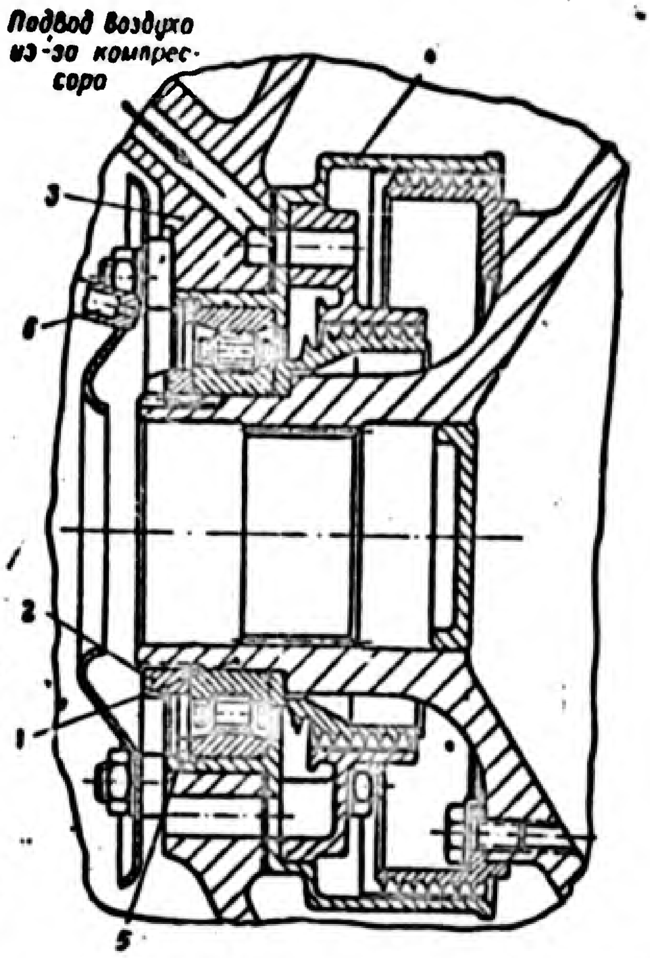


Fig. 161. Design of support with a single roller bearing. 1) Journal; 2) nut; 3) support housing; 4) cup; 5) spring ring; 6) oil injector; a) air feed from behind the compressor.

bearing in the housing.

Figure 155 shows an example of the design geometry of a support with a single support-thrust ball bearing. The specific feature of such supports is the presence, between the inside race of the bearing and the thrust collar on the shaft, of an adjustment ring 16 (see Fig. 155, a) permitting change in the axial position of the rotor relative to the housing. As shown in Fig. 155, b, the adjustment ring 30 has radial vanes to improve the circulation of an oil-air emulsion through the bearing.

Reduction of heat flow to the bearing from the surrounding more intensely heated parts is achieved by a d crease in wall thickness

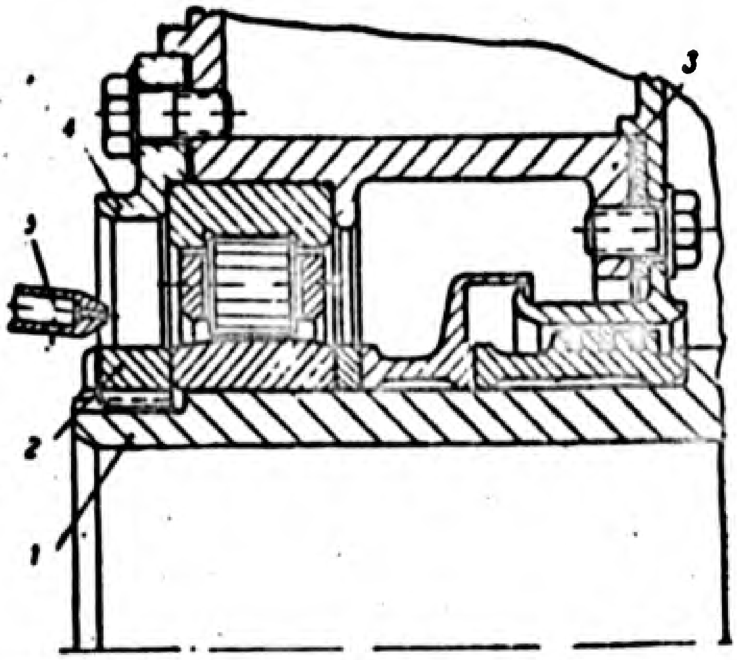


Fig. 162. Design of support with a roller bearing and installation of an outside race in a steel housing. 1) Journal; 2) nut; 3) steel housing; 4) clamping ring; 5) oil injector.

at points connecting the housing flanges and the cup and also by the installation of heat-insulating screens made of asbestos cloth or glass wool.

To insure uniform distribution of loading between the parallel-working support-thrust bearings (Fig. 163), conditions are maintained such that the simultaneous choice of operational clearances under the action of axial loading is insured. This is achieved by choice of the thickness of the rings 5 and 7, installed between the bearing reces.

In the assembly of the support, about the same fit is maintained on the shaft as in the housing. In the same way the rigidity of the housing and shaft for each bearing and the conditions of lubrication and cooling for these should be identical.

Turbine roller bearing supports (Fig. 164) are designed so that the inside race is ordinarily mounted on bushing 1, having small sur-

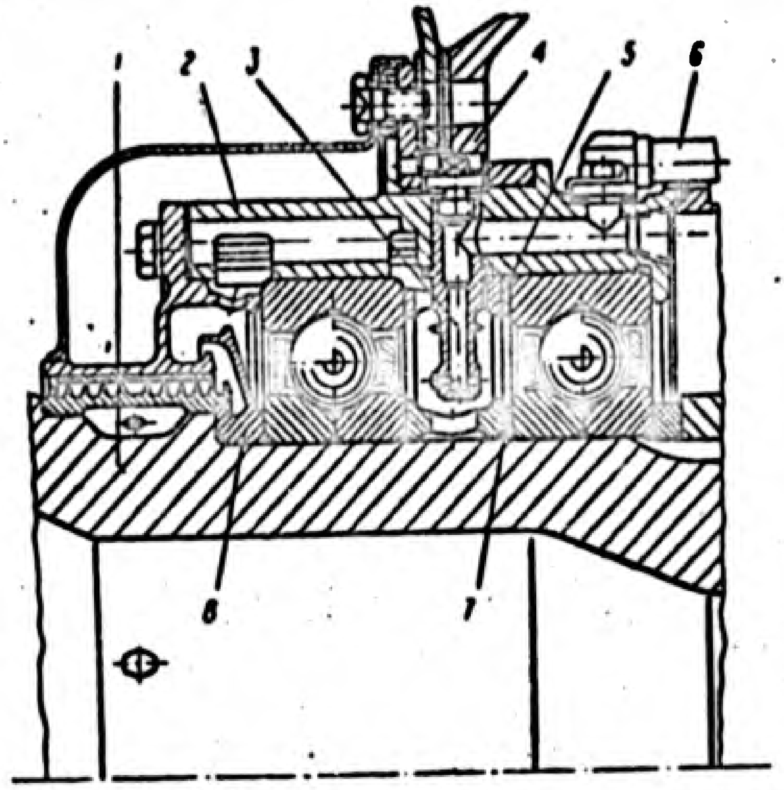


Fig. 163. Design of support with double parallel-working support-thrust bearings. 1) Journal; 2) socket; 3) injector [spray nozzle]; 4) support housing; 5 and 7) sized rings; 6) oil feed tube; 8) adjustment ring.

faces of contact with shaft 2 of the turbine through a cylindrical collar 6, with respect to which it is centered.

Supplying cooling air between sleeve 1 and shaft 2 of the turbine, and also between the struts of housing 8 (Fig. 164, b) is an efficient means of cooling the bearing and insulating it thermally.

Sleeve 1 at its forward end bears on spiral rectangular splines 7, situated on the shaft. The pitch of the splines is chosen so as to coincide with the direction of the relative velocity W of the airflow in the rated regime.

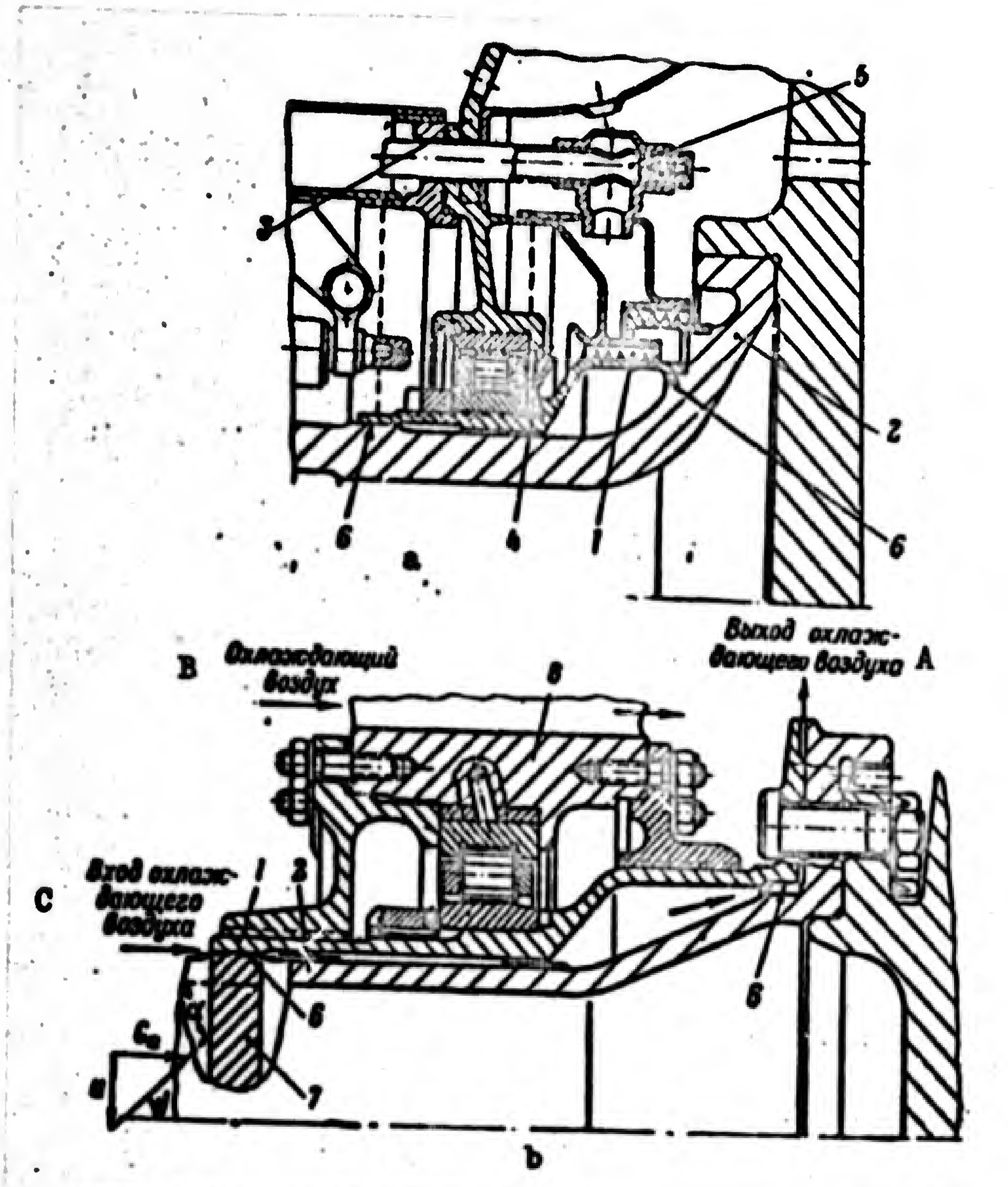


Fig. 164. Design of a turbine rotor support with roller bearing. a) With inside race mounted on an intermediate bushing; b) with air cooling; 1) intermediate bushing; 2) turbine shaft; 3 and 8) support housings; 4) ring with radial vanes; 5) tube for gas feed from labyrinth cavities; 6) cylindrical collars; 7) splines; A) cooling air outlet; B) cooling air; C) cooling air inlet.

Chapter 12

LUBRICATION SYSTEMS

1. GENERAL INFORMATION. ARRANGEMENTS OF LUBRICATION SYSTEMS.

Lubrication of flexible couplings in GTD [gas-turbine engines] is intended to decrease friction and parts wear, to remove the heat generated by friction and transmitted from neighboring more intensely heated parts, and to protect against corrosion and work hardening and also for removal of solid particles that might get into the space between the rubbing surfaces.

Frictional forces are generated as a result of the relative motion of touching bodies. Depending on the nature of the relative motion of the bodies in contact, the friction is classified as sliding or rolling friction. Sliding friction may be of three types:

dry friction, generated between nonlubricated surfaces of solid bodies;

liquid friction, when the rubbing surfaces of the bodies are separated by a layer of liquid;

boundary friction, when the lubricated surfaces of the bodies are partially in direct contact. Rolling friction appears by reason of the deformation [indentation] arising at the point of contact between a body rolling over another.

When component parts are in rolling contact with one another, the introduction of a lubricant between them has no significant effect on the magnitude of the frictional forces, and even increases them. However, in this case parts lubrication is necessary, since the presence of an oil film on the parts promotes more uniform distribution of the concentrated loads and improves the durability of the parts. Furthermore, rolling friction in all cases inevitably also involves some sliding friction, due not only to the mutual deformation of the rolling parts, but also because of the presence of races and separators, in rolling contact with the bodies.

The load-bearing capability of an oil layer, all other conditions being equal, depends on the viscosity of the oil. The more viscous the oil, the greater its load-bearing capacity. However, hydraulic friction arises in the oil at the same time. The type of oil which has a flatter viscosity-temperature characteristic, i.e., the kind of oil for which the effect of temperature on viscosity is not significant, is preferable. Too great oil viscosity at low temperatures decreases circulation through the engine and makes starting harder, and too low oil viscosity at high temperatures may lead to boundary-layer friction and rubbing-surface wear.

The kind of oil used is determined by the loads on the parts, their operational temperatures, and the type of bearings used. The operational temperature of the oils does not exceed 120 - 140° C in GTD [gas-turbine engines] at subsonic flight speeds, and may reach 250 - 400° C in GTD [gas-turbine engines] at supersonic flight speeds.

For work under elevated temperature conditions, existing types of oil are ordinarily improved by use of special additives. Besides this, new kinds of synthetic lubricants are being created, i.e. liquid, gaseous lubricants, and solid greases. The basic solid lubricants are graphite, molybdenum disulfide, etc. At the same time, special cooling systems for bearings are used, and in there the cooling and lubricating materials are gases with high heat capacity (helium,

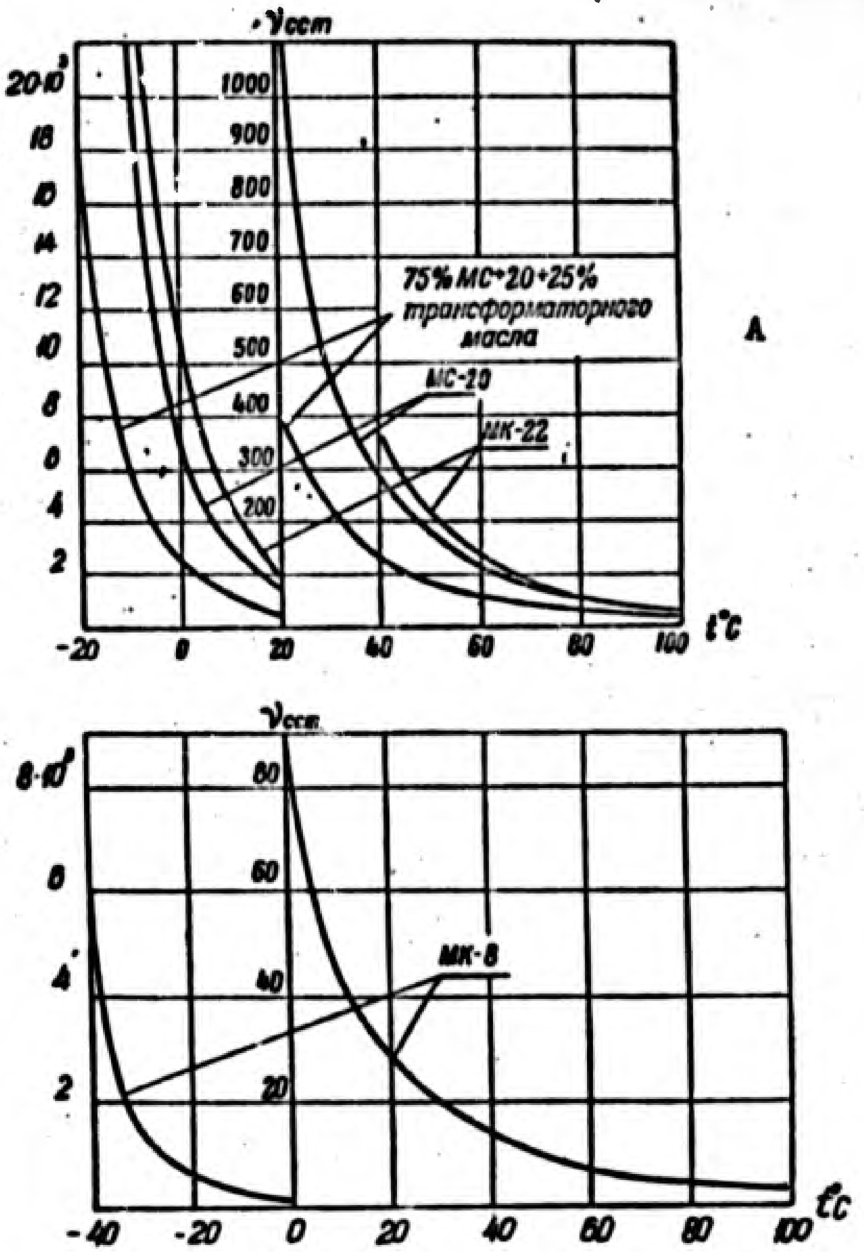


Fig. 165. The viscosity-temperature characteristic of oil. A) 75% MS + 20 to 25% transformer oil.

for example), insuring intensive heat removal from the parts [27].

For TRD [turbojet engines], having only roller bearings, working at high rpm, mineral (MK - 8) and transformer oils with reduced viscosity and flat viscosity-temperature characteristics (Fig. 165) as well as oils consisting of diesters of organiz acids [27] are more widely used at present.

From the point of view of increasing the contact-fatigue resistance of bearings and heavily loaded gears of TVD [turboprop engine] reduction gears, it is desirable to use oils with high viscosity. However, in this case, the operational working conditions for the bearings of the turbocompressor rotor and for engine pick-up are impaired.

The contradictory requirements imposed on lubricants for TVD [turboprop engines] in some cases are met by use of oil of medium viscosity, obtained by mixing oils with low (MK - 8) and high (M^{s} - 20 or MK - 22) viscosity. In other cases, two independent lubrication systems are used, each utilizing oils of different viscosity. In a lubrication system for a reduction gear, oil of high viscosity is used, and as the lubricant for the other points of the engine and the operation of the automatic-control system, low-viscosity oil is used.

A lubrication system for a TVD [turboprop engine] should insure reliable lubrication of the flexible couplings for any engine position in space, under any external conditions, at low oil circulation. The lubrication system must be airtight, convenient in use, and should exhibit low hydraulic resistance.

Closed circulation (one-circuit and two-circuit), open and combined lubrication systems are used in GTD [gas-turbine engines.

The single-circuit closed system, used in the case of moderate flight speeds and altitudes is especially simple in design geometry and economical in use. In this system the oil is repeatedly used for lubrication, and circulates in a closed circuit: reservoir to engine, to reservoir. The schematic layout of such a system as applied to a TRD [turbojet engine] with a centrifugal compressor is shown in Fig. 166. As can be seen from the figure, housing 5 of the lubrication system serves as the oil reservoir. The housing of the lubrication units is intensively ventilated by an airflow, and thus, in addition to its basic purpose, also serves as an oil radiator.

In engines with axial compressors the inlet section is often used as an oil reservoir-radiator, and the oil in this case is cooled by air flowing into the compressor. In powerful GTD [gas-turbine engines] with increased heat transfer to the oil, an oil reservoir 1 and radiator 19 are included in the lubrication system of the aircraft (Fig. 167).

Air and fuel-cooled oil radiators are used in engines. The latter are more economical because of the reduced flight drag and reduced heat loss, since the heat removed from the oil heats the fuel entering the combustion chamber. At the same time, in this arrangement the heating of the fuel removes the possibility of the freezing up of the fuel filters under low temperature conditions.

In one-shot engines operating at an augmented gas temperature, the open lubrication system may be used; the oil in such a system, after being used just once, is discarded into the atmosphere. The open system is very simple in design. The fuel [combustible] may be used as both lubricant and coolant in this system [27].

In high-temperature GTD [gas-turbine engines], the open system may be used for lubrication and cooling of the rubbing parts and bearings, working under conditions of high temperature (for example, turbine bearings), since under such conditions the use of the oil rapidly reduces its quality and it cannot be used too many times. For lubrication of the cooler parts, for reasons of economy, the ordinary closed system might be used. Such lubrication systems (Fig. 168) have been called combined systems.

All especially important and highly loaded bearings, gears, and splined couplings are lubricated under pressure, i.e., oil is fed continuously through jet or centrifugal nozzles under pressure developed

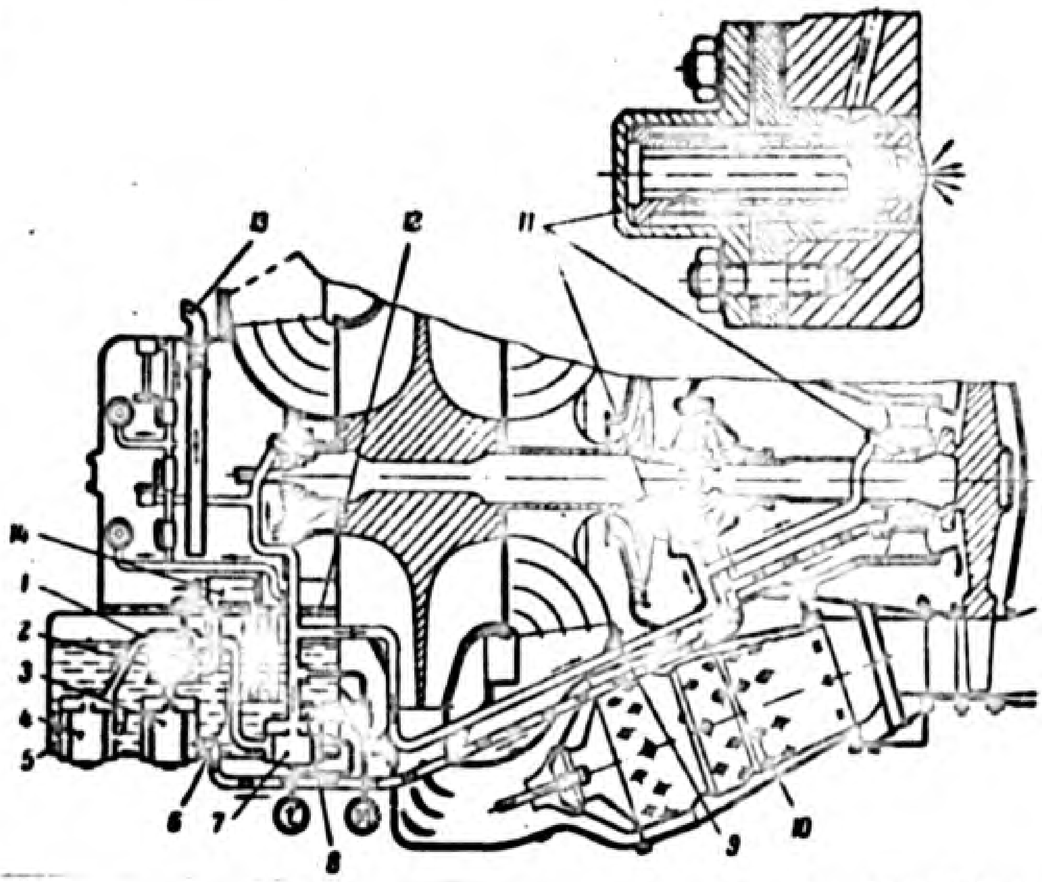


Fig. 166. Diagram of a single-circuit closed lubrication system of a TRD [turbojet engine] with centrifugal compressor. 1) pressure pump; 2) suction pump; 3) filter of the main suction line; 4) low-pressure filter of the main pressure line; 5) lubrication accessory housing; 6) carburetor jet; 7) high-pressure filter of the main pressure line; 8) reduction valve; 9) main pressure line; 10) main suction line; 11) spray nozzle injectors; 12) diaphragm (pen-type gas trap); 13) breather; 14) centrifugal gas separator.

by an oil pressure pump. Jet feed insures intense oil injection between rubbing parts, excellent heat removal, and flushing out of solid particles. Friction surfaces not pressure lubricated are lubricated by splashing oil on moving parts.

In order to prevent the saturation of the oil by gases, and also to decrease heating and decomposition, the stay time of the oil in the engine should be as small as possible. For this reason the GTD [gas-turbine engines] work on the principle of a dry housing. In such systems all the oil, after use in lubrication, is drained by pumps to the reservoir rather than being accumulated in sumps in the housing.

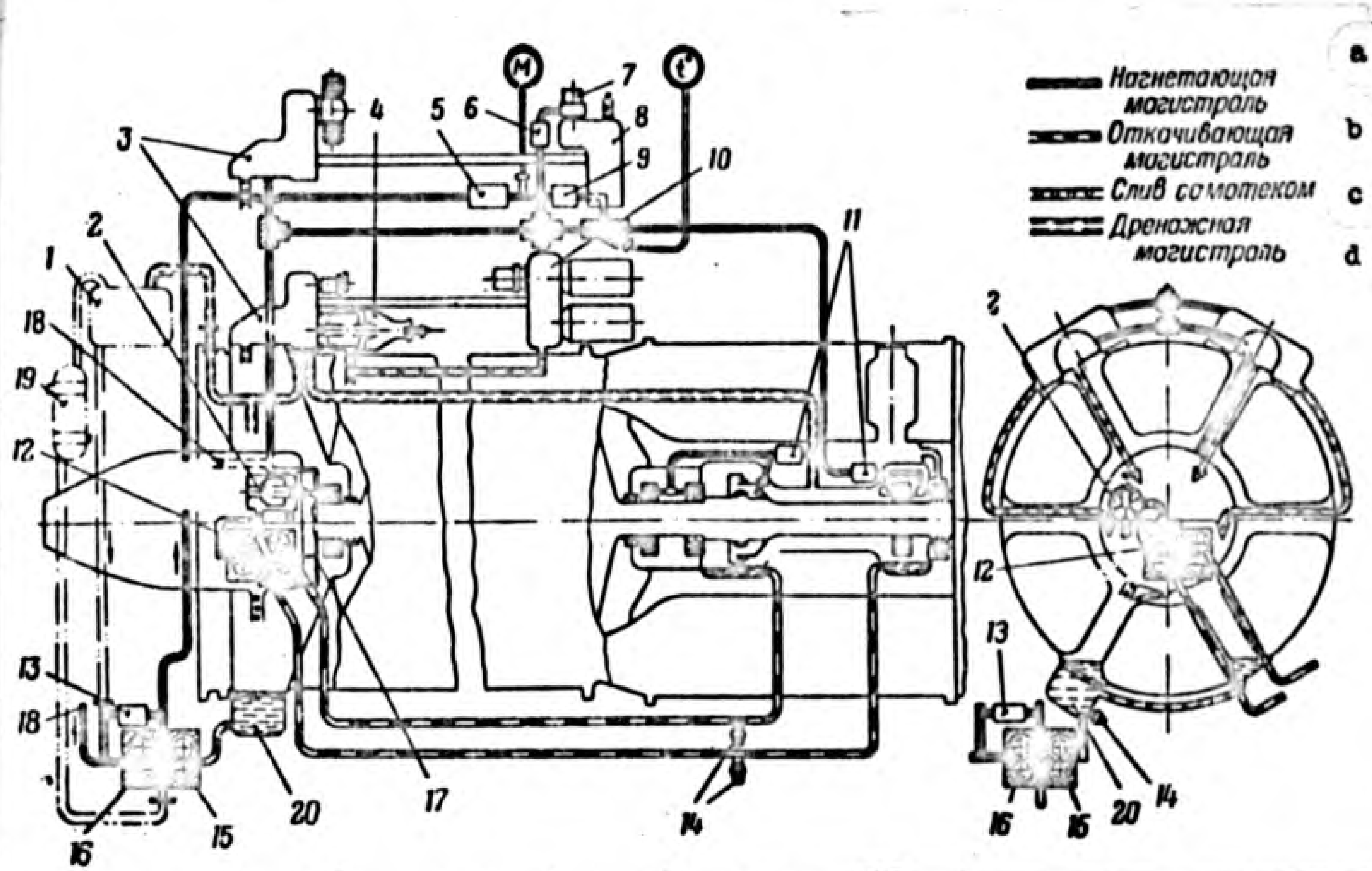


Fig. 167. Diagram of a single-circuit closed lubrication system of a TRD [turbojet engine] with an axial compressor. 1) Oil reservoir; 2) centrifugal gas separator; 3) intermediate drive compartments; 4) centrifugal breather; 5) filter; 6) auxiliary filter; 7) centrifugal sensor; 8) engine accessory drive compartment; 9) check valve; 10) aircraft accessory drive compartment; 11) auxiliary filters ahead of the spray nozzles; 12) oil booster pumps; 13) reduction valve; 14) overflow stopcock; 15) main suction pump; 16) pressure pump; 17) discharge from the main pressure line to the intake to the supercharge pumps; 18) discharge of oil to the turbostarter; 19) oil-fuel radiator; 20) oil sedimentation tank; a) pressure manifold; b) suction manifold; c) gravity drain; d) drainage manifold.

The following are the controlling parameters for lubrication systems:

oil pressure at engine inlet (3 - 5 kg/cm^2);

oil temperature at inlet (60 - 80° C);

oil temperature at engine outlet (110 - 150° C).

The pressure required in the pressure manifold is determined by the quantity of oil circulated through the engine, and by the hydraulic resistance of the manifold.

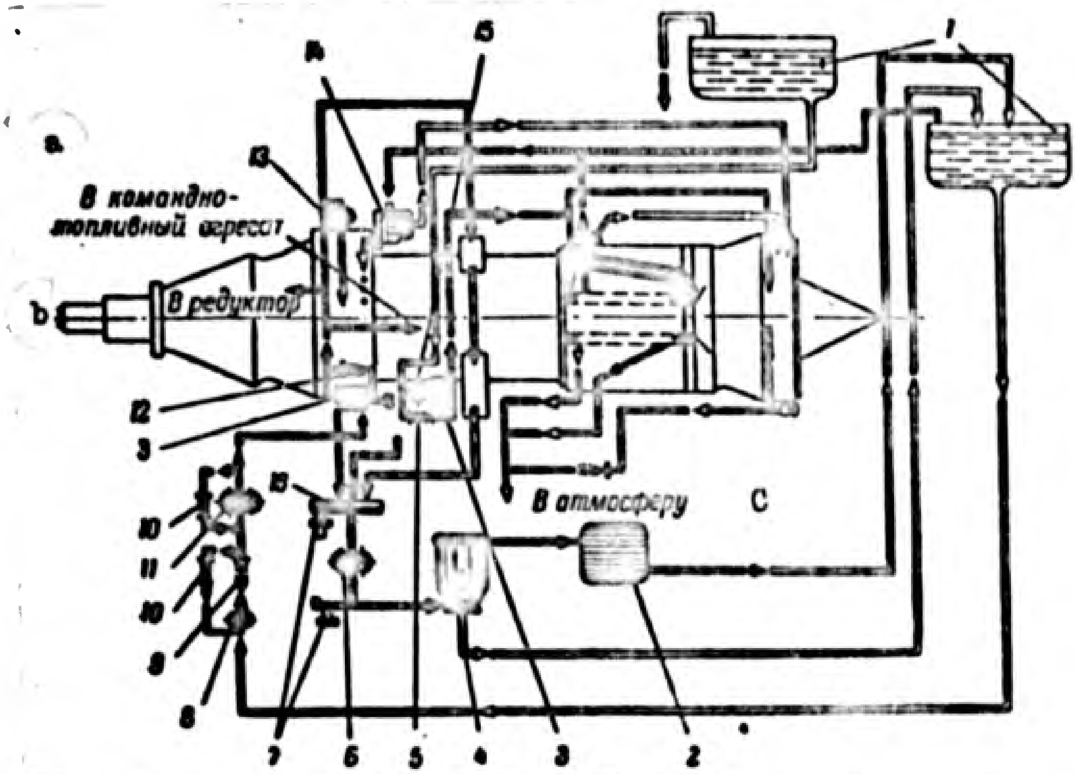


Fig. 168. Diagram of conbined lubrication system. 1) Oil reservoir; 2) oil-air radiator; 3) filters; 4) centrifugal gas separator; 5 and 10) reducing valves; 6) suction pump; 7) overflow stopcocks; 8) booster pump; 9) check valve; 11) pressure pump; 12) safety valve; 13) torquemeter pump; 14) centrifugal breather; 15) spare pressure pump; 16) oil sump; a) to the control fuel assembly; b) to the reduction gear; c) to the atmosphere.

2. REQUIRED OIL FEED. QUANTITY OF OIL IN THE LUBRICATION SYSTEM.

The required oil circulation through the engine is determined by the amount of heat removed by the oil. This heat is determined by addition: the heat developed by reason of mechanical losses in the bearings, accessory transmissions, and also in the reduction gear of the TVD [turboprop engine], and the heat transmitted to the lubrication points from the nearby heated parts by heat conduction.

The oil circulating through the engine includes the feed through the turbocompressor rotor bearings, the accessory drive, and through the reduction gear to the propeller (on TVD [turboprop engines). The quantity of oil circulation through the rotor bearings increases in relation to the rpm and loading: for compressors with roller bearings, 0.5 - 3.0 liters/min; for compressors with ball bearings, 4 - 10 liters/min; and for turbines with roller bearings, 3 - 10 liters/min.

The oil feed through the accessory drive compartments and the reduction gears is determined by the power transmitted, the mechanical efficiency of transmission, the heat capacity of the oil and its permissible heat rise upon flow through the engine. For modern GTD [gas-turbine engines] oil heating in the engine amounts to 40 - 60° C.

The over-all oil volume in the lubrication system is obtained by summing the volume of oil contained within the internal engine system and the oil consumed during operation, and also the volume of oil necessary for feeding the propellers and accessories, and the circulation reserve.

The term oil consumption refers to the irrevocable loss occuring during the course of the process of operating the GTD [gas-turbine engine] the loss consists of leakage through the labyrinth seals, discharge through the breather system, and also burning and decomposition due to overheating.

The circulation reserve depends on the time required for the completion of a complete circulation cycle for the entire oil supply through the GTD [gas-turbine engine] system, and this generally takes 0.5 to 1.0 minute.

The volume of the oil reservoir should be about 10 - 20% greater than the volume required to fill the lubrication system. This extra volume is necessary to permit expansion of the oil on heating and to make possible foam extinction in the reservoir, thus reducing the ejection of oil through the system of breathers in the reservoir.

3. DESIGN OF LUBRICATION SYSTEM ELEMENTS

The lubrication system in the general case consists of an oil reservoir, oil radiator, oil pumps (booster, pressure and suction), oil filters, outside and inside manifolds, gas separators, breathers, valves (reduction, shutoff, and safety), and also spray nozzles and monitoring equipment.

Pressure oil pumps are usually installed below the oil reservoir to insure the pressure level of the oil at the entrance into the pump, which promotes an increase in pump capacity. In most cases the pressure oil pump is installed in the lower part of the GTD [gas-turbine engine].

In order to insure the altitude capability of the lubrication system, the capacity of the pressure pump is set greater than the required oil feed rate through the engine by a factor of about 1.5 - 2.5. The excess in pump capacity compensates for the fall in that capacity with increased flight altitude.

The capacity of the suction oil pumps, as a rule, is greater than that of the pressure pumps because they handle foaming oil having high content of gas and vapor. The number of suction oil pumps depends on the type of engine, the number of oil settling chambers and their design. An engine with centrifugal compressor, having a small rotor length and oil cavities around the bearings, isolated from each other, has one suction pump 2 (see Fig. 166), mounted in housing 5 of the lubrication accessories.

In multisupport GTD [gas-turbine engines] with axial compressors having large rotor length, the design is provided with several oil collectors which are connected by a common chamber (the rear compressor bearing and thr turbine bearing). In this case the number of suction pumps and oil collectors is the same. The size of each suction pump is chosen from the condition of insuring the possibility of pumping out all the oil collected in only one of the oil collectors during a prolonged dive or climb.

The over-all capacity of the suction pumps in such engines is usually 2 - 3 times greater than the capacity of the pressure pump.

It should be noticed that an insufficient margin in suction-pump capacity may lead to loss of oil from the reservoir in the engine, and in the case of an excessive suction-pump capacity margin, they will suck out a large quantity of air with the oil, and deliver it to the reservoir, which leads to emulsification (foaming) of the oil. To prevent this phenomenon, in some GTD [gas-turbine engines] the oil from the various oil collectors is pumped by various oil pumps into one common oil settling chamber 20 (see Fig. 167) from which the oil is pumped into the reservoir by one main suction pump with small capacity margin, equal to 1.5 - 2.0.

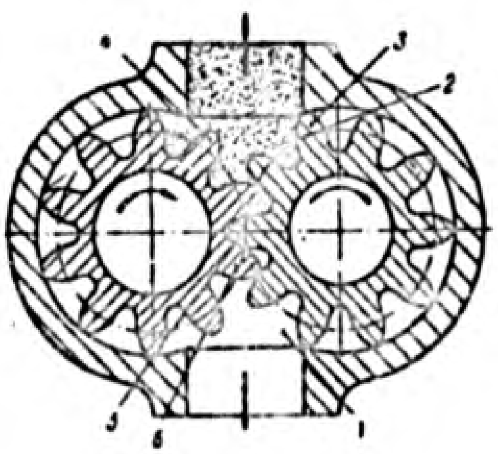


Fig. 169. Diagramatic sketch of gear-pump operation. 1) Suction chamber; 2) pressure chamber; 3, 4, 5 and 6) teeth.

Suction pumps are located as near as possible to the oil collectors or directly in them, in order to decrease the resistance at the inlet and to improve the pumping. In case there is a large distance between the pumps and the oil collectors, oil is sometimes fed from the pressure line to the outlet pipe of the suction pumps in order to improve their draining capability (see Fig. 167). The supply oil removes the air locks from the cavity of the pump unit and facilitates the creation of a good vacuum at the pump inlet. An analogous effect is obtained in cases where oil is fed from the pressure line to the inlet of the suction pump.

In GTD [gas-turbine engine] designs it is primarily gear pumps that are used, and these are outstanding in simplicity of design, small size and weight, and also reliability and long service life.

Figure 169 shows a diagram of the operation of a gear pump with externally engaged gears, having the same number of teeth. Given the direction of gear rotation as shown, the oil from suction chamber 1 is carried in the spaces between the teeth into pressure chamber 2. On the side of the oil outlet from the pump, teeth 3 and 4 mesh and squeeze the oil from the spaces, and on the suction side, teeth 5 and 6 come out of mesh; finally, the oil is sucked into the free space. Part of the oil, remaining in the clearances between teeth, when they are engaged, is carried around again into the suction chamber.

The capacity of a gear pump depends on the geometric dimensions, the rpm, and the delivery coefficient.

The delivery coefficient of the pump accounts for the decrease in actual oil delivery relative to the delivery calculated in conformity with the geometric dimensions of the gears. This coefficient depends on the rotational speed of the gears, the inlet pressure, recirculation through the clearances, the viscosity, and the volume of the oil carried back by the teeth into the suction chamber.

For the oil pumps used on engines, the delivery coefficient amounts to 0.8 - 0.9, and its decrease with increasing altitude is compensated by the fact that the capacity of the installed pumps is ordinarily greater by a factor of 1.5 - 2.5 than the required circulation through the engine, as has been stated above.

In the locking of oil in the tooth cavities, which occurs during meshing, considerable loads may be imposed on the pump bearings.

In order to avoid this, usually there are relief slots 1 and 2 in the pump housing (Fig. 170, a) or in the nonworking sides of the teeth (Fig. 170, b).

The slots are usually made with an outlet on the delivery side, since an outlet on the suction side decreases the delivery coefficient (up to 6 - 8%).

A reduction valve is intended to maintain the given oil pressure in the pressure manifold of a GTD [gas-turbine engine] which is set in the limits 3.5 to 5 kg/cm^2.

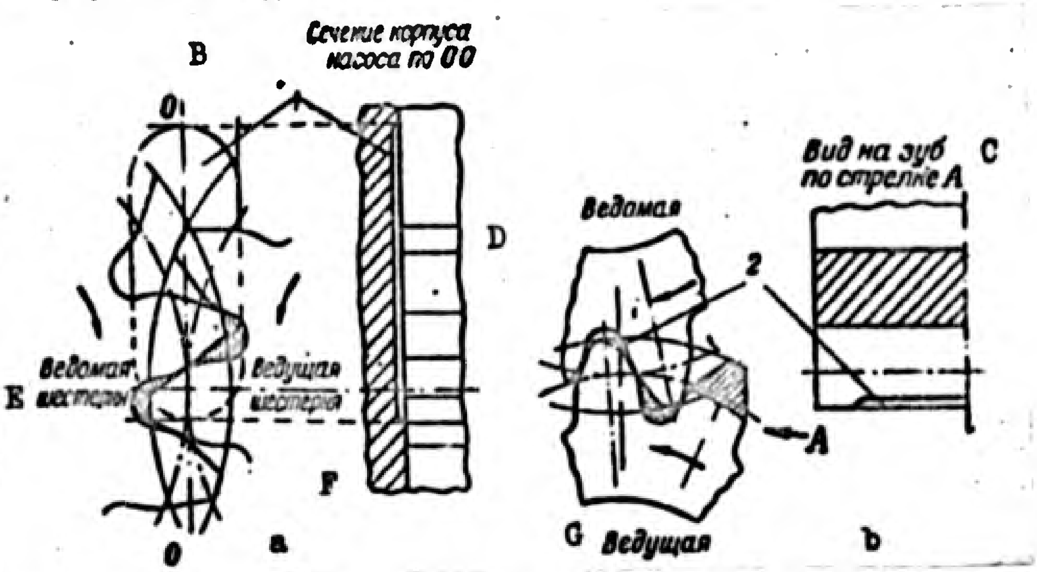


Fig. 170. Relief grooves. a) In the pump housing; b) on the nonworking sides of the teeth; 1 and 2) grooves; B) cross-section of pump housing along 00; C) view toward tooth along arrow A; D) driven; E) driven gear; F) driving gear; G) driving.

Figure 171 shows a diagramatic sketch showing the manner in which a reducing valve is incorporated in to the manifold and design to maintain pressure by bypassing an excess quantity of oil to the suction side of the pump.

Disk valve 2 is pressed against seat 1 by spring 3; the initial pressure of the spring is set by screw 4.

The quantity of oil passing through the reducing valve is determined by the difference between the capacity of the pressure pump and the oil feed through the engine in the given regime.

With an increase the quantity of oil passing through the valve, the flow passage of the valve is increased by its greater lift. In this connection, in order to maintain the valve in the open position, greater oil pressure behind the pump is necessary, since raising the valve increases the tension force of the spring. The stiffer the spring, and the greater the lift of the valve in increasing the recirculation of theoil, the higher the pressure developed by the pump.

The dependence of the quantity of oil passing through the reducing valve (flowing into the engine), and also of the pressure of the oil at the pump outlet on the rpm is shown in Fig. 172, a. If the pump is not turning, its capacity and the oil pressure at the outlet are equal to zero. In proportion to the increase in rpm, the pumping capacity (curve 1) and the pressure (curve 3) increase. Upon reaching some rotational speed n_1, the pressure reaches the magnitude for which the reducing valve is set. After this the valve begins to open, and the oil pressure rises very little because of the increase in the lift of the valve. Only part of the oil flows in the pressure line of the engine (curve 2), and an ever-increasing quantity of oil begins to flow through the reducing valve (the region cross-hatched with vertical lines). In a given case at a nominal rotational speed of the rotor, about half of the quantity flowing through the pump is recirculated through the reducing valve, which is determined by the given margin in its capacity.

Upon ascent to an altitude where because of the decrease in delivery coefficient the output of the oil pump decreases (curve 1, Fig. 172, b), the quantity of oil recirculated through the reducing valve (in Fig. 172, b cross-hatched by vertical lines) decreases. In this case, the pressure after the pump (curve 3) decreases somewhat. Upon reaching some altitude H_{kr}, called the critical altitude, the delivery of the pump falls enough that all the oil is directed into the engine with the valve closed. Beginning with this height, the reducing valve ramains completely colsed, and the oil pressure and its circulation through the engine (curve 2) begin to decrease at a faster rate than it had on approaching the critical altitude.

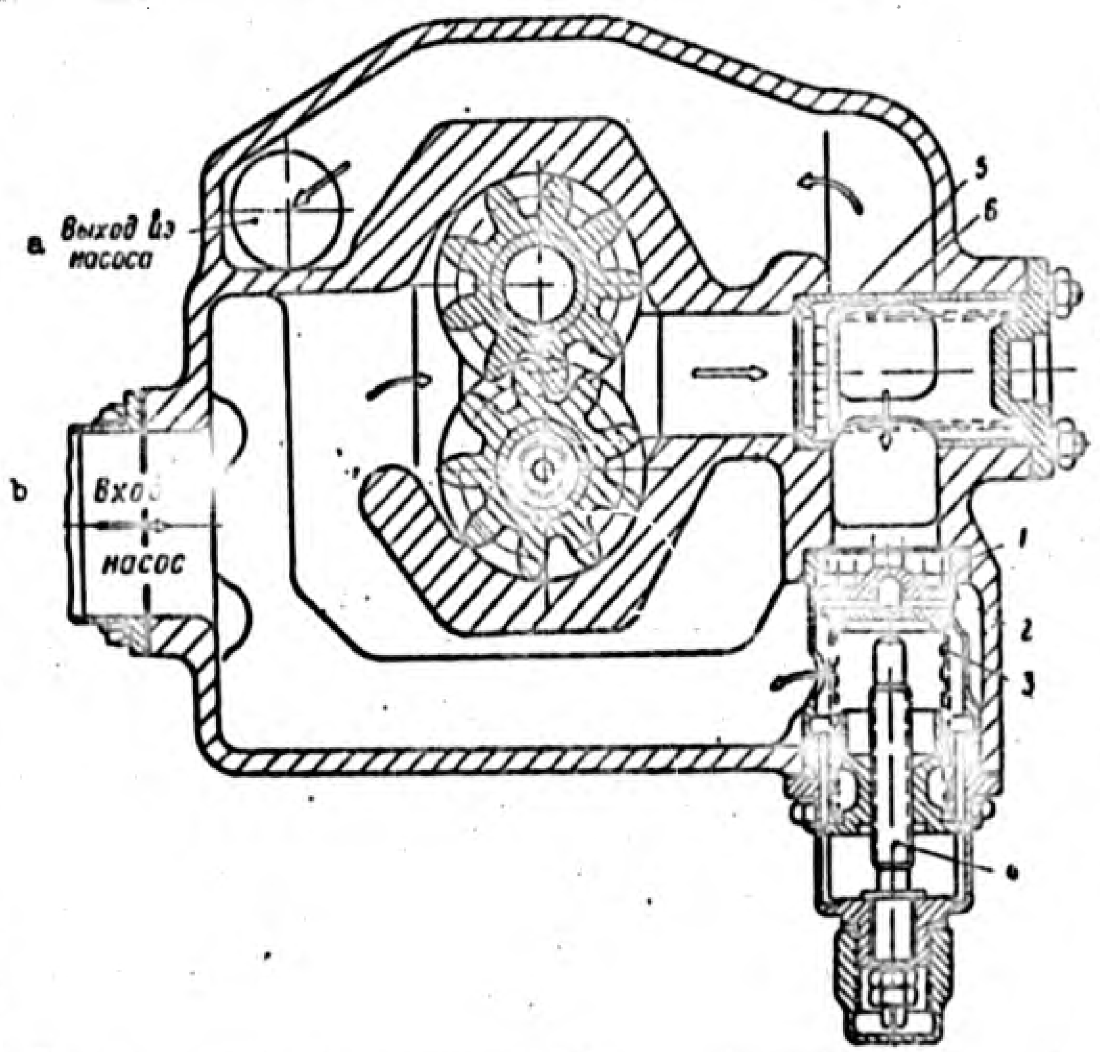


Fig. 171. Diagram showing incorporation of reducing and check valves in the manifold and design. 1) Seat of the reducing valve; 2) disk valve; 3) spring; 4) regulating screw; 5) check valve; 6) spring; a) outlet from pump; b) inlet to pump.

With an increase in the capacity margin of the pressure oil pump, the altitude-capability of the lubrication system of the GTD [gas-turbine engine] (see curves 1', 2', 3') increases.

The work of the reducing valve and the oil pressure at the pump outlet depend on the oil temperature. With an increase in oil temperature, its viscosity decreases and the circulation of oil through the engine increases. Decrease in recirculation of oil through the valve leads in turn to a drop in oil pressure. In cold oil, conversely,

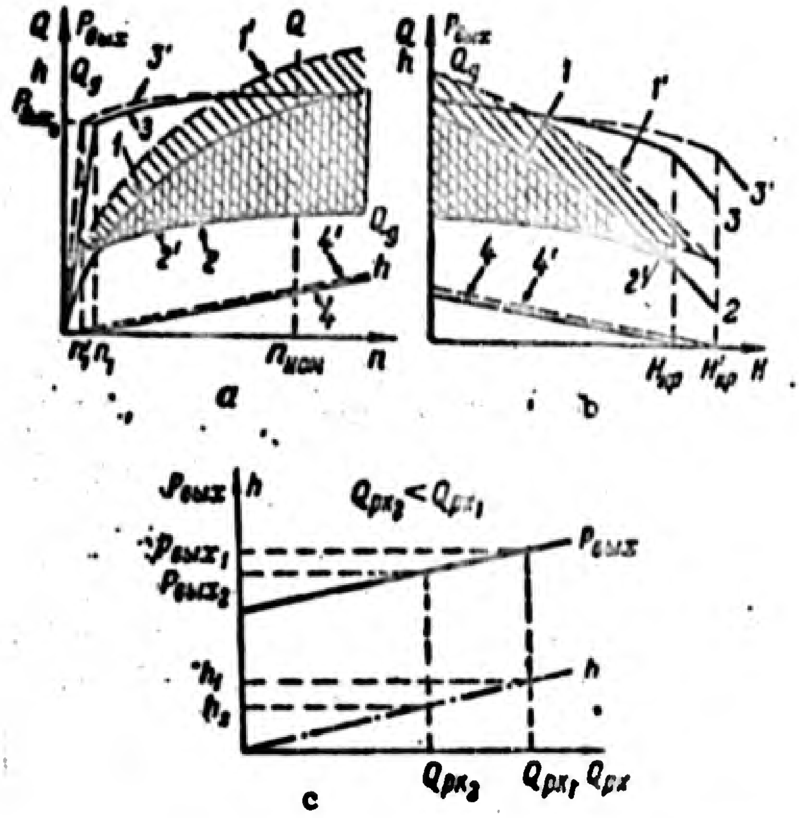


Fig. 172. For an analysis of the operation of a reducing valve (parameters with primes refer to increased pump capacity. a) Dependence of pump capacity (curve 1), oil circulation through the engine (curve 2), pressure at outlet of pressure pump (curve 3) and amount of reducing-valve lift (curve 4) on rpm $\underline{n}$ of GTD [gas-turbine engine] rotor; b) dependence of same parameters on flight altitude H; c) dependence of height $\underline{h}$ of valve lift and pressure P_{vykh} at pump outlet on quantity of oil Q_{rk} recirculated by reducing valve.

recirculation through the reducing valve and pressure increase (Fig. 172, c).

On installation of an oil filter 5 (see Fig. 167) after the pressure pump, when reducing valve 13 maintains the pressure constant after pump 16, any change in the resistance of the filter, e.g., when it is clogged, will be reflected in the oil circulation to the engine. In systems in which reducing valve 8 (see Fig. 166) is installed after the high pressure filter 7, change in the resistance of the filter has no effect on the oil feed to the engine.

A shutoff valve 5 is installed on the side of the oil outlet from the pump (see Fig. 171) or of a high-pressure filter (Fig. 173) to prevent oil backflow from the reservoir into the oil cavities of the engine. This backflow occurs in an engine that is shut down, under the pressure of the oil level in the reservoir (when the level at which the reservoir is installed is higher than the level at which the pump is installed). Spring 6 of the check valve is set at a pressure of 0.2 - 0.5 kg/cm^2, exceeding the pressure of the oil level at the inlet to the oil pump. During operation of the engine, the shutoff valve is opened under the pressure developed by the pump, and all the oil flows through the pump into the engine system.

Oil filters are intended to rid the oil of solid particles produced by coking and the decomposition of the oil, wear of parts and their corrosion, and also of the solid particles which enter into the lubrication system from the atmosphere during engine operation.

Filters may be located both in the pressure line, and also in the suction line. In the pressure line, filters may be located either ahead of the pump (low-pressure filter 4, see Fig. 166), or behind the pump (high-pressure filter 7). In the suction line, filters are also located either ahead of or behind the pump.

When filters are installed in the suction line, solid particles in the oil are collected directly upon outlet from the engine, which prevents clogging the oil radiator, the reservoir, and the pressure oil pump. However, when the filter is installed in that manner, the required oil pressure at the outlet from the suction pump is increased.

Less widely used in GTD [gas-turbine engines] are the plate (slit) filters (Fig. 173, a) and most widely used are the assembled filters, consisting of grid filtering elements (Fig. 173, b), which

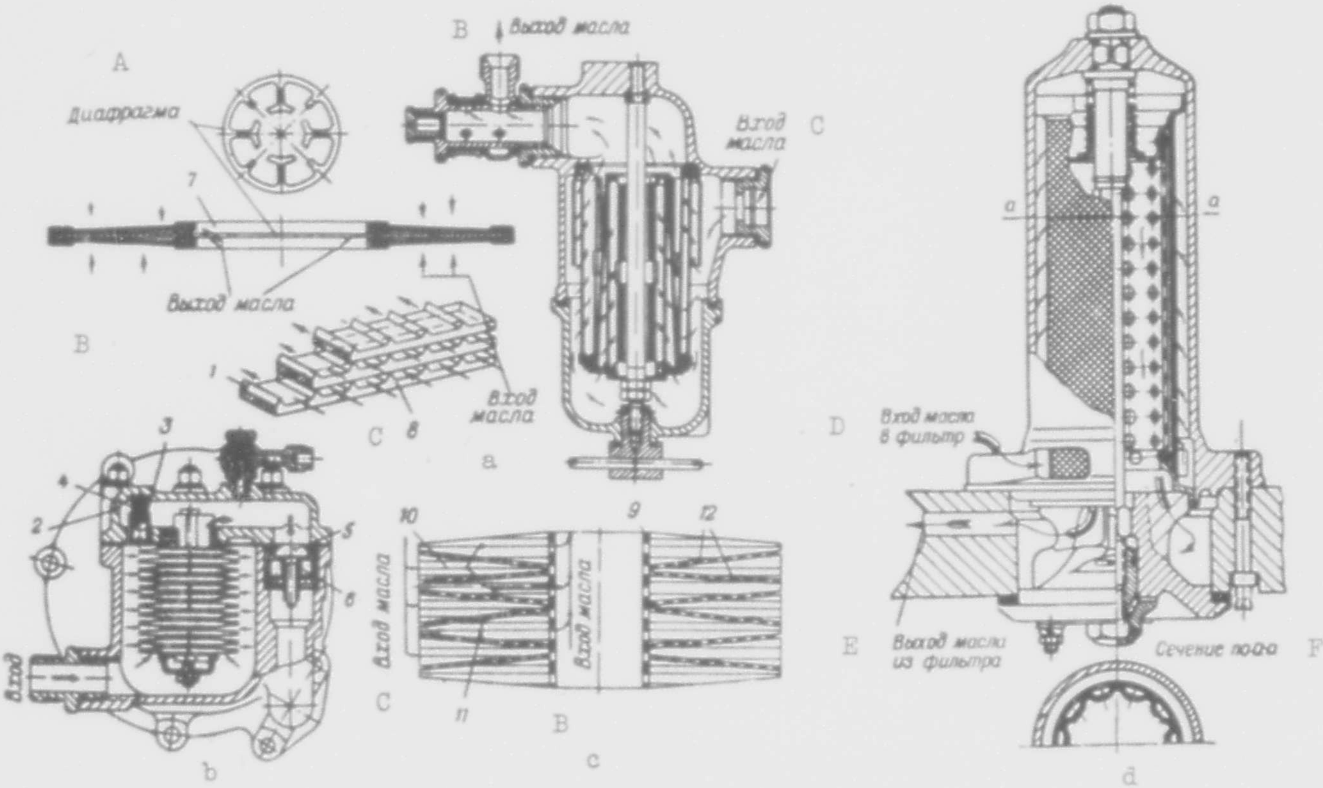


Fig. 173. Design of filters. a) Plate (slit) type; b) grid filter-element assembly; c) disk type; d) cylindrical grid type; 1) projection on wire surface; 2) cover; 3) safety valve; 4) spring; 5) shutoff valve; 6) shutoff valve spring; 7) grid filtering element of the assembled filter; 8) filtering element of the plate filter; 9) perforated stem; 10 and 11) inlet and outlet corrugated separators; 12) grid; A) diaphragm; B) oil outlet; C) oil inlet; D) oil inlet to filter; E) oil outlet from filter; F) section along aa.

results from the effort to increase the interval between inspection of filters. The time between inspection of the assembled filters is substantially greater: in comparison with the simple grid cylindrical filters (see Fig. 173, d), 3 - 5 times longer; in comparison with the disk type (Fig. 173, c), about 1.5 - 2 times as long. This is explained by the fact that the useful area of the assembled filter, for the same dimension, is greater by a factor of about 7 than the cylindrical and greater by a factor of about 1.5 than the disk grid filter.

The filtering elements of assembled grid filters are fabricated of brass or steel wire with a mesh of 225 to 5000 per cm^2. The supporting grids have a mesh of 30 - 40 per cm^2.

The slit filters are fabricated of wire of a special crosssection; the slits are formed by projections 1 situated on the surface of the wire.

To prevent the failure of the filtering grid in case it becomes clogged, the filters are generally fitted with safety valves 3 (Fig. 173, b) providing a bypass for the oil around the filter. Spring 4 of the safety valve creates tension corresponding to the opening pressure of 1.3 - 1.5 kg/cm^2.

Filters are mounted and installed in the system so that they can be removed easily for cleaning, without draining the oil from the system and without removing other accessories. For example, to remove the filter shown in Fig. 173, b it is enough to remove only cover 2.

Gas separators are installed for purification of the oil from gases. The less the oil is saturated with gases, the less it foams, the more effective its cooling in the radiator, and the more reliable the lubrication and the high-altitude capability of the system.

The simplest gas separator is a trough; it is installed in the oil reservoir (see 12 in Fig. 166) or in the oil sump (see 16 in Fif. 168). When a thin layer of foamy oil enters the trough, the bubbles burst because of surface tension and the gases are released from the oil. Such gas separators are used in the case of limited oil circulation and negligible gas content in the oil.

For great oil circulation and considerable gas content in the oil, the centrifugal gas separators are used.

The design of a centrifugal gas separator is shown in Fig. 174. Rotor 2, consisting of a cylinder with radial partitions, is spun at a speed of 30 - 40 m/sec. Under the action of centrifugal forces, the oil is thrown to the periphery and is drained off into the radiator (reservoir). Gases separated out near the rotational axis are released to the atmosphere through the hollow center of the shaft 3, through a system of breathers.

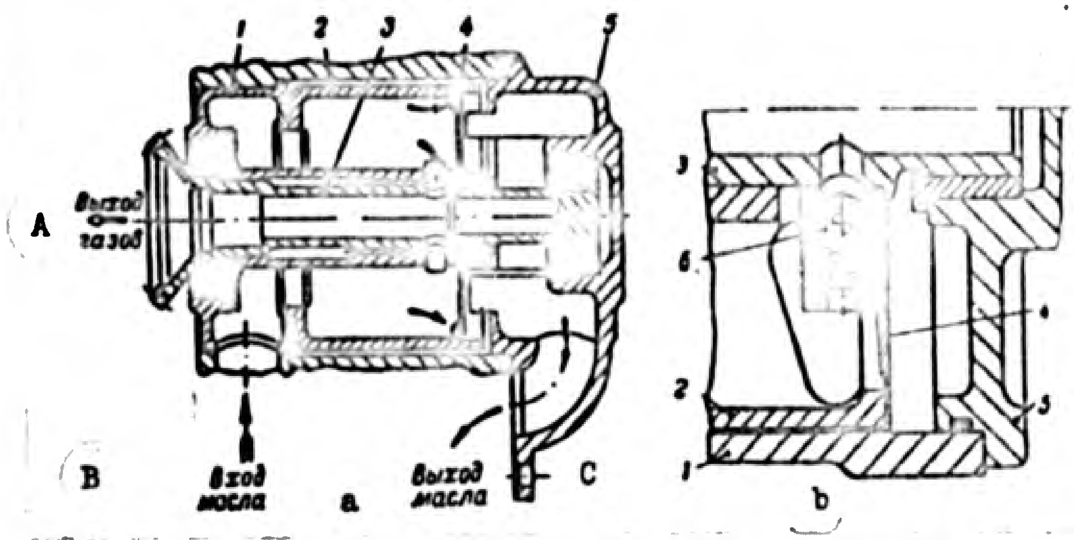


Fig. 174. Design of a centrifugal gas separator (on the right, the safety-valve unit). 1) Bearing cup; 2) rotor; 3) shaft; 4) diaphragm; 5) cover; 6) safety valve; A) gas outlet; B) oil inlet; C) oil outlet.

In order to prevent the induction of unheated oil (on GTD [gas-turbine engine] startup) safety valves 6 (Fig. 174, b) are installed in the system of breathers through apertures for gas exhaust, because of the great resistance at the outlet from the gas separator. The valves are opened to permit the escape of the gases under the action of the centrifugal forces of the rollers, upon reaching a preset rotational speed.

To avoid an excessive pressure increase in the lubrication cavities of the housings and the oil reservoir due to heating, oil vaporization, and the bursting of the gases through the labyrinth seals, the inner cavities of the housings, and the reservoir are connected with the atmosphere through a system of breathers. In this system of breathers, gas motion always proceeds from the inner cavities to the atmosphere. Because of the vaporization and atomization of the oil upon rotation of the parts, saturation of the air cavities by vapors and droplets of oil takes place, and this leads to an increase in discharge [flow] through the breather system. In order to decrease the discharge of oil into the atmosphere, centrifugal breathers (Fig. 175) are installed in the breather system, and these in design and principle of operation are analogous to centrifugal gas separators. The channels and ducts of the suction manifold should exhibit low hydraulic resistance. This is achieved by selecting the appropriate channel diameters for the condition that the speed of the oil should be maintained within the limits 0.3 - 1.0 m/sec and by a possible decrease in the length of the suction manifold.

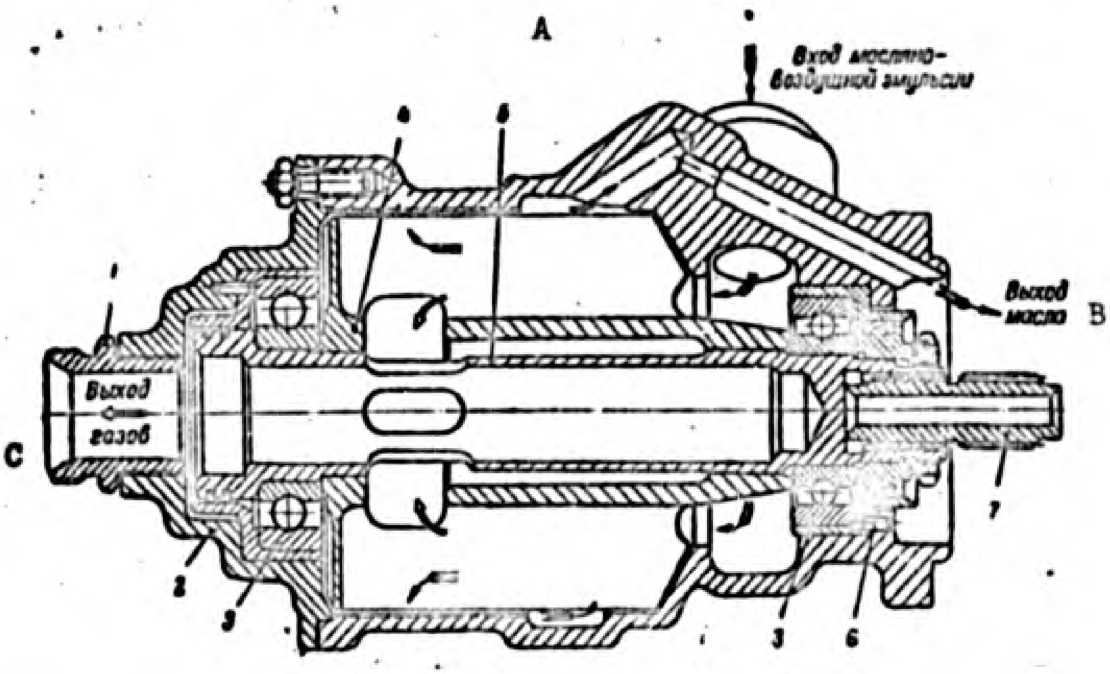


Fig. 175. Design of a centrifugal breather. 1) Connecting pipe for discharge of gases; 2) cover; 3) bearing; 4) rotor; 5) shaft; 6) bushing of the ring-holder; 7) spring gear drive; A) oil-air emulsion inlet; B) oil outlet; C) gas outlet.

The speed of oil flow in the pressure ducts is set higher, in the limits 1.5 - 3 m/sec. The drainage ducts have diameters of 15 - 30 mm. Oil ducts are made of steel or aluminum alloys. In order to eliminate vibration of the ducts, they are held fast by yokes and special clamps (Fig. 176), installed at intervals of 30 - 40 outside tube diameters.

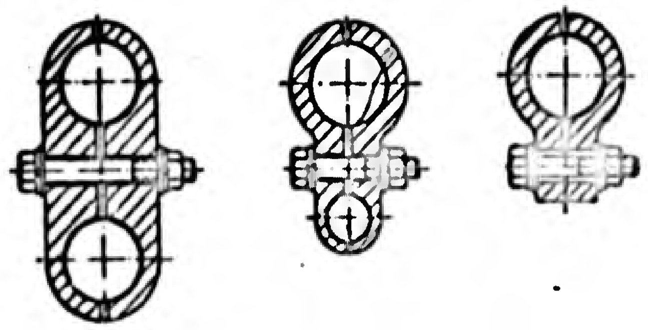


Fig. 176. Types of fasteners for attachment of oil ducts.

The connection of the ducts is shown in Fig. 177. In order to avoid thermal stresses either flexible connections are provided (Fig. 177, a), or they are provided with appropriate bends. Sealing in flexible connections is insured by use of rubber rings.

Oil feed to the bearings is accomplished by jet and centrifugal nozzles, and also through holes located in rings (see Figs. 159, d and e; Fig. 160, g). Centrifugal nozzles 11 (see Fig. 166) atomize the oil, and this impairs heat removal from the bearing and increases the saturation of the oil with gases. In case of the use of jet nozzles, the nozzle axis is set at an angle of 10 - 20° to the shaft axis and in such a manner that the oil shall impinge on the clearance between the inside ring and the bearing separator. With the jet nozzle set up in this manner, the amount of oil getting through between the rollers (balls) to the other side of the bearing without striking parts of the bearing is reduced.

In order to avoid heating and foaming of the oil upon impact with the lubricated surface, its speed at the nozzle outlet should not exceed 20 - 25 m/sec.

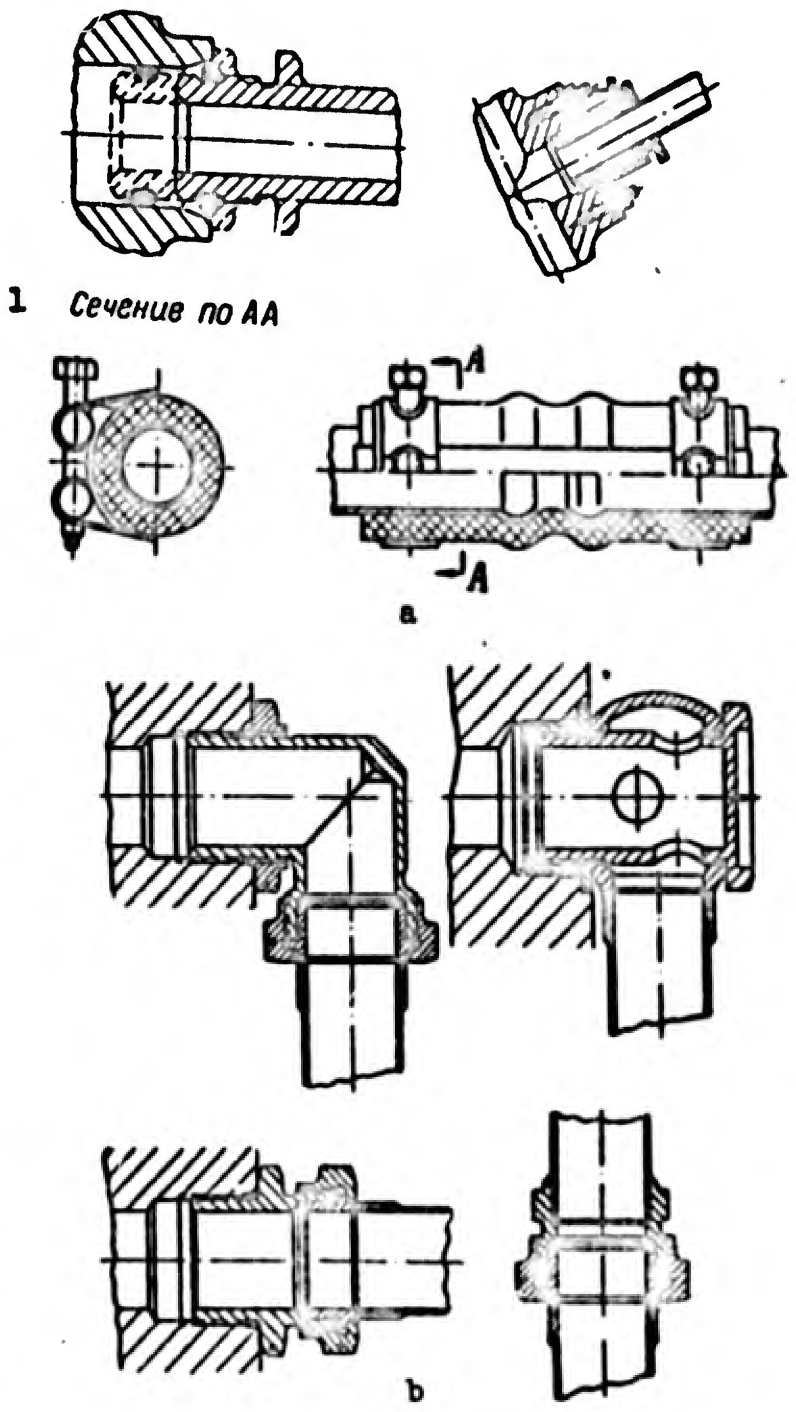


Fig. 177. Typical connections of oil lines. a) Flexible coupling; b) rigid coupling; 1) section through AA.

Part six

PROPELLER HUBS AND REDUCTION GEARS OF TVD [TURBOPROP ENGINES]

Chapter 13

HUBS OF VARIABLE PITCH PROPELLERS (VISh)

Thrust in TVD [turboprop engines is generated mainly by the propeller, driven from a common or separate turbine of the engine. The propeller consists of blades and a hub. The blades of a variable-pitch propeller may be rotated freely about their axes by use of a special mechanism for changing the propeller pitch. Change in propeller pitch is necessary for changing the power absorbed by the propeller, and for regulating the rpm. Increase in propeller pitch, that is increase in the setting angle of the blades, leads to an increase in the power absorbed by the propeller, and vice versa.

1. LOADS ACTING ON THE BLADES AND HUB OF A PROPELLER. ARRANGEMENTS OF MECHANISMS FOR CHANGING PROPELLER PITCH.

As a propeller turns, each element of the blade is subject to an aerodynamic and centrifugal force (Fig. 178), P_{aer} and P_{tsb}. The aerodynamic forces acting on the propeller blade elements cause blade bending, and are transmitted through the mounting to the propeller hub. The sum of the axial components of the aerodynamic forces $P_{aer.os}$ of all the blades constitutes the thrust P of the propeller, and the sum of the moments due to the circumferential components of these forces $P_{aer.\ okr}$ relative to the rotational axis of the propeller constitutes the reaction moment of the propeller, equal to the torque

M_t of the turbine, transmitted to the propeller shaft from the engine. The centrifugal forces cause tensile stresses in the blades and are transmitted through the mounting to the propeller hub, where they are added vectorially and cancel out. Besides this, the centrifugal forces of the blades create a twisting moment, which tends to twist the blade so as to decrease the pitch of a variable-pitch propeller [VISh]. This moment is absorbed by the mechanism for changing the propeller pitch. Besides the loads listed for the case of a rotating propeller, there are also the unbalanced centrifugal forces due to propeller disbalance, and the gyroscopic moment of the propeller, due to aircraft maneuvers, when the rotational axis of the propeller is turned in space. The gyroscopic moment bends the propeller shaft.

In modern aviation engines variable-pitch propellers are most widely used with a hydraulic mechanism for blade rotation. In the pitch-change system of a hydraulic VISh there is always a pump and hydraulic motor (servomotor). The pump is most often of the gear type and is usually combined with a propeller speed regulator in a single assembly. The hydraulic motor turning the blades is located directly in the propeller hub.

The hydraulic motors (servomotors) used are either of the piston type, or geared type. The use of a piston servomotor requires an additional apparatus for transforming the translational motion of the piston to the rotational motion of the blades. Such an apparatus us usually a crank-and-link (parts 3, 4 and 5 in Fig. 179, a) or crankgear (parts 5 and 8 in Fig. 179, c) mechanism.

Hydraulic servomotors may be of single or double-sides action.

If a single-sided servomotor is present in a VISh system, the propeller may have either a direct or a reverse control circuit.

A propeller for which fine pitch is achieved by the centrifugal

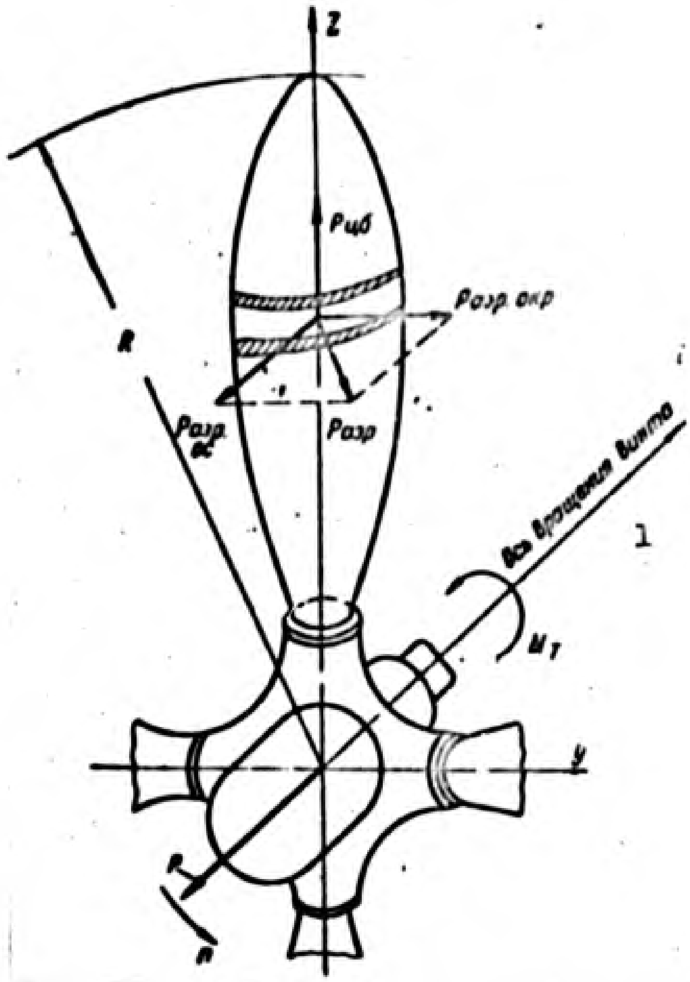


Fig. 178. Diagram of the forces and moments acting on a propeller blade and hub. 1) Propeller rotational axis.

forces of the blades and coarse pitch by a force developed in the hydraulic servomotor is known as a reverse control-circuit propeller. A diagram of the mechanism used to change the setting of blades in such a propeller is shown in Fig. 179, a.

A propeller with a direct control circuit is the name applied to a propeller in which the setting of the blades to fine pitch is accomplished by means of a hydraulic servomotor, and for coarse pitch, by the centrifugal forces of special counterweights carried on the butt end of each blade. A diagram of the mechanism for changing the setting of the blades in such a propeller is shown in Fig. 179, b.

In a propeller with a double-sided servomotor the setting of the blades for fine and coarse pitch is accomplished by means of the hydraulic servomotor. A diagram of the mechanism for setting the blades of the propeller with double-sides action is shown in Fig. 179, c.

In case the hydraulic system of a reverse-circuit propeller is not in good working order, the centrifugal forces of the blades always set them at ground [minimum] pitch, determined by the position of the minimum pitch stop. Excessive unloading of the propeller (setting at too fine a pitch) may cause overspeeding of the engine and revolutions in excess of the maximum permissible rpm. Furthermore, a greatly underloaded propeller develops thrust insufficient to maintain flight.

A propeller with a direct circuit does not have this deficiency. In case the hydraulic system is out of order, in the direct-circuit propeller the centrifugal forces of the counterweights set the blades in the position of maximum pitch, determined by the position of the coarse pitch stop. The power required by the propeller at the given rpm exceeds the power developed by the engine, and the engine slows down. In this case, however, a substantial decrease in thruse may occur, accompanied in the single-shaft TVD [turboprop engine] by an increase in the temperature of the gases ahead of the turbine. Among the shortcomings of a direct-circuit propeller is its increased weight due to the presence of heavy counterweights.

Double-sided propellers are outstanding because of their extreme reliability. Inasmuch as when the propeller is set for fine pitch, the moment developed by the servomechanism is always reinforced by the moment of the centrifugal forces from the blades, the oil pressure may be lower than when the setting is for coarse pitch, when the moment from the centrifugal forces counteracts the servomechanism. To set the propeller for fine pitch, oil is often used from the main

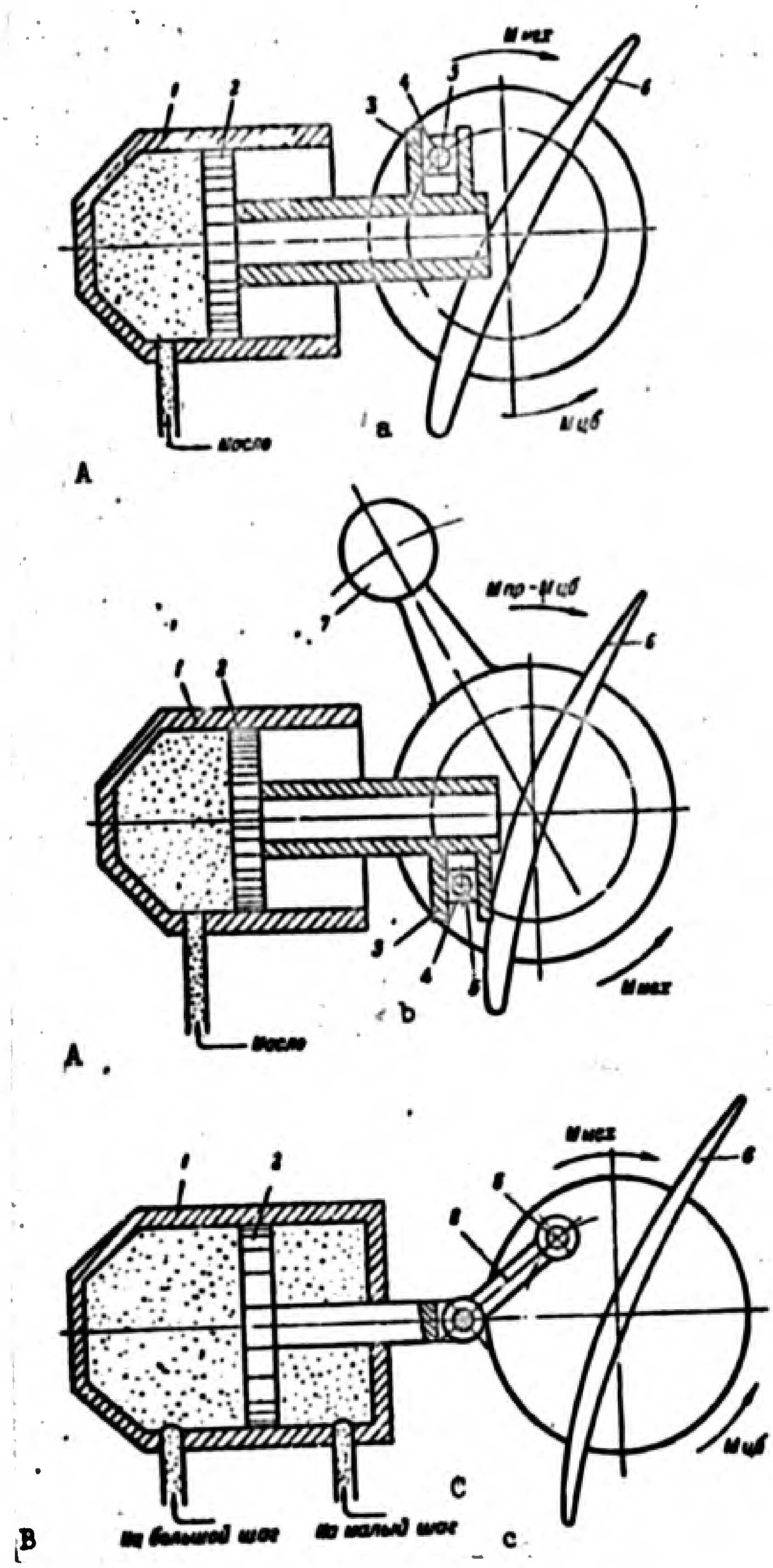


Fig. 179. Diagrams of variable-pitch propellers. a) Propeller with reverse circuit; b) propeller with direct circuit; c) propeller with double-sided servomotor; 1) cylinder; 2) piston; 3) cross-arm; 4) "biscuit"; 5) blade pin; 6) blade; 7) counterweight; 8) connecting rod; A) oil; B) to coarse pitch; C) to fine pitch.

engine line at a pressure of 6 - 8 kg/cm^2, which is fed constantly into one of the servomotor cavities and acts on one side of the piston. However, for setting the propeller for coarse pitch in this case, oil from a special pump at a pressure of 20 - 50 kg/cm^2 is used. Such pressure on one side of the piston is quite sufficient to overcome the forces acting on the other side of the piston and consisting of the sum of the forces arising by action of the oil pressure from the main oil line, and to overcome the forces due to the moments of the centrifugal forces of the blades.

To prevent setting the propeller at fine pitch in case the oil pressure falls, the system often includes a hydraulic locator for fixing the blade angle. This locator generally consists of an oil-pressure operated valve. In case the oil pressure falls, the valve closes under the action of a spring, and closing the servomotor cavity, fixes the propeller blades at the chosen setting angle. The centrifugal forces of the blades are not able to turn the propeller to fine pitch, since the oil outlet from the servomotor is closed. Such a device may be used also on propellers with reverse circuits.

Propellers with hydraulic control are outstanding in simplicity of design, compactness of mechanism, convenience in control and regulation, and also because of high reliability in operation. Such advantages of the hydraulic propellers in comparison with other types results in especially extensive use of propellers with hydraulic controls.

2. DESIGN OF PROPELLER HUBS.

The propeller hubs sustains substantial loads, so that it should exhibit adequate strength and rigidity. For this purpose, the hub housing is made of alloy steel and has a complex configuration. The design of the attachment of blades to the hub should insure free rotation of the blades with a mechanism for changing the propeller pitch. Roller bearings are usually used in the attachment of blades to the hub. The mechanism for changing the propeller pitch should develop sufficient force to turn the blades, at a speed of 3 - 7 degrees per second. A propeller on a multi-engine aircraft, further, should also permit feathering of the blades to decrease the aerodynamic resistance of the propeller in case one of the engines fails in flight or is shut down at the pilot's discretion. To insure reliability of propeller operation at low temperatures of the surrounding medium, many designs provide for the in-flight heating of the hub.

Propeller blades are usually attached to the hub by means of a threaded key. In this case (Fig. 180) the centrifugal forces of blade 1 is transmitted through thread 2 to cup 3, and through the supporting roller bearing 4 to housing 5 of the hub. The force of thrust, the circumferential force, and the bending moment of the blade are also transmitted to cup 3 and through support bearings 6 to housing 5. Collar 7 insures reliable retention of the blade in the cup and transmission of the torque from the blade to the cup due to the friction between them. Hubs with threaded blade attachment ensure rapid replacement of blades and portability of the propeller in the partially dismantled state.

Figure 181 shows a structural diagram of a VISh with threaded attachment of blade 9 to cup 8. The cup is connected with housing 7 of the propeller hub by use of three rings of ball support-thrust bearings. The bearings do not have rings, the races for the balls being made directly on the surface of cup 8 and housing 7. All the forces from the propeller blade are transmitted to the housing through these bearings. Hub housings for propellers may be split (see Fig. 180) or unsplit (see Fig. 181).

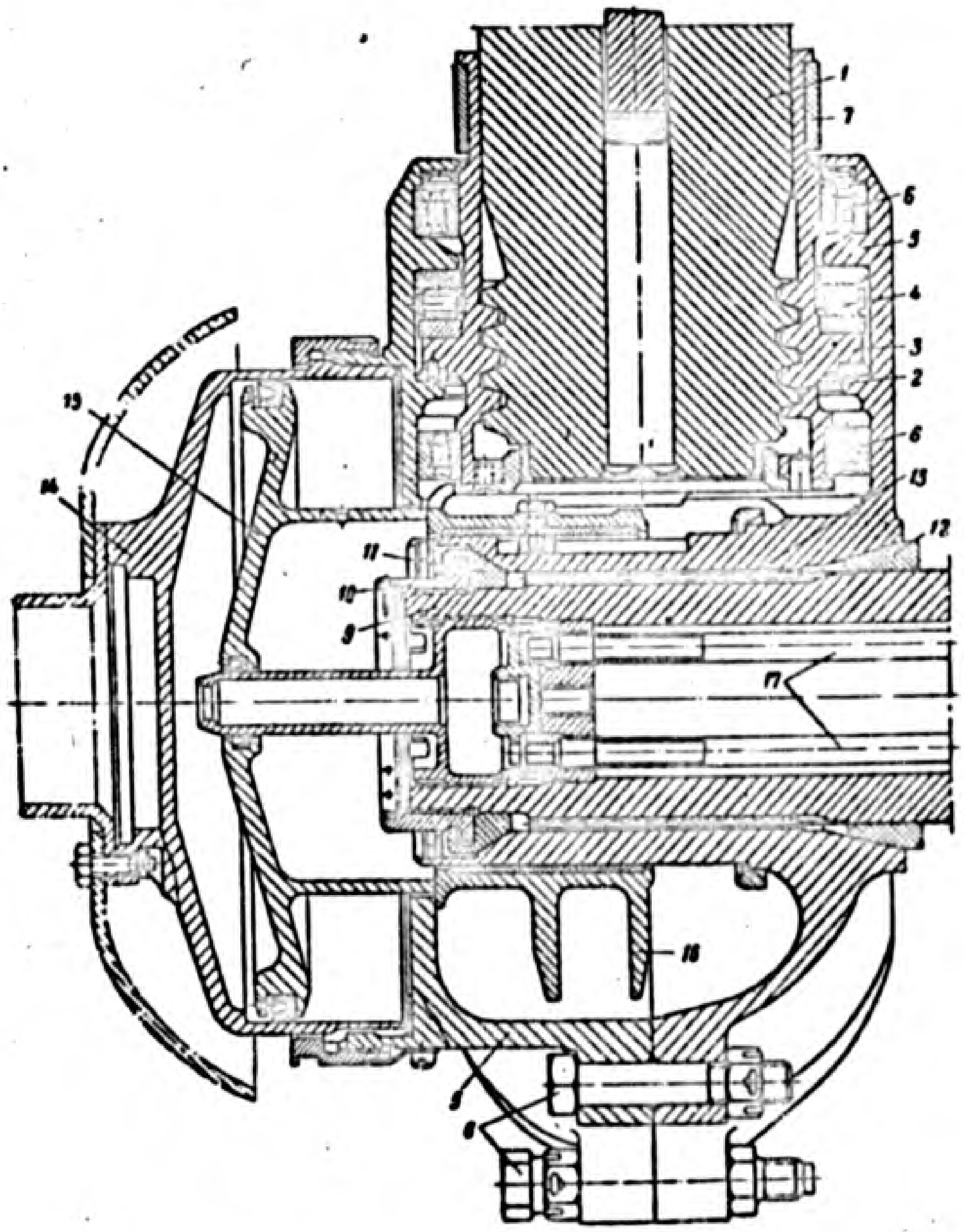


Fig. 180. Structural diagram of the hub of an AV - 5L - 24 propeller. 1) Blade; 2) thread; 3) cup; 4) support roller bearing; 5) hub housing; 6) support bearings; 7) collar; 8) lock bolts; 9) reduction gear shaft; 10) nut; 11 and 12) fore and aft centering cones; 13) splines; 14) cylinder; 15) piston; 16) cross-arm; 17) oil ducts of the VISh.

Connection of the parts of a split propeller housing is accomplished by lock bolts 8 (see Fig. 180). Attachment of the hub of such a propeller to shaft 9 of the propeller (to the shaft of the engine reduction gear is accomplished by use of nut 10, and also the fore and aft centering cones 11 and 12. The splined connection 13 of the shaft and hub serve to transmit torque from the engine to the propeller.

The propeller hub with an unsplit housing (see Fig. 181) is attached to the shaft by use of end splines 13 and dowels 15. The dowels transit thrust and bending moment from the propeller hub to the propeller shaft. The end splines, situated on the rear face of the hub housing and the forward face of the shaft flange, insure centering of the propeller relative to the reduction gear shaft and transmission of torque.

The piston hydraulic servomotor (see Fig. 181) consists of cylinder 1, piston 2, and a crankgear mechanism for transformation of the translational motion of the piston into rotational motion of the propeller blades. Piston 2 and cylinder 1 are fabricated from aluminum alloy. Cylinder 1 is attached to housing 7 of the hub by use of a nut. Piston 2 has a central hole for oil feed and is fitted with sealing cuffs.

Along with the servomechanism, an oil duct for feeding oil is also located inside the propeller hub,

In order for the oil not to congeal in the hub during prolonged flight in one regime under conditions of reduced temperature, many propeller designs, provide for oil circulation to heat the hub. In this case a jet is installed between the high- and low-pressure cavities to meter the quantity of circulating oil.

In many propeller hubs there are valves insuring the permanence of the blade-angle settings in case of a fall in the pressure of the oil fed to the servomotor, as well as centrifugal valves to maintain the angular setting of the blades in the case of an increase in rpm above the permissible limit (see Fig. 181).

Coaxial propellers are widely used on turboprop engines of high

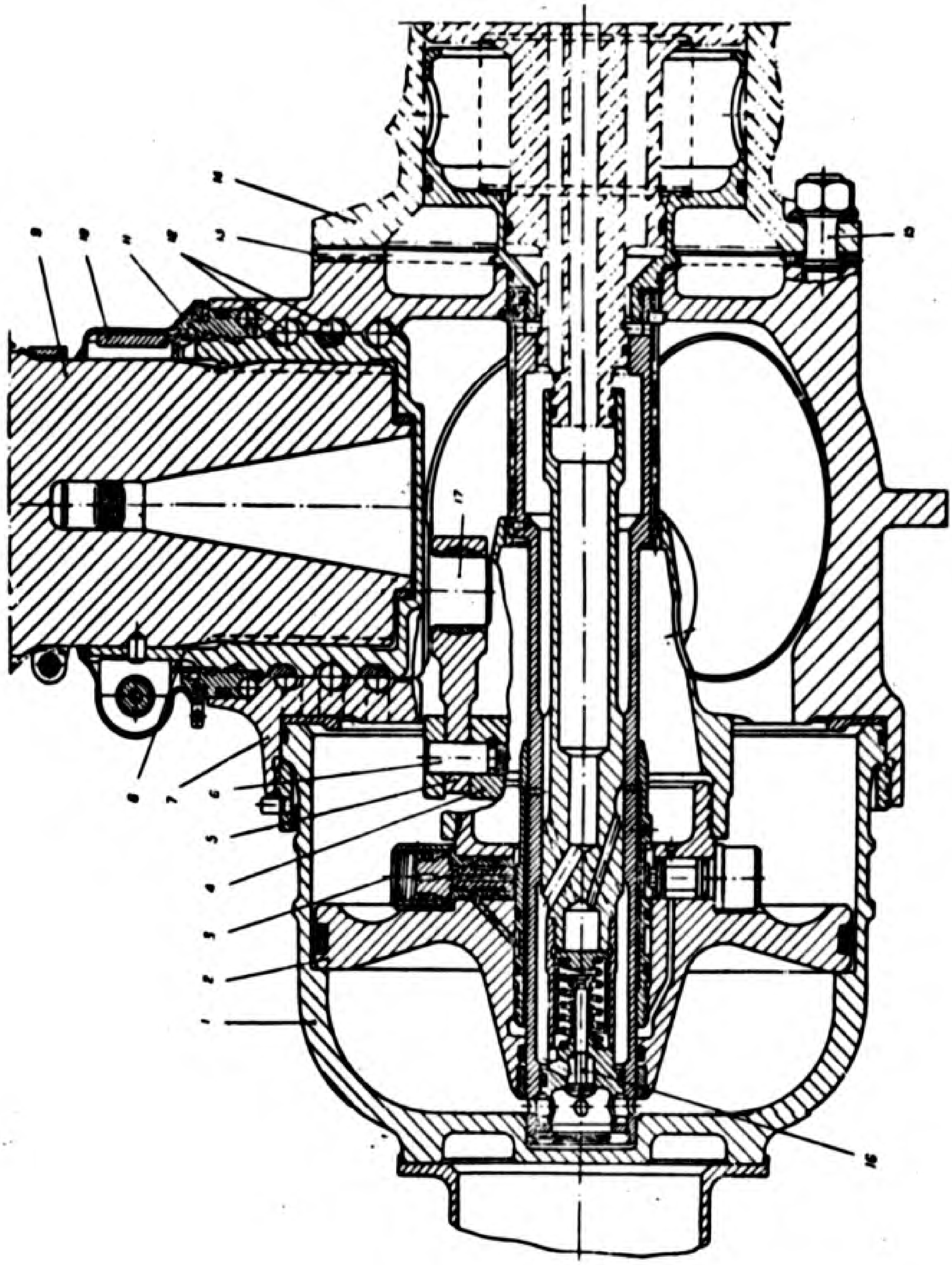


Fig. 181. Structural diagram of the hub of the AV - 68I propeller.
1) Cylinder; 2) piston; 3) centrifugal propeller pitch stop; 4) cross-arm; 5) connecting rod; 6) pin; 7) hub housing; 8) cup; 9) blade; 10) collar; 11) nut; 12) ball bearing; 13) splines; 14) propeller shaft; 15) dowel; 16) valve of pitch stop; 17) cup pin.

power, and these exhibit substantial advantages in comparison with single propellers. The diameters of coaxial propellers for the same power absorption is less than with a single propeller, which makes it possible to decrease the required height of the aircraft landing gear. When coaxial propellers are used, the magnitude of the reaction moment which constantly acts on the aircraft is significantly reduced, and the action of the gyroscopic moment due to aircraft maneuvering is nearly entirely eliminated. Coaxial propellers consist of a combination of two propellers located in tandem, turning in opposite directions. In this case, the forward propeller has no substantial design features as compared to a conventional single propeller. In the design of the rear propeller, provision must be made for passing the shaft of the forward propeller through the hub of the rear propeller. Therefore, the cylinder and piston of the rear propeller servomotor are made annular in shape.

Chapter 14

REDUCTION GEARS FOR TURBOPROP ENGINES

1. GENERAL INFORMATION

Depending on power and engine design, the gas turbines of TVD [turboprop engines] usually run at speeds ranging from 7000 to 16,000 rpm, while the propellers attached to these engines run at 700 to 1200 rpm. Propeller speed and size is dictated by the conditions that permit a propeller of given power to run with minimum losses. Reduction gears used with TVD are designed to transfer power from turbine to propeller with a reduction in speed.

The gear ratio $\underline{i}$ of a reduction gear is defined as the ratio of the propeller speed n_v to the turbine-shaft speed n_t, i.e., $i = n_v/n_t$. On TVD in service, reduction-gear ratios run from 1/8 to 1/16.

Tractor or pusher propellers may be used with TVD. At the present time, tractor propellers are most commonly used with TVD. Here the reduction gear is either located in the front section of the engine ahead of the compressor so as to form a single structure, or at some distance from the engine (offset reduction gear).

The reduction gear should be made as small as possible so as to reduce its weight and the hydraulic losses at the compressor intake. With high-power turboprop engines, this attempt to reduce size and weight of reduction gears makes it necessary to place the maximum amount of stress on the main reduction-gear elements, so that they must be made from high-grade steels with a high degree of precision.

In order to carry off the considerable amounts of heat evolved in a reduction gear owing to power losses to friction, TVD reduction gears are provided with oil systems in which the oil flows are several times greater than the amounts of oil pumped through the oil systems of turbojet engines.

As a rule, hydraulically actuated variable-pitch propellers (VISh) are used in conjunction with TVD; they are controlled with the aid of oil supplied to the VISh through a system of oil lines and oil distributors. The considerable oil pressure in VISh control systems makes it necessary to use special oil seals, especially between stationary and rotating reduction-gear elements.

When the engine is in operation, the structural elements of a reduction gear are subject to:

– a torsional moment that loads the shafts and gears of the reducer;

– propeller thrust, which exerts a tensile force on the propeller shafts which is transmitted through a thrust bearing to the reduction-gear housing;

– propeller inertial forces that appear during maneuvers or when the aircraft lands (these forces bend the shafts and load the bearings and housing of the reduction gear);

– gyroscopic propeller moments, appearing during maneuvering of the aircraft (the gyroscopic moment due to a single propeller bends the shaft and is transmitted through the bearings to the reduction-gear housing; the loads on reduction-gear structural elements due to gyroscopic moments associated with coaxial propellers depend on the arrangement of the stress-bearing system of the propeller hubs and the reduction gear);

– a transverse force due to the oblique slipstream produced when the aircraft flight involves sideslip. This force bends the shafts and loads the reduction-gear bearings.

2. TYPES AND KINEMATIC ARRANGEMENTS OF TVD REDUCTION GEARS

On the basis of the relative positions of the propeller and engine shafts, TVD reduction gears may be classified as *coaxial*, where the shaft axes coincide, and as *offset*, where the propeller-shaft axis is displaced with respect to the turbine-shaft axis. Reduction gears with offset shafts are used basically in coupled TVD (paired engines) to transmit power from the two engines to the common propellers, and in TVD using offset reduction gears. As a rule, reduction gears for coupled TVD are made up of simple gear trains.

Planetary gearing is used in coaxial TVD reduction gears; at high reductions, such gearing offers advantages over simple gearing with respect to weight and size. In certain cases, a TVD reduction gear may be made in the form of a combination of simple and planetary gearing.

Planetary reduction gear with simple planet gears to drive a single propeller (Fig. 182). The drive pinion z_1 is connected to the engine turbine; it turns at a speed n_t. Drive pinion z_1 engages the satellite gears z_2, which turn on shafts in the satellite carrier (cage); the satellite carrier is connected to the propeller shaft. When the reduction gear turns, the satellite gears z_2 roll along the stationary gear z_3 and rotate, carrying along the satellite carrier and propeller shaft at a speed n_v. The gear ratio for such a planetary reduction gear will equal

$$i=\frac{1}{1+\frac{z_3}{z_1}}. \tag{17}$$

The dimensions of the drive pinion z_1 are limited by conditions of strength, and thus for given dimensions (of gear z_3), it is difficult to obtain high reductions with a simple planetary reduction gear.

Planetary reduction gear with compound satellite gears to drive a single propeller (Fig. 183). In contrast to the preceding reduction-gear arrangement, in this case the satellite gears are of the compound type, with two gears of different diameter on a common shaft. The large-diameter gear z_2 engages the drive pinion z_1, while the small-diameter gear z_3 engages the stationary gear z_4.

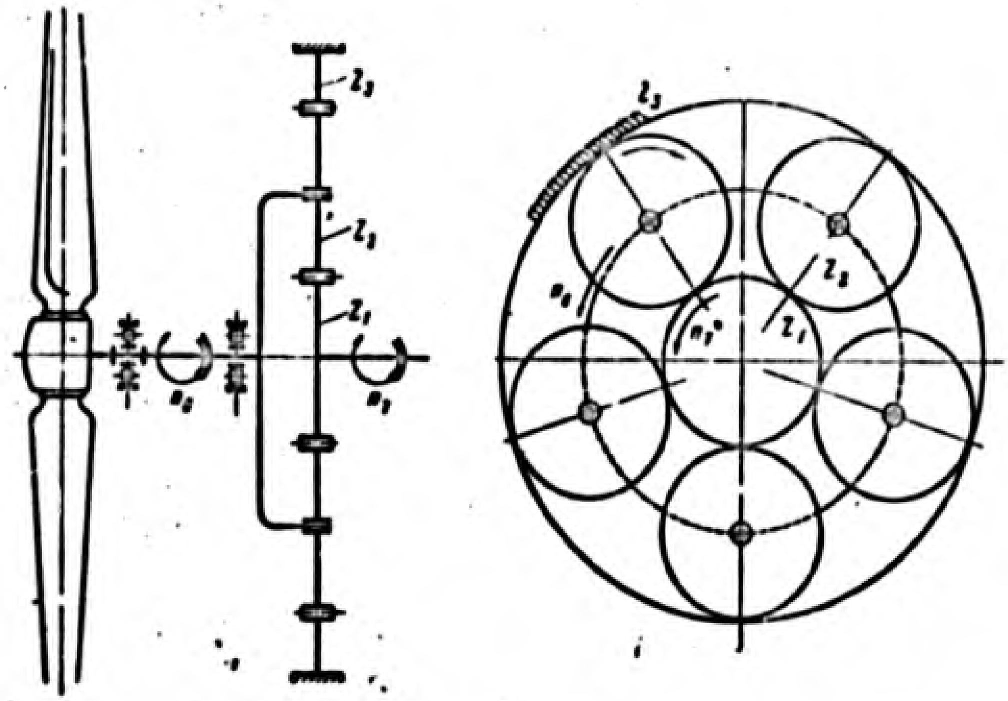


Fig. 182. Planetary reduction gear with simple satellite gears to drive a single propeller.

The gear ratio of a planetary reduction gear using compound satellites is determined from the formula

$$i=\frac{1}{1+\frac{z_2 z_4}{z_1 z_3}}. \tag{18}$$

Comparing these two arrangements, we can see that for identical dimensions of the drive pinion z_1, and exactly identical over-all dimensions of the reduction gear, the arrangement using the compound satellite gears enables us to obtain a higher reduction (provided that $z_2 > z_3$), than where an ordinary planetary gear is used with simple satellites. It is clear that for a given gear ratio and over-all reduction-gear dimensions, the diameter of drive pinion z_1 in the compound-satellite arrangement will be greater than where the device uses simple satellites; as a consequence, the peripheral force on the teeth of gear z_1 will be less.

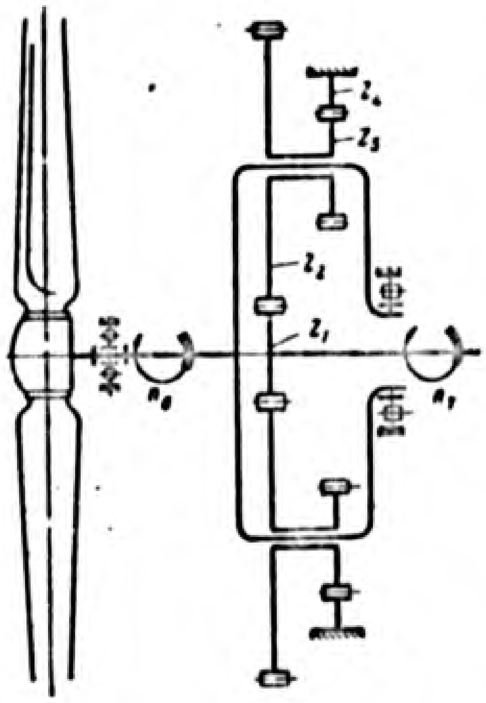


Fig. 183. Planetary reduction gea·· using compound satellites to drive a single propeller.

Closed differential reduction gear with simple satellites to drive single propeller (Fig. 184). The drive pinion z_1 engages the satellite gears z_2. The gears z_2, as in a planetary reduction gear, turn on shafts of the satellite carrier, which is connected to the propeller shaft. In contrast to the reduction gears discussed previously, in this case gear z_3 is not stationary, but rotates in a direction opposite to that in which the propeller shaft turns, as it is connected to this shaft by a simple intermediate gear train, consisting of the gear z_6, which turns together with the propeller shaft, the idler z_5, and the gear z_4, which turns together with gear z_3.

The intermediate gearing need not be of the simple form as in the example given, but can be planetary gearing made up of either spur or bevel gears. The gear ratio of the reduction gear under consideration is found from the formula

$$i = \frac{1}{1 + \frac{z_3}{z_1} + \frac{z_3 z_6}{z_1 z_4}}. \qquad (19)$$

By comparing Formulas (17), (18), and (19), for calculating the gear ratios of the systems just discussed, we can see that the differential closed reduction gear can yield a higher reduction for the same dimensions than a planetary reduction gear using simple or compound satellite gears. In the differential closed reduction gear (see Fig. 184), the engine torque is transmitted to the propeller shaft in two

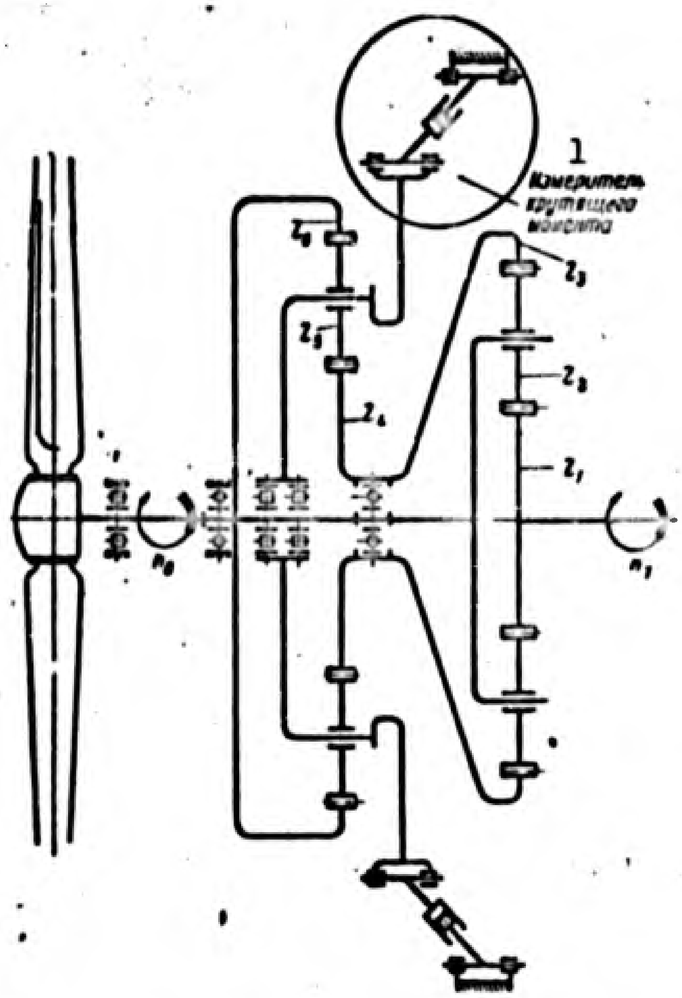


Fig. 184. Differential closed reduction gear for driving single propeller. 1) Torquemeter.

ways: from gear z_1 through satellite gear z_2, the satellite-gear shaft and satellite carrier to the propeller shaft, as well as from gear z_1 through satellite gear z_2, gears z_3, z_4, z_5, and z_6 to the propeller shaft.

Where the gear ratios and dimensions of a differential closed reduction gear and a planetary reduction gear are the same, it is possible to increase the diameter of gear z_1 in the former while reducing the dimensions of satellite z_2, thereby reducing the peripheral meshing forces and the load absorbed by the satellite-gear bearings.

Differential reduction gear using simple satellite gears to drive two coaxial contrarotating propellers (Fig. 185). Drive pinion z_1 engages the satellite gears z_2. The satellites z_2 roll along gear z_3, turning the satellite carrier and shaft of the front propeller in the

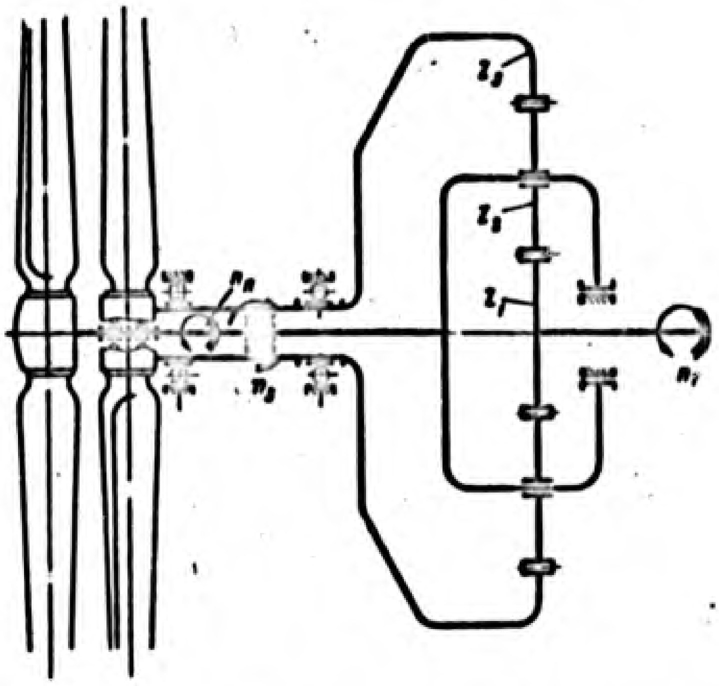


Fig. 185. Differential reduction gear using simple satellites to drive two coaxial propellers.

direction of rotation of the drive pinion z_1 at speed n_p, while simultaneously turning gear z_3 and the shaft of the back propeller in the opposite direction at speed n_z.

For a constant turbine speed n_t at the reduction gear, the speed of each of the propellers will depend on the blade angle. Thus, for example, when the load on the back propeller is lightened (the blade angle reduced), its speed will increase, while the speed of the front propeller will be reduced for a constant turbine speed, and vice versa.

With variable-pitch front and back propellers, given identical speeds are maintained by controlling each of them with the aid of a separate centrifugal governor, rotated by its own propeller and holding the speed constant by changing the blade angle.

The gear ratio of such a differential reduction gear will equal, where the propellers turn at the same speed,

$$i = \frac{n_z}{n_t} = -\frac{n_p}{n_t} = \frac{1}{1 + 2\frac{z_3}{z_1}}. \tag{20}$$

In a differential reduction gear, the propeller torques remain constant regardless of speed providing the engine turbine torque M_t is constant; here the torque of the front propeller will be greater than that of the rear propeller by an amount M_t. Thus, at equal propeller speeds, the power developed by the front propeller will be greater than the power developed by the back propeller.

For a differential transmission driving two coaxial propellers, the reduction-gear housing will be loaded by the torque imposed by the transmission. The difference in the torques equaling $-M_t$, is balanced by the drive-shaft torque, and the supports (attachment fittings) of the engine will receive only the torque due to the nozzle-box assemblies of the turbine driving the propeller.

Differential reduction gear using compound satellites to drive two coaxial contrarotating propellers (Fig. 186). In this reduction

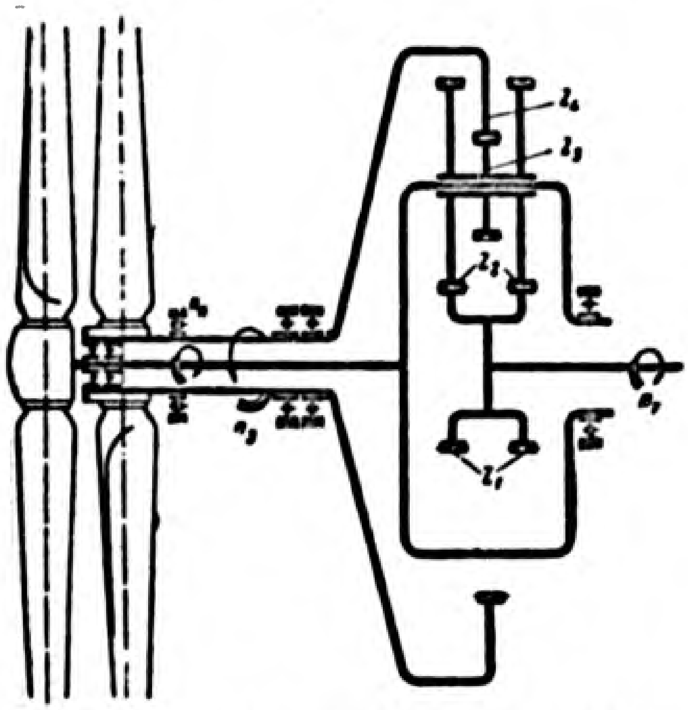


Fig. 186. Differential reduction gear using compound satellites to drive coaxial propellers.

gear, which also is kinematically indeterminate, the satellite gears are of the compound type, with gears of different diameters. The large-diameter gear z_2 engages the drive pinion z_1, while the small-diameter gear z_3 engages gear z_4. In the compound satellite shown in the figure, the large gear z_2 is split and located on the edges of the small-diameter gear z_3. Gear z_1 is also of the split type. This type of transmission ensures symmetric loading of a satellite and reduces the nonuniform loading of transmission teeth owing to skewing of the gear shafts. For identical propeller speeds, the gear ratio of a compound-satellite differential reduction gear is found from the formula

$$i = \frac{n_n}{n_T} = -\frac{n_3}{n_T} = \frac{1}{1 + 2\frac{z_2 z_4}{z_1 z_3}}. \qquad (21)$$

Differential reduction gears using simple satellites permit greater speed reduction to be obtained than do planetary gears consisting of the same gear elements; this is even more true where compound satellites are used.

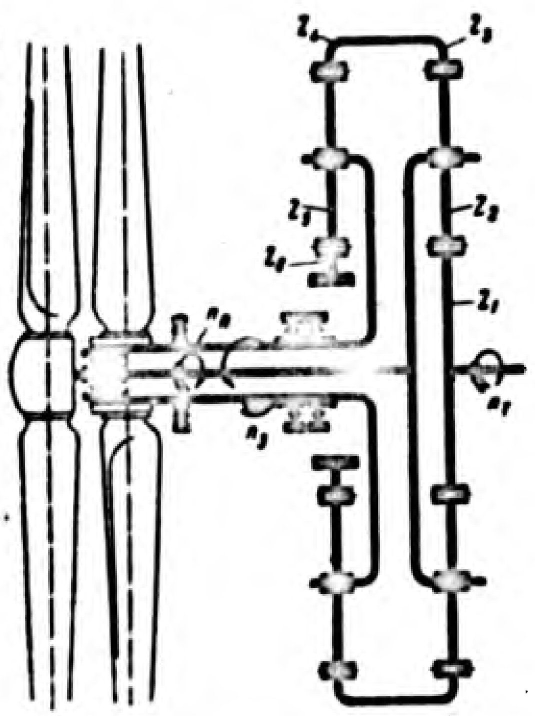


Fig. 187. Symmetric differential planetary reduction gear for driving coaxial propellers.

<u>Symmetric differential planetary reduction gear for driving two coaxial propellers</u> (Fig. 187). Here, in contrast to a differential reduction gear with simple satellites, the shaft of the rear propeller is connected to gear z_3 with the aid of planetary gearing consisting of gears z_4, z_5, and z_6.

The gear ratio of such a reduction gear, where the propellers turn at the same speeds and the diameters of symmetric gears are the same, is determined from the formula

$$i = \frac{n_n}{n_T} = -\frac{n_3}{n_T} = \frac{1}{2\left(1 + \frac{z_3}{z_1}\right)}. \qquad (22)$$

A still greater speed reduction can be obtained with a symmetric differential planetary reduction gear than with a differential reduction gear, with identical transmission dimensions and drive-pinion diameters.

Owing to the symmetry of the gearing, the peripheral forces on the teeth of all gears will be the same. The propeller torques will also be of equal magnitude. In the given case, the reduction-gear housing will be loaded by the torque from gear z_6, which is balanced by the torque of the turbine nozzle-box assembly. Thus the supports of a TVD using this type of transmission will not be loaded by any torque.

3. CONSTRUCTION OF BASIC REDUCTION-GEAR ELEMENTS

The basic structural elements of a TVD reduction gear are the housing, gears, shafts, satellite carriers, and bearings.

The housings of reduction gears are usually cast from magnesium or aluminum alloys, and consist of several parts joined together by studs or bolts. A reduction-gear housing should be sufficiently strong and rigid, since it carries forces of considerable magnitude, transmitted from the propeller, shafts, and other rotating parts of the reduction gear. The stiffness of this housing greatly affects the operating effectiveness of the reduction-gear gearing and bearings. Housing stiffness is increased by making the walls conical or spherical, using reinforcing ribs, joining separate parts of the housing by braces, rods, and similar stress-absorbing elements to form a single rigid box-like structure. Here proper location of housing structural joints and the type of stress-carrying connections between the reduction-gear housing and the engine housing are of great importance. As a rule, the reduction-gear housing carries the variable-pitch propeller speed governor, oil pumps, propeller-control elements, the elements used to monitor operation of propellers and reduction gear, and certain aircraft accessory units.

As an example, let us consider the housing of the reduction gear

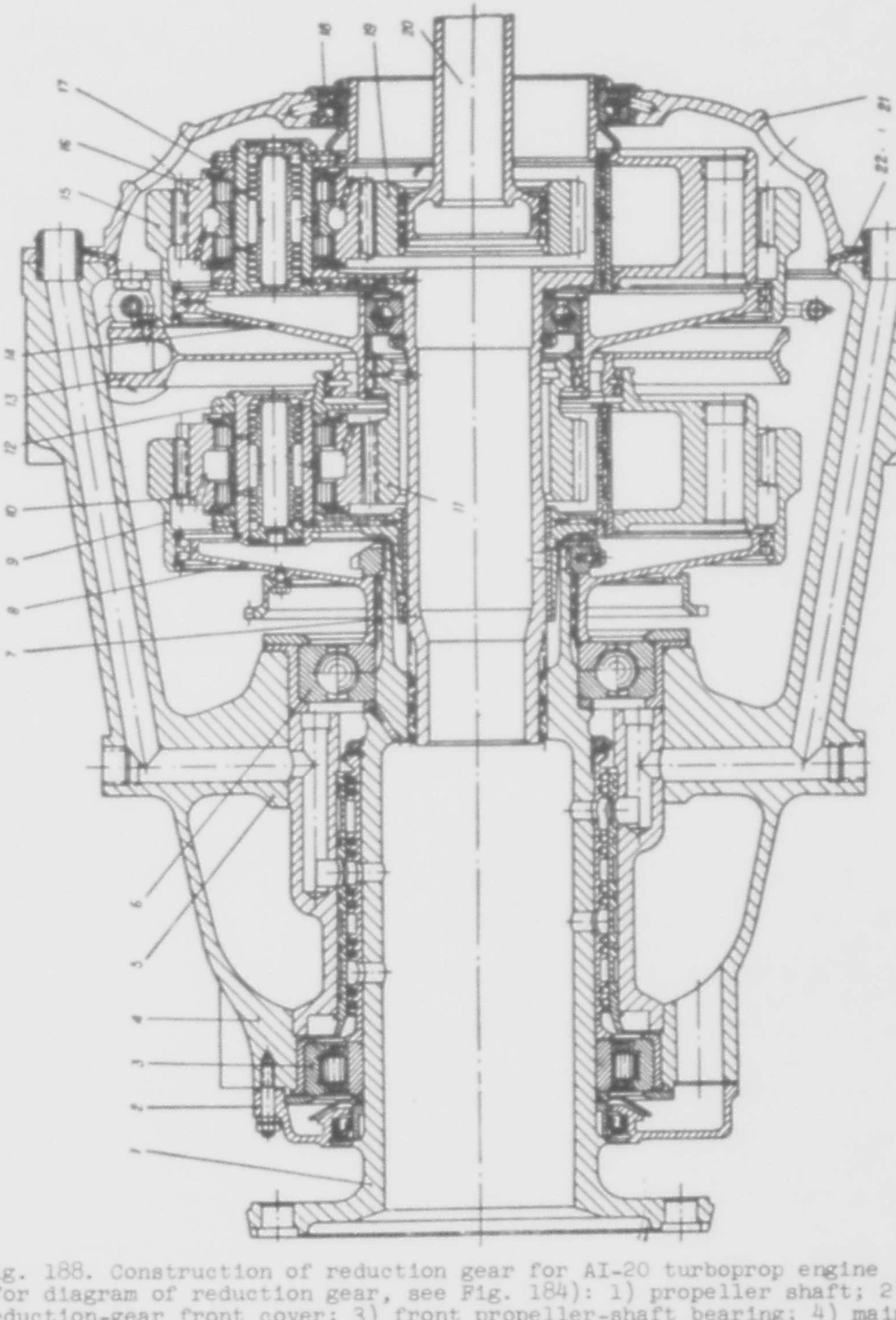


Fig. 188. Construction of reduction gear for AI-20 turboprop engine (for diagram of reduction gear, see Fig. 184): 1) propeller shaft; 2) reduction-gear front cover; 3) front propeller-shaft bearing; 4) main housing; 5) partition; 6) propeller-shaft rear bearing; 7) reduction-gear shaft; 8) spider; 9) internal gear of intermediate link gearing; 10) idler of link gearing; 11) drive pinion of link gearing; 12) housing of link intermediate gearing; 13) torquemeter lever-mechanism gear; 14) boss (splined connector); 15) internal gear; 16) satellite; 17) satellite carrier; 18) rear reduction-gear bearing; 19) drive-pinion teeth; 20) reduction-gear drive shaft; 21) rear-bearing housing; 22) flange.

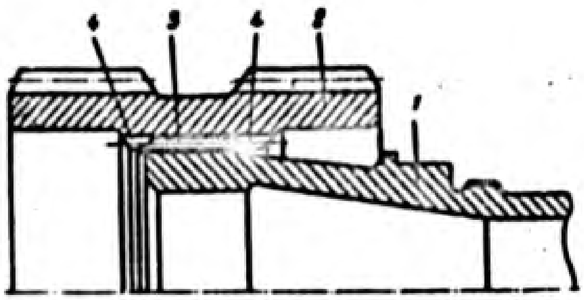


Fig. 189. Splined coupling between drive pinion and shaft: 1) shaft; 2) drive pinion; 3) involute splines; 4) clamp rings for axial positioning of gear.

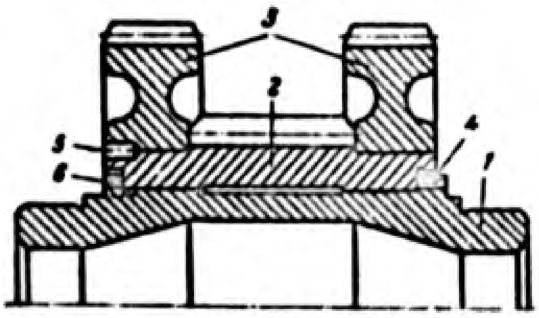


Fig. 190. Construction of compound satellite gear with split (twin) large gear: 1) satellite shaft; 2) small gear; 3) large gear; 4) splines; 5) pin; 6) nut.

shown in Fig. 188. The reduction-gear housing consists of two parts: the main housing 4, and the rear-bearing housing 21. The main housing 4 is cast from magnesium alloy and has the shape of a truncated cone. The front of the main housing and the transverse partition 5 have cutouts for the bushings of propeller-shaft bearings 3 and 6. The rear-bearing housing 21 of the reduction gear is spherical in shape and, together with the main housing, is attached by flange 22 to the compressor housing.

The gears carry the heaviest loads of all the structural elements of TVD reduction gearing. They are made of high-grade steels and are precision machined. Small-diameter gears of simple form are usually carburized, while large-diameter gears of complex shape are nitrided, since the gears deform less in nitriding than in carburizing, owing to the lower temperature to which they are heated. Spur gears with straight ground teeth are most frequently used in TVD reduction gears. Occasionally, reduction gears using helical gears are encountered.

Gears with long teeth are very sensitive to axial skewing, which leads to large maldistribution of the load along the length of the teeth, nonuniform wear, and premature failure. Gear-shaft skewing may be reduced, for example, by symmetric loading of satellites with respect to the vertical axis of symmetry. It is for just this purpose that the reduction gear shown in Fig. 186 is provided with a drive pinion and large satellite gears of compound type. The construction of the drive pinion in this reduction gear is shown in Fig. 189, and the satellite gear in Fig. 190.

Drive pinions of TVD reduction gears are usually small in diameter. The drive-pinion teeth may be integral with the shaft or hub, or may be connected to them by pins, flanges, or splines (see Figs. 188, 189, and 191). Centering of a gear tightly fitted onto the shaft is accomplished either along the side surfaces of involute splines or, with square splines (see Fig. 191), with the aid of special centering cones. Where splines are used to connect these parts, there may also be radial play (see Figs. 188 and 189), which makes it possible to have automatic positioning of the gear during operation of the reduction gear so as to equalize the load on the teeth of the satellites engaged with it.

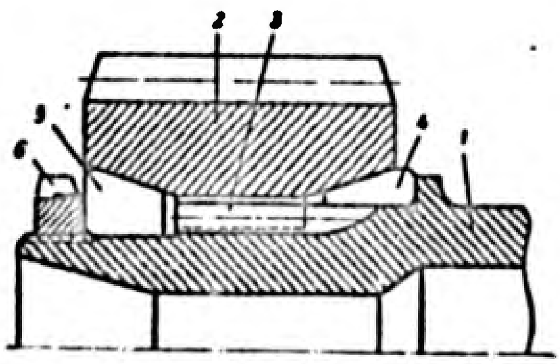


Fig. 191. Attachment of drive pinion to shaft: 1) shaft; 2) drive pinion; 3) splines; 4 and 5) front and rear centering cones; 6) nut.

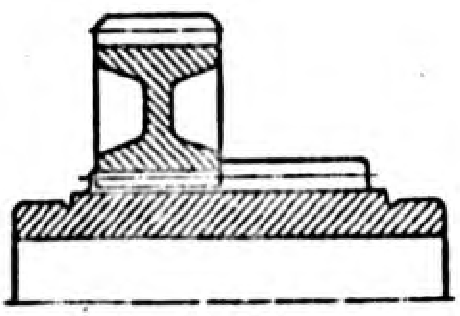


Fig. 192. Construction of compound satellite.

Multielement compound satellites, as a rule, are not made as integral structures so as to permit the teeth of each gear to be polished. Here the small-diameter gear is made integral with the satellite shaft, while the remaining gears are press-fitted onto the shaft and connected by splines (Fig. 192) or pins (Fig. 190). The three gears of the satellite (see Fig. 190) are connected together by axial pins 5, and in accordance with the reduction-gear assembly conditions, are installed on the

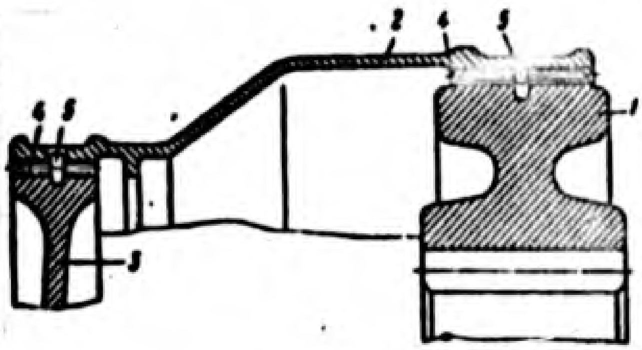


Fig. 193. Splined connection of large internal gear to hub: 1) internal gear; 2) spline connection; 3) hub; 4) spline; 5) wire rings for axial positioning of parts.

common shaft 1. The hub of the center gear basically operates in torsion, while the transverse loads are taken up by the shaft, which works only in bending.

Large internal gears (see Figs. 188, 193, and 194) are connected to coupling elements with the aid of involute splines with large radial gaps (0.1-0.2 mm), as a rule, which provide a certain degree of radial freedom for the gears. Such a design, together with the self-positioning feature of the drive pinion, provides more uniform distribution of the power transmitted among the satellites. Here, if the reduction gear has only three satellites, only one third of the total power will be transmitted to each of them.

Satellite carriers of TVD reduction gears are loaded by the centrifugal forces of the satellite gears and their shafts, as well as by forces that appear where the gears mesh. A satellite carrier will also absorb other loads if it is connected into the reduction-gear shaft stress-carrying system. Satellite carriers have a complicated box-like structure. The cage may be one-piece (see Figs. 188 and 194) or split, depending on the method used to install the satellites in the cage. The cage of a satellite carrier is integral with the shaft (see Fig. 188), or attached to it by splines or flanges.

In the design shown in Fig. 194, a one-piece satellite-carrier

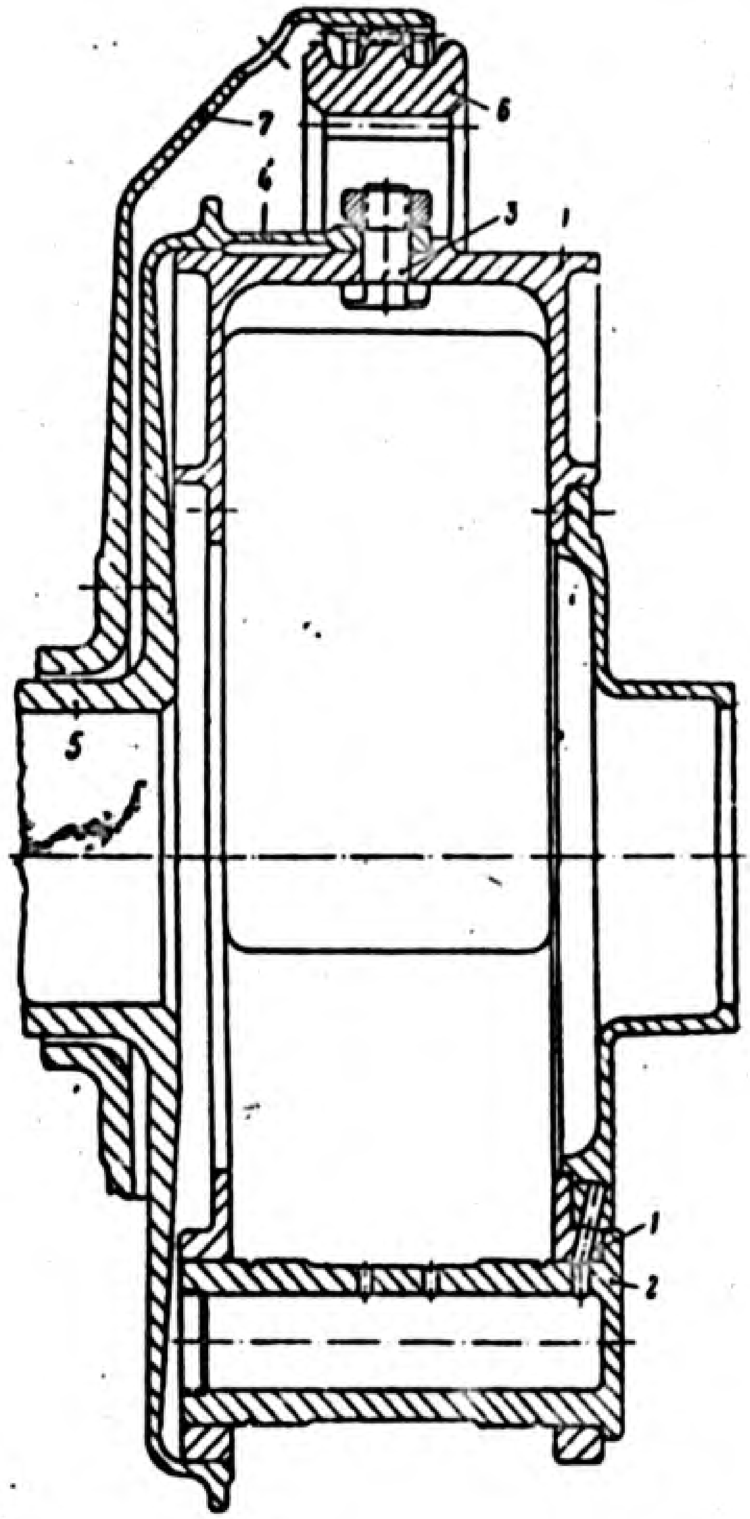


Fig. 194. Satellite carrier and large gear z_3 of reduction gear (a diagram of the reduction gear is shown in Fig. 185): 1) satellite carrier; 2) satellite shaft; 3) radial bulk; 4) flange of front propeller shaft; 5) shaft of front propeller; 6) gear z_3; 7) flange of rear propeller shaft.

cage is connected to the cylindrical shaft flange with the aid of radial bolts, for reasons of production technology.

Shafts of TVD reduction gears are made from high-grade alloy steels. All loads from the reduction-gear pinions and from the propel-

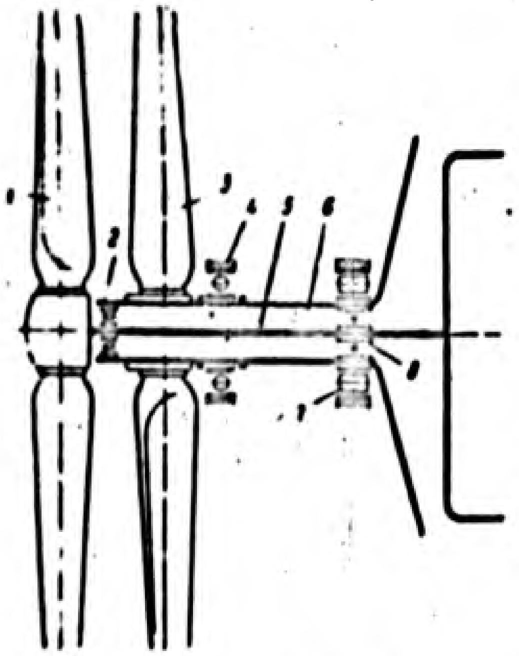


Fig. 195. Stress-carrying system of coaxial propellers, with inside shaft loaded by bending and torsional moments: 1) front propeller; 2) front propeller shaft radial bearing; 3) rear propeller; 4) rear propeller shaft radial-thrust bearing; 5) front propeller shaft; 6) rear propeller shaft; 7) rear propeller shaft radial bearing; 8) front propeller shaft radial-thrust bearing.

lers are transmitted to the shafts. At the end of the shaft, axial splines (see Fig. 180) or a flange and box splines (see Figs. 181 and 188) are used to attach the propeller hub. Coaxial propellers are usually attached to a flange on the outside shaft of the rear propeller.

Ball and roller bearings are used for the shafts; ball bearings simultaneously transmit propeller thrust to the reduction-gear housing. In the reduction-gear design shown in Fig. 188, the front roller bearing is a radial bearing, while the rear ball bearing is a radial-thrust bearing that absorbs the propeller thrust.

Stress-carrying systems for propeller shafts are determined by the number and location of the bearings and by the nature of the applied loads.

Basically, two types of stress-carrying systems are used for the shaft of a single propeller. Where the first arrangement is used, all loads from the propeller (thrust forces, inertial forces, gyroscopic moment, etc.) are absorbed by the propeller shaft and transmitted through the bearings to the reduction-gear housing; in this case, the ball bearing absorbing the propeller thrust may either be placed near the propeller (see Fig. 182), or following the radial roller bearing (see Figs. 184 and 188). In the latter case, the bearing loads are distributed more uniformly, but the propeller shaft will turn out to be loaded by the propeller thrust for its entire length.

Where the second arrangement is used (see Fig. 183), the propeller loads are transmitted to the propeller shaft and the attached satellite carrier, which has a second bearing at its rear. Such a stress-carrying system makes it possible to increase the distance between the propeller-shaft bearings and thus improve bearing operating conditions. The satellite carrier must be made stiffer and stronger, however.

The stress-carrying systems for the shafts of coaxial propellers are determined to a considerable extent by the methods used to mount and interconnect their hubs, i.e., the propeller stress-carrying system. In the shaft stress-carrying system for the reduction gear shown

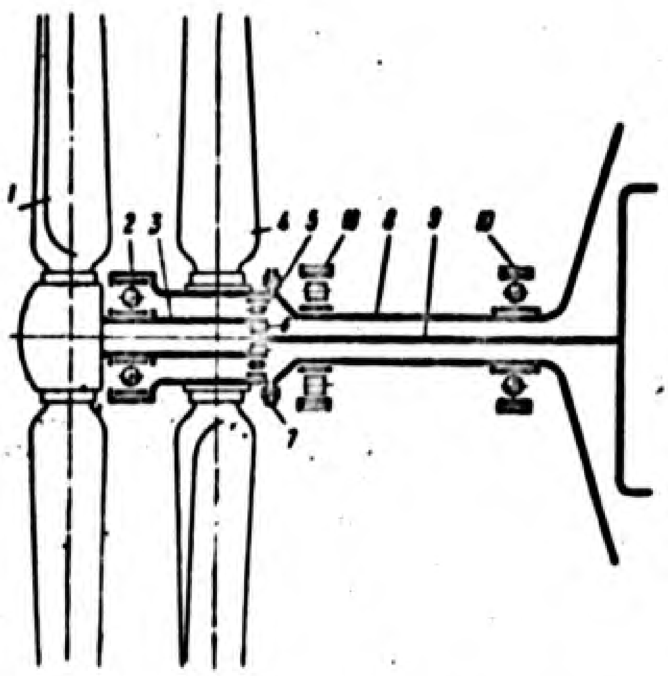


Fig. 196. Stress-carrying system for coaxial propellers in which the inside shaft is loaded by a torsional moment alone: 1) front propeller; 2) front propeller shaft radial-thrust bearing; 3) front propeller shaft; 4) rear propeller; 5) front propeller shaft radial bearing; 6) splines used to transmit torsional moment; 7) flange used to attach propeller hub to reduction-gear shaft; 8) reduction-gear shaft; 9) front propeller drive shaft; 10) reduction-gear bearings.

schematically in Fig. 195, each of the shafts is loaded by a bending moment due to its own propeller, including the gyroscopic moment. As a result, the reduction gear is considerably heavier. We can see that a more sensible propeller stress-carrying system is one in which all loads from the front propeller, except for the torsional moment, are transmitted to the rear propeller hub and then to the outside shaft of the rear propeller. Here the inside shaft is used only to transmit torque to the hub of the front propeller. In this stress-carrying system, the gyroscopic moments of the contrarotating propellers will cancel where the speeds and moments of inertia of the propellers are the same, and will not be transmitted to the reduction-gear shafts. Figure 196 gives a schematic diagram of such a stress-carrying system.

4. TORQUEMETERS (IKM)

Mechanisms for torque measurement (IKM) are usually included in the TVD reduction-gear structure, and can perform the following functions:

- measurement of engine torque for indication and simultaneous limitation of TVD takeoff power;
- torque measurement for indication and limitation of engine power during flight at low altitudes and high speeds, which is especially important for high-altitude TVD in which power is limited by reduction-gear strength;
- torque measurement for determination of engine power at cruising speeds to permit the pilot to maintain a given flight regime;
- providing input to the automatic propeller feathering mechanism under emergency conditions or where the engine is shut down in flight.

Figure 197 shows the basic arrangement of a lever-type hydraulic IKM for a turboprop engine using a planetary reduction gear. The torque M_{kr} acting on the stationary gear of the reduction gear is balanced by the moment created by the oil-pressure forces on the piston of the IKM hydraulic cylinders. When engine torque rises, the pistons are pushed into the hydraulic cylinders, covering the oil-discharge apertures, resulting in an increase in the oil pressure within the hydraulic sys-

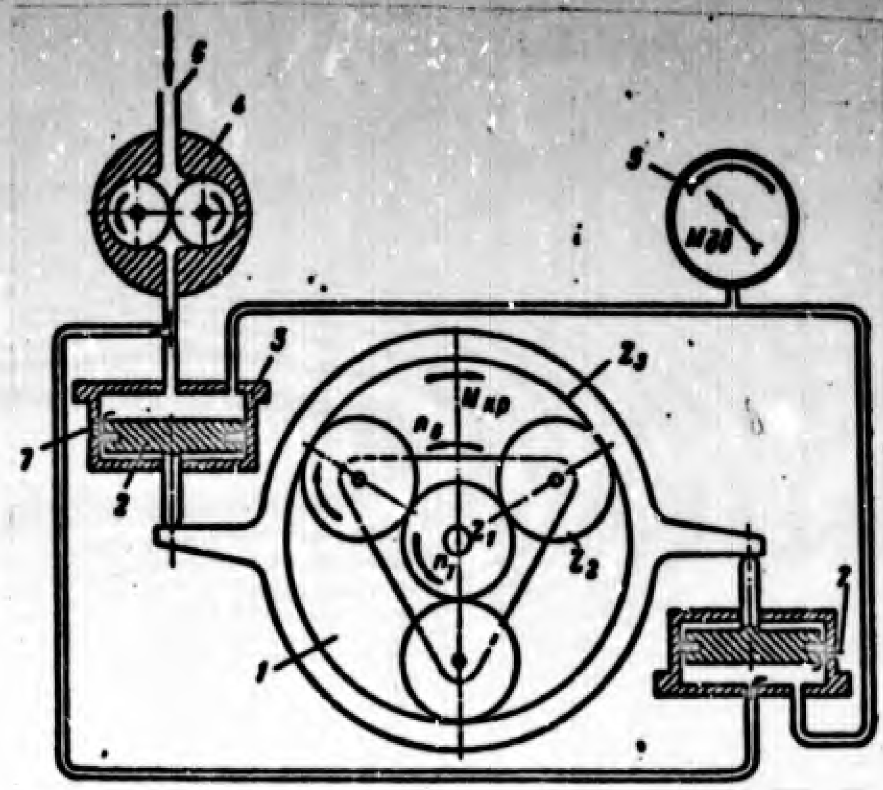


Fig. 197. Diagram of lever-type hydraulic IKM: 1) planetary reduction gear (for diagram of reduction gear, see Fig. 182); 2) piston; 3) hydraulic cylinder; 4) oil pump; 5) pressure gauge; 6) oil inlet; 7) oil drain.

tem. The oil pressure in the measuring line of the IKM hydraulic systems is directly proportional to the engine torque, and is measured by a pressure gauge or indicator of any type, installed in the cockpit near the pilot. The pressure-gauge scale may be calibrated in units measuring engine torque (for example, in kg), or directly in TVD power units, i.e., in horsepower, provided engine speed remains constant under all operating conditions.

It is also possible to measure engine torque using a hydraulic system to balance the torque of a fixed reduction-gear member with the aid of a special ball joint connecting the fixed gear to the reduction-gear housing. The basic arrangement of a ball-type hydraulic IKM is shown in Fig. 198. The torque of the fixed gear is taken up by balls acted on by normal forces N due to the peripheral forces T. Then the axial forces P appear at the balls, and are balanced by the oil-pressure forces on the piston of the IKM hydraulic mechanism. The magnitude of the balancing oil pressure is automatically maintained by the piston itself, which increases or reduces the amount of oil taken from the cavity of hydraulic cylinder 3 through the discharge vent 8 when equi-

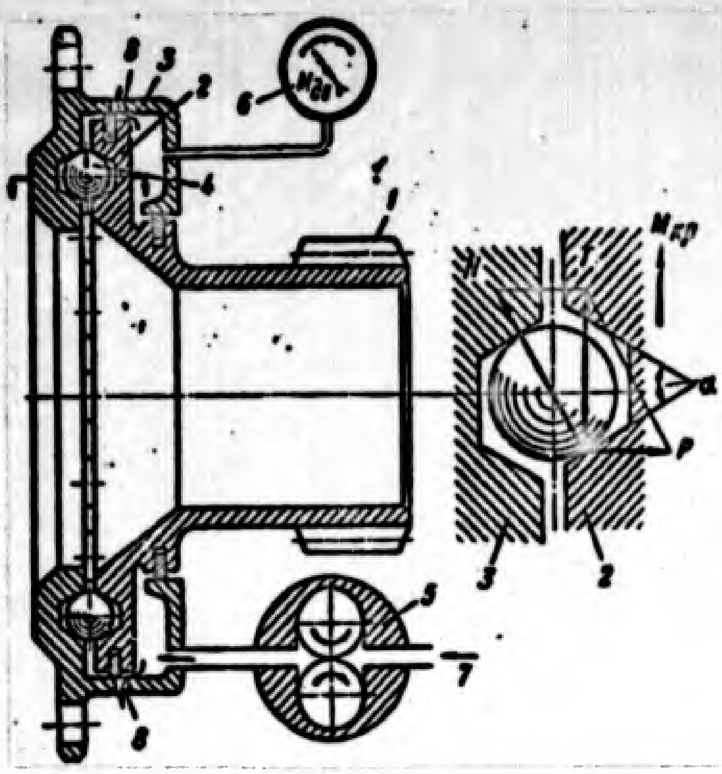


Fig. 198. Ball-type hydraulic IKM: 1) reduction-gear fixed gear; 2) piston; 3) hydraulic cylinder; 4) ball; 5) oil pump; 6) pressure gauge; 7) oil inlet; 8) oil-discharge vents.

librium is upset. Thus, the oil pressure in the hydraulic system is proportional to the engine torque, and is set automatically, as in the case of a lever-type hydraulic IKM mechanism. It should be noted that the calibration curve for a ball-type IKM may change with increased work hardening or wear of the surfaces in contact with the balls, owing to the high contact stresses.

The lever- and ball-type IKM systems just discussed can only be used with reduction gears having a fixed gear (for example, for the reduction gears shown in Figs. 182, 183, and 187), or with some other fixed element (for example, the housing of the idlers z_5 in the reduction gear shown in Fig. 184). In open-link differential reduction gears, torsion-type torquemeters may be used with hydraulic or electrical systems for measuring the angular deformation of a twisted shaft.

Figure 199 shows a torsion-type IKM using a hydraulic system to measure the torque. The drive shaft 1 of the reduction gear is connected by splines 2 to the TVD rotor, and carries torque to drive pinion 3, which is integral with the shaft. The torsion angle of this shaft is directly proportional to the transmitted torque. The inside shaft 4 is connected at the right end to the main shaft with the aid of the radial pins 5. The left end of shaft 4 carries the disk 6 of a hydraulic valve which closes the oil outlet from housing 7, press-fitted into the main shaft. As the engine torque increases, the angle

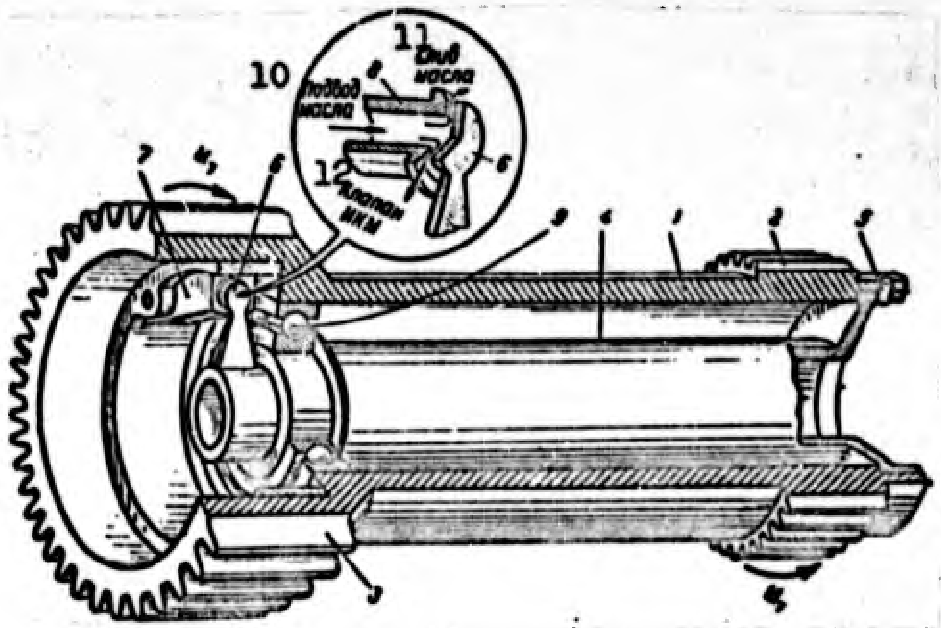


Fig. 199. Torsion-type IKM: 1) reduction-gear drive shaft; 2) splines; 3) drive pinion; 4) shaft; 5) pin; 6) hydraulic-valve disk; 7) valve housing; 8) valve seat; 9) bearing; 10) oil supply; 11) oil discharge; 12) IKM valve.

through which the main shaft is twisted will also increase, and the gap between the disk and seat 8 of the hydraulic valve will be reduced. Then the flow of oil out of the valve cavity will be reduced, and the oil pressure will increase. Oil is supplied to the hydraulic valve from a special pump with constant throughput; thus the oil pressure in the line is proportional to the engine torque.

Part Seven

ELEMENTS OF FUEL-SUPPLY SYSTEMS FOR GAS TURBINE ENGINES

GENERAL INFORMATION

Fuel-supply systems for gas turbine engines are designed to supply fuel to the combustion chambers. In the general case, the engine is provided with three fuel-supply systems: starting, main, and afterburner systems.

The starting fuel system is designed to supply fuel to the main combustion chambers when the engine is started. The starting system is automatically cut out after the starting process has been completed.

The main fuel system supplies fuel to the main combustion chambers throughout the entire period of operation of the engine.

The afterburner fuel system supplies fuel to the afterburners when they are on.

The main fuel system includes: the high-pressure main pump, fuel injectors, fuel booster pumps, fine and coarse filters, oil-fuel and fuel-air radiators, control and regulating elements for the fuel feed rate, lines and monitoring-measuring instruments (pressure gauges, flow meters, etc.).

The main fuel pump 6 (Fig. 200) supplies fuel to the injectors under the pressure needed for good atomization.

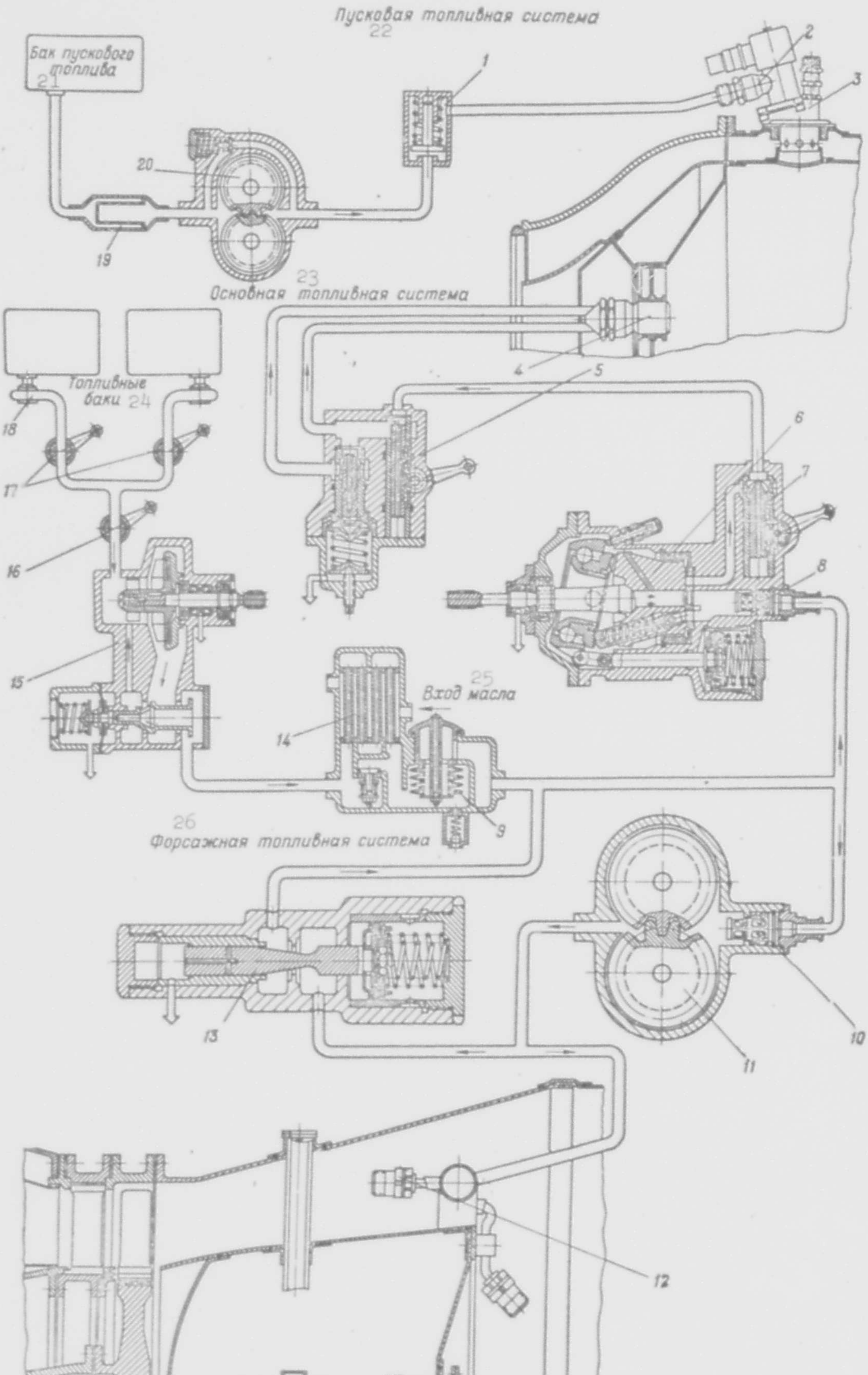
The fuel injectors 2, 4, and 12 introduce fuel into the chambers, providing a suitable fuel-air mixture for combustion and uniform distribution of the mixture over the chamber volume.

Fuel booster pumps 15 and 18 create the pressure needed to overcome the hydraulic resistance of lines and elements of the engine fuel-supply system located before the main fuel pump, as well as the pressure needed to overcome the phenomenon of cavitation at main-pump intakes (for more on this subject, see Chapter 15, the section headed "Piston Pumps"). The pumps 18 are installed directly in the tanks, and pump 15 on the engine.

The filters 9 and 19 for rough filtering are installed in the fuel system ahead of the main pumps, while the fine filters 8 and 10 are installed ahead of injectors, nozzles, and valves are designed to screen out solid particles and ice.

The oil-fuel radiator 14 is normally used to cool the oil by means of fuel, since air cooling of oil requires the installation of an oil-air radiator, which produces additional drag. Where fuel temperature is high, however, it is not only impossible to use an oil-fuel radiator to cool the oil, but it also becomes necessary to cool the fuel in a fuel-air radiator. The air in a fuel-air radiator is taken from a refrigeration system.

Control and regulating elements of the fuel supply system consist of shut-off valves 17 and stop (fire) valves 16 inserted into the aircraft fuel-supply system, automatic fuel feed-rate regulators, throttle valves 7, and stopcocks 5, designed to shut off the engine.


Пусковая топливная система
22
Бак пускового топлива
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1
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Основная топливная система
Топливные баки
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Вход масла
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Форсажная топливная система
13
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11
12

350

Fig. 200. Fuel-supply system: 1) Return valve; 2) starting fuel system injector; 3) starting spark plug; 4) main fuel injector; 5) stopcock; 6) main fuel pump; 7) throttle valve; 8 and 10) fine filters; 9 and 19) coarse filters; 11) afterburner fuel pump; 12) afterburner fuel system injector; 13) fuel-transfer valve; 14) oil-fuel radiator; 15 and 18) booster pumps; 16) stop valve; 17) shut-off valve; 20) starting fuel pump; 21) starting fuel tank; 22) starting fuel system; 23) main fuel system; 24) fuel tanks; 25) oil inlet; 26) afterburner fuel system.

Chapter 15

FUEL PUMPS

1. TYPES OF FUEL PUMPS, CHARACTERISTICS, AND FIELDS OF APPLICATION

Fuel pumps for present-day gas turbine engines should possess maximum throughput in the 10,000-20,000 kg/hr range at maximum outlet pressure (depending on type of injector) of up to 40-150 kg/cm^2 [27].

Fuel-supply systems may employ piston, gear, or centrifugal pumps. Each of these pump types has its own advantages and disadvantages that determine possible regions of application.

Piston pumps are the most common variety. The basic advantages of these pumps are the possibility of using high fuel pressures and of regulating fuel flow rate for constant speed.

Piston pumps are more complicated to manufacture than the other types of pump. They are very sensitive to corrosion, mechanical impurities, coke and gum, fuel water content, and to high ambient temperatures. All of these factors increase the friction forces between pistons and rotors, which may lead to seizing of the pistons and pump failure.

Piston pumps may be used as main pumps in engine fuel-supply systems requiring fuel pressures in the 80 to 150 kg/cm^2 range.

Gear pumps are very simple in construction and, in addition, are fairly insensitive to the grade of fuel used or its water content, to fuel and ambient temperatures. Gear pumps have 1.5-2 times the throughput of piston pumps of the same size and weight. Gear pumps are also very reliable in service.

Together with these advantages, gear pumps have the following basic drawbacks: difficulty of providing more than 50-60 kg/cm^2 of fuel pressure at the outlet and of ensuring constant throughput when engine speed changes. Gear pumps are provided with fuel-transfer systems for operation under conditions of reduced fuel flow rates. Fuel transfer increases the amount of power expended to drive the pump and causes a rise in the temperature of the pump and the fuel circulating in the lines.

Gear pumps have come into use as main pumps in fuel-supply systems using pressures of up to 50-60 kg/cm^2, as well as in starting systems and fuel-supply systems for turbostarters.

Centrifugal pumps are characterized by the possibility of obtaining very high throughput with small size and weight. Centrifugal-type pumps are the least sensitive to the quality of fuel employed.

Centrifugal pumps are mainly used as booster pumps, but can also be used as main pumps in fuel-supply systems for engines requiring fuel flow rates exceeding 8000-10,000 kg/hr and pressures ahead of injectors in the 50 to 70 kg/cm^2 range [27].

2. PISTON PUMPS

A piston pump (Fig. 201) has a rotor 13 whose circumference has openings for the pistons 6; the openings are inclined with respect to the axis of rotation. The end of the rotor presses against the distributor valve 1, which has intake and pressure ports 15 and 16, which communicate with the corresponding lines. The spherical end of the piston rests in an oblique disk 8. By means of piston 14, the inclined plate may be set at various angles to the plane of the distributor valve, turning about shaft 7.

In the position shown in the diagram, piston a is at the lower dead point – the minimum distance from the distributor valve. The rotor vent lies over the material between the intake and pressure ports (in the diagram, the valve has been arbitrarily turned). When the rotor turns out of this position, the piston slides along the inclined disk, executes a translational motion, and moves away from the valve. The rotor port communicates with the valve intake port. As the piston moves, a vacuum is created under it, and the free volume is filled with fuel from the intake line.

After a half revolution, the piston reaches the top dead spot (piston b in the diagram), and is at the maximum distance from the valve. As rotation continues, the piston moves toward the lower dead point and expels fuel through the valve pressure port.

In one revolution of the pump rotor, the piston supplies a volume of fuel to the pressure line equal to the volume between the top and bottom dead points. This volume is proportional to the piston stroke and its diameter.

The piston is 14-15 mm in diameter. The piston stroke depends on the angle of inclination of the disk. The greater the angle of inclination of the disk to the plane of the valve, the longer the piston stroke (Fig. 202). In pump designs in use, the piston stroke is 15-25 mm.

Pump throughput is determined by the amount of fuel supplied by one piston, the number of pistons, and the pump rotor speed. As a rule, there is an odd number of pistons: 5, 7, 9, or 11. The most common types of pump use seven or nine pistons.

The following considerations dictate the choice of an odd number of pistons. The amount of fuel supplied by the piston is proportional to the speed of reciprocating motion, which changes with the angle of rotation of the pump rotor (Fig. 203). The fuel feed rate of a piston is at a minimum when the piston is near the top or bottom dead points.

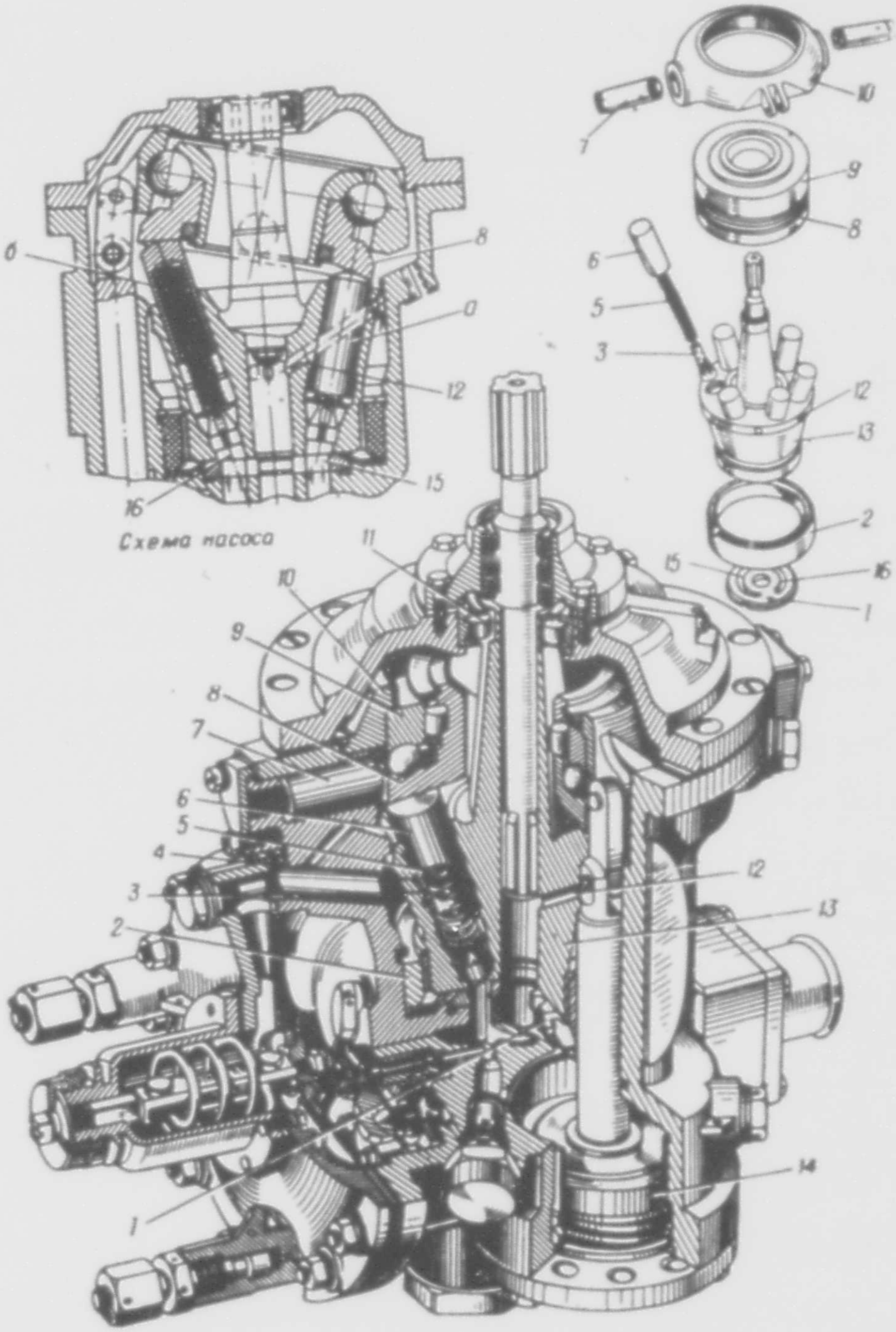

Fig. 201. Plunger-type fuel pump for VK-1A engine: a and b) pistons; 1) distributor valve; 2) copper-graphite friction bearing; 3) guide; 4) ring groove; 5) spring; 6) piston; 7) shaft of inclined disk; 8) inclined disk; 9) inclined-disk bearing; 10) inclined-disk housing; 11) roller bearing; 12) inclined borings; 13) rotor; 14) piston; 15 and 16) intake and pressure ports; 17) diagram of pump.

Pump throughput is made up of the amounts of fuel supplied by the individual pistons and, as we can see from Fig. 203, this will also vary with the rotor angle of rotation. The uneven fuel feed rate also causes fuel-pressure pulsations at the pump outlet, leading to pulsating combustion of the fuel in the chambers. Feed-rate pulsations will be reduced as the number of pump pistons increases. With an even number of pistons, however, supply-rate pulsations will be greater than with an odd number of pistons, even where there is one less piston in the odd-number set. This is explained by the fact that with an even number of pistons, the two diametrically opposed pistons will simultaneously have minimum supply rates (points A in Fig. 203). With an odd number of pistons, when one piston is producing a zero feed rate, the other pistons, located over the pressure port, will supply fuel at a rate that is considerably different from zero.

Cavitation may appear during operation of a piston pump owing to the vacuum at the pump inlet. By cavitation we mean the liberation of air (or fuel vapors) dissolved in the fuel when the fuel pressure drops. As a result of cavitation, not all of the volume freed by the piston will be filled with fuel, but part of the volume will be occupied by air and fuel vapors, and the pump throughput will decrease. This phenomenon appears with especial strength in high-altitude flights, where the drop in atmospheric pressure reduces the pressure in the fuel tanks.

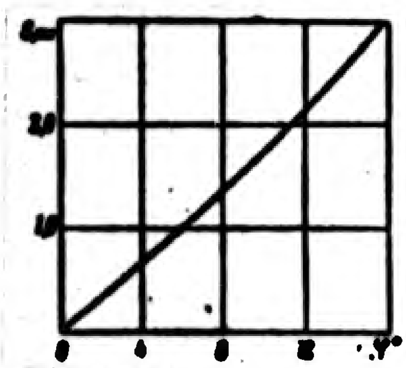

Fig. 202. Piston stroke S as a function of disk angle of inclination φ.

In order to prevent cavitation, it is necessary to add booster pumps to maintain the pressure at 1.5 to 5-6 kg/cm^2 at piston-pump inlets.

The pistons used in pumps are thin-walled cylinders. The central bore of the piston contains a spring 5 with guide 3 (see Fig. 201).

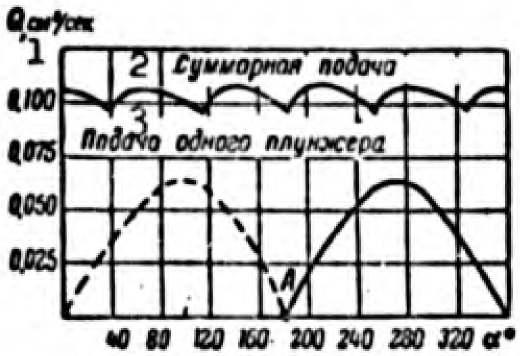

Fig. 203. Fuel feed rate Q as a function of pump-rotor angle of rotation α. 1) Q, cm^3/sec; 2) total feed rate; 3) feed rate due to one piston.

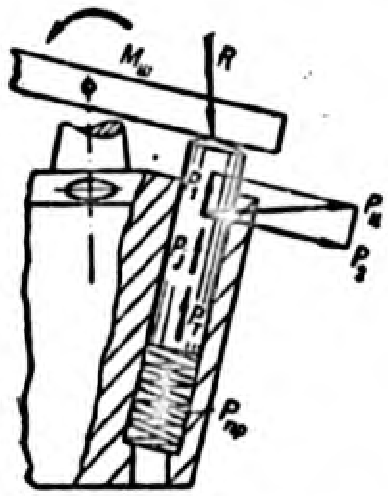

Fig. 204. Diagram showing forces and moments acting on piston and inclined disk: P_t) force due to fuel pressure; P_{pr}) spring force; P_j) inertial force; P_{ts}) centrifugal force; P_1) component of centrifugal force along piston axis; P_2) component of centrifugal force perpendicular to piston axis; R) force exerted by piston on inclined disk; M_{sh}) moment acting on disk.

Figure 204 gives a diagram of the forces acting on a piston. The force due to fuel pressure will take on two constant values in turn over the course of a single revolution: when the piston communicates with the intake line, it equals roughly 3 kg and is directed toward the distributing valve, since the pressure in the pump housing is greater than the pressure in the intake line; when it communicates with the pressure line, the force due to fuel pressure reaches 100-180 kg and is directed toward the inclined disk.

During the nonuniform reciprocating motion of the piston, an inertial force of 5-7 kg appears, which tends to pull the piston away from the inclined plate for about half of the intake and pressure strokes, while tending to press the piston against the plate for the rest of the time. Since during the intake stroke, the piston can move away from the inclined plate, the spring force is so chosen as to exceed the fuel-pressure force and inertial force during intake. In pump designs in use, the spring forces are 10-15 kg.

Rotation of the piston together with the rotor causes a centrifugal force reaching 150-200 kg. This force may be resolved into two components, one directed along the piston axis and the other perpendicular to this axis. The force acting along the piston axis helps to press the piston against the inclined plate and to overcome the force of friction which appears basically owing to the second centrifugal-force component. The friction force depends on the surface finish of the rubbing surfaces and the materials of which they are made. In order to reduce friction, the pistons are lapped in the rotor openings.

The rotors are made of aluminum bronze or steel. By using bronze, it is possible to improve heat exchange between rotor and fuel, and also to reduce the friction force between piston and rotor. Steel rotors are used where strength requirements dictate. Where steel rotors are used, bronze bushings are press-fitted to act as piston guides in order to reduce friction. Several ring grooves 4 (see Fig. 201) are made in the guides or in the piston channels. The grooves are designed to remove solid particles appearing in the clearance between rotor and piston and also to equalize fuel pressure over the piston circumference. There is fuel in the gap between rotor and piston, and if its pressure is not equalized along the circumference, the piston will fit eccentrically into the channel, press against one side, and will operate without lubrication, thus increasing friction forces.

In the space between the ports under the piston, the rotor has oblique borings 12, which connect the space within the rotor bore, which communicates with the intake line, and the pump-housing cavity. These borings act as a centrifugal pump and are designed to increase the pressure within the housing. The excess pressure in the housing over the pressure in the intake line helps to press the rotor against the distributor valve. This reduces fuel leakage through the end gap between rotor and distributor valve.

The distributor valve is made of steel or bronze; to reduce friction between the rotor and valve, they should be made of different materials. The end of the distributor valve facing the rotor acts as a thrust bearing for the rotor.

The rotor radial supports are the roller bearing 11 and copper-graphite journal bearing 2. In some designs, the copper-graphite bearing is replaced by a roller bearing.

The inclined disk acts as a combination thrust and radial bearing. The second race of bearing 9 is press-fitted into housing 10 of the inclined disk. Thus, when the pump operates, the inclined disk is also forced to rotate by the piston friction forces. Since the points of contact between the pistons and the inclined disk are located at different distances from the rotor axis of rotation and, consequently, they move at different circumferential velocities, and since the piston friction force against the disk varies (with the rotor angle of rotation), the inclined disk does not turn uniformly, but slips with respect to the piston. As a result, while such pumps are operating, the points of contact between the pistons and disk will change, which facilitates uniform wear of the disk surface.

The pistons exert forces on the disk that create a moment with respect to the disk axis of rotation (see Fig. 204). In pump designs in use, this moment is so directed as to tend to turn the inclined disk to the zero angle. A force of 60-70 kg is required to maintain the disk in a given position.

3. STRUCTURAL FEATURES OF GEAR-TYPE FUEL PUMPS

The main operating feature of gear-type fuel pumps that distinguishes them from similar oil pumps lies in the fact that they must provide considerably greater flow rates and fuel pressures with little pulsation.

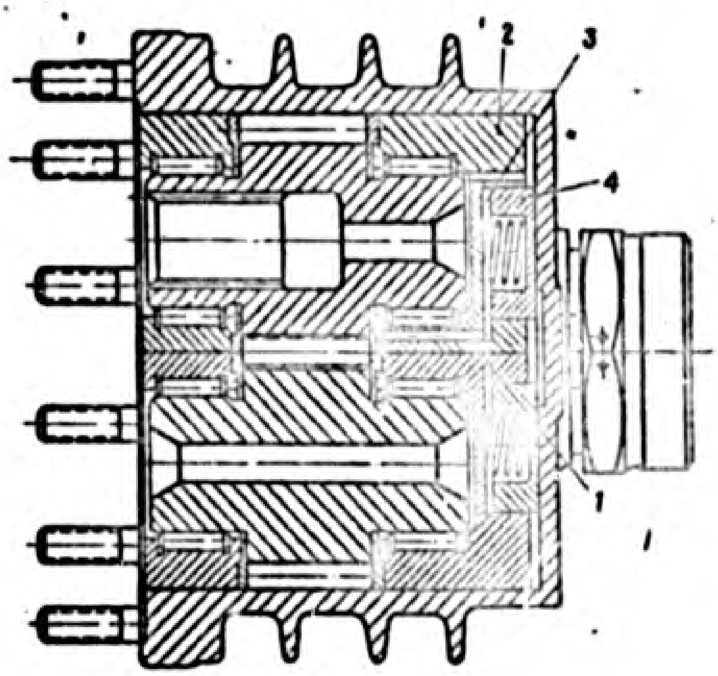

Fig. 205. Gear-type fuel pump with "floating" end bushings: 1) boring; 2) end bushing; 3) cavity; 4) piston.

In order to reduce flow pulsations, the gears in such fuel pumps must have larger numbers of teeth than corresponding oil pumps, despite the fact that an increase in the number of teeth increases pump size. In the gear-type fuel-pump designs in use, the gears have 10-17 teeth.

At high fuel pressures at gear-pump outlets, considerable leakage is observed through the radial and especially through end gaps. Leakage is especially severe where the pump housings are not very stiff. Fuel reaching the gap bends the housings, leading to a still greater increase in the gap. Thus fuel pumps are characterized by massive housings, which are sometimes spherical or even provided with ribs.

The most effective method of reducing leakage through end gaps is the utilization of so-called "floating" end bushings 2 (Fig. 205). The bushings are located between the housing and the gears. Fuel is supplied from the pressure line through borings 1 to the space 3 between the piston 4 and the bushing. The force due to the fuel pressure presses the pistons against the housing and the bushings to the gear faces. In order to reduce wear, bronze facings are used on the ends of these bushings.

Since the loads on gear-type fuel pumps are high, the shafts are supported in most cases by needle bearings. Journal-type bearings are used in starting-system pumps or booster pumps. The bearings are usually lubricated by the fuel, supplied from the end gaps.

4. CENTRIFUGAL FUEL PUMPS

In a centrifugal pump (Fig. 206), fuel from the inlet duct 1 is supplied to the rotating impeller 2. Running through the impeller channels, the fuel is thrown out at high pressure and speed into the scroll

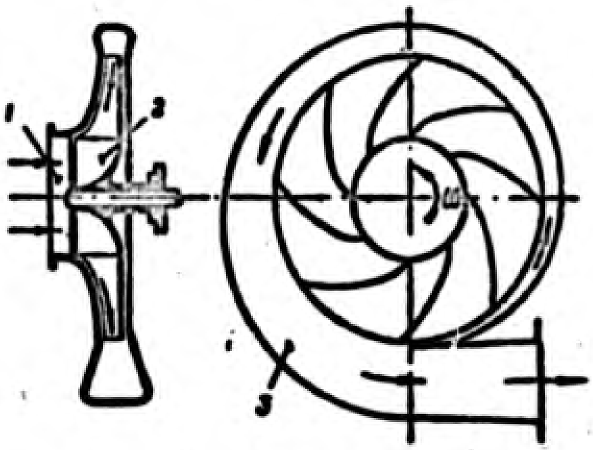

Fig. 206. Centrifugal pump: 1) inlet duct; 2) impeller; 3) scroll.

3. The pump impeller may have radial vanes or vanes bending away from the direction of rotation. Such swept-back vanes are the most commonly used type.

Impellers in common use may be open, half-shrouded (see Fig. 206), or shrouded (Fig. 207). Shrouded impellers are especially difficult to manufacture.

Impellers are loaded by high axial forces appearing owing to fuel pressure. In order to reduce the pressure behind the impeller, holes 5 are drilled in the wheel hubs, through which fuel is transferred to the intake side. In order to reduce this flow, slit-type seals 6 are used for the impeller.

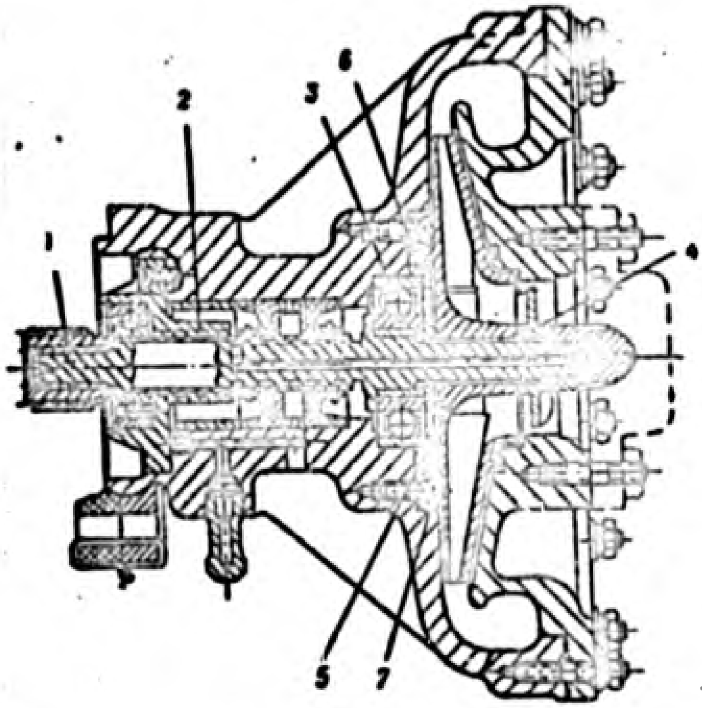

Fig. 207. Centrifugal pump with shrouded impeller: 1) pump shaft; 2) bushing; 3) ball bearing; 4) axial impeller; 5) port; 6) seal; 7) screw.

Impeller shafts usually turn in rolling-friction bearings. If the pump is driven by the engine rotor, the ball bearing nearest to the drive mechanism is lubricated with oil supplied from the engine drive housing. Ball bearing 3 is lubricated by fuel, which is also used to cool the pump shaft 1. Fuel passing through bearing 3 arrives in the helical channels under bushing 2 and then goes through the radial and axial borings in the shaft body, emerging on the intake side.

As with other types of pump, centrifugal pumps tend toward cavitation. In order to reduce cavitation, it is necessary to increase the pressure ahead of the pump inlet, and in certain cases, an axial impeller 4 is installed at the centrifugal-pump inlet. The axial impeller has a design throughput that is two to three times greater than that of the centrifugal impeller.

Chapter 16

FUEL JETS

1. TYPES OF FUEL JETS, CHARACTERISTICS, AND POSSIBLE REGIONS OF APPLICATION

Fuel may be introduced into a chamber in vapor or liquid form, and vaporizing or atomizing nozzles are used for this purpose, respectively.

Vaporizing nozzles (Fig. 208) are simple in design and easy to manufacture, since high precision is not required.

From the nozzle 1, fuel is supplied under slight pressure to U-shaped vaporizing tubes located in the combustion chamber. Entering the tubes, the fuel is heated, vaporized, and mixed with air. A rich fuel-air mixture is formed with an air-fuel ratio $\alpha = 0.25$-0.30, and is then directed into the combustion chamber.

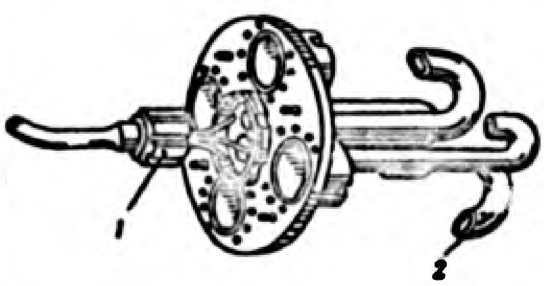

Fig. 208. Vaporizing fuel jet: 1) jet; 2) vaporizing tube.

Vaporizing jets are not widely used owing to several inherent disadvantages associated chiefly with difficulties in final installation and adjustment.

Atomizing nozzles supply a jet of fuel to the combustion chamber which is then broken up into fine drops, i.e., atomized. High combustion efficiency, stable combustion, and reliable starting requires the atomizing jets to furnish an optimum combination of fuel atomization fineness and jet shape, which is fundamentally determined by the cone angle, called the flair angle. Atomization increases the fuel-air contact surface, aids in mixture ignition and complete combustion. Excessively fine atomization reduces combustion stability with lean mixtures, since rapid vaporization of the fine drops produces a uniform air-fuel mixture with a narrower range of ignition than a nonuniform mixture. Coarse atomization also impairs combustion efficiency.

It is especially favorable to have under all engine operating conditions drops of various sizes with a mean diameter of the order of 70-100 μ and the following flair angles: in starting the engine, upon ignition, 60-70°, at the end of the starting period, and with low flow rates, 100-120°, and in rated operation, 80-90°.

Jets operating from a single manifold should provide identical fuel flow rates under all operating conditions. The fuel flow rate should also remain stable under variations in fuel temperature and viscosity.

Atomizing nozzles may be of the jet or centrifugal type.

Jet injectors (Fig. 209) use a cylindrical nozzle; the fuel leaves the nozzle with a small flair angle. Jet injectors are not in common use, as they do not provide good fuel atomization and distribution over the chamber volume with acceptable fuel pressures ahead of the injector.

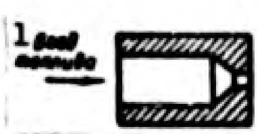

Fig. 209. Jet fuel nozzle. 1) Fuel inlet.

Centrifugal injectors, which may be of the variable or fixed types, are the most widely used.

Fixed centrifugal injectors are used in starting and afterburner fuel-supply systems.

Variable centrifugal injectors are used for the chambers of engines requiring a broad range of fuel flow rates.

2. FIXED CENTRIFUGAL INJECTORS

A fixed centrifugal injector (Fig. 210) has a swirl chamber 4, nozzle 3, and intake duct 2 placed tangentially with respect to the swirl chamber. Fuel is supplied by the pump under pressure to the tangential inlet duct, from which it goes to the swirl chamber. In the swirl chamber, the fuel moves in a spiral trajectory from the periphery

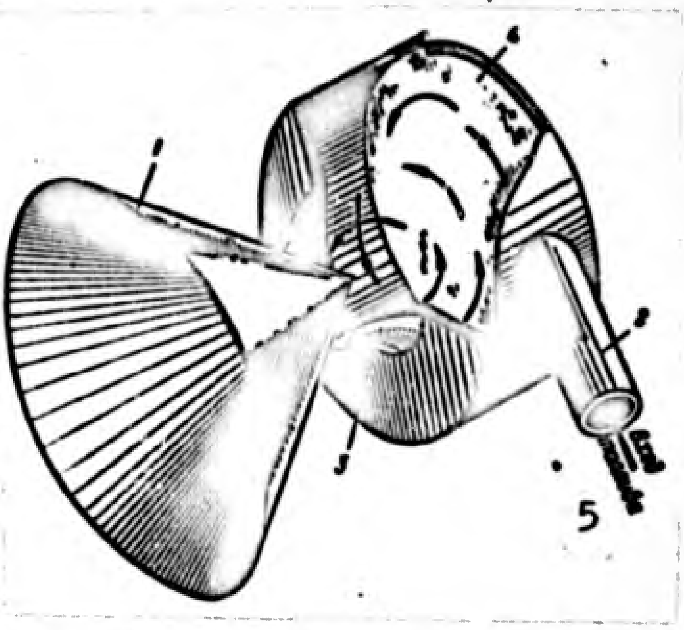

Fig. 210. Operation of fixed centrifugal injector: 1) fuel cone; 2) inlet duct; 3) nozzle; 4) swirl chamber; 5) fuel inlet.

to the center. As it moves to the center, the fuel velocity rises and the pressure drops. As they leave the nozzle, the fuel particles move in straight lines forming a hollow cone 1. The center part of the nozzle and swirl chamber are filled with turbulent air. The boundary of the air vortex lies on the swirl-chamber radius where the fuel pressure drops to a value equaling the pressure of the air in the combustion chamber. The cone of fuel leaving the injector has a vertex angle α, which is also called the flair angle.

The flair angle depends solely on injector geometry, and is constant under all conditions in a fixed-delivery injector.

The injector throughput depends on the geometry and, in particular, on the diameter of the nozzle, radius of the swirl chamber, and total area of the tangential inlet ducts, as well as on the pressure of the fuel ahead of the injector. The rate of flow through the injector in-

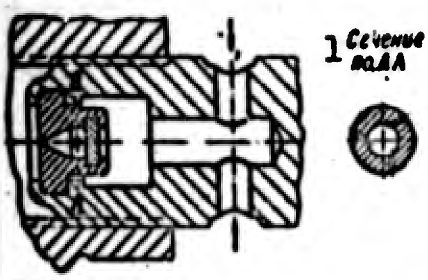

Fig. 211. Fixed centrifugal injector. 1) Section through AA.

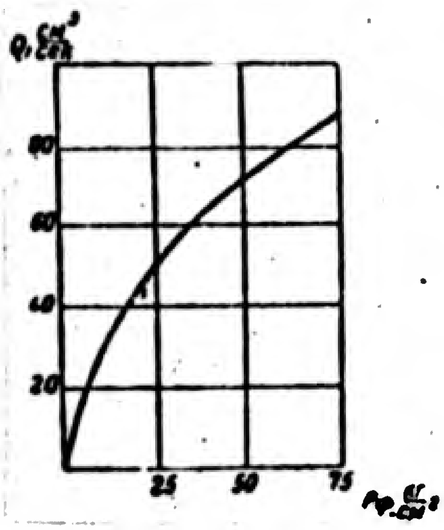

Fig. 212. Flow rate Q through fixed injector as a function of the fuel pressure p_f ahead of the injector.

creases with rising pressure and increasing area of nozzle and inlet ducts. An increase in swirl-chamber radius leads to a reduction in fuel flow rate. This may be explained by the fact that with a large chamber radius, the fuel is whirled more vigorously, fuel pressure falls more abruptly, the air-vortex boundary lies on a larger radius, and the diameter of the nozzle cross section through which the fuel flows is decreased.

The quality of fuel atomization depends on the difference between the fuel pressure at the injector inlet and the air pressure in the combustion chamber. Experiments have shown that with a pressure drop less than 3-4 kg/cm^2, the fuel leaves the jet as a bubble that is not well dispersed into drops. As a result, it becomes necessary to limit the minimum fuel pressure ahead of the injectors.

As the engine operates under various conditions, the amount of fuel needed in the combustion chambers will vary. In a fixed injector, the fuel flow rate is varied by changing the fuel pressure ahead of the injector. In practice, owing to a variation in pressure from the minimum to the maximum permissible values will change the flow rate by a factor of 4 to 5 with fixed injectors.

Such a flow-rate variation is quite satisfactory for starting injectors and afterburner injectors, but is not adequate for main-chamber injectors.

Thus fixed injectors are usually used for starting (Fig. 211) or in afterburner chambers. Centrifugal injectors are very simple in design.

Figure 212 gives the characteristic curve for a fixed centrifugal injector; it shows the way in which fuel flow rate depends on the pressure ahead of the injector.

3. VARIABLE CENTRIFUGAL INJECTORS

The range of fuel flow rates is increased in comparison with fixed injectors by using variable injectors. Variable injectors may be classified as two-nozzle, duplex, and bypass types.

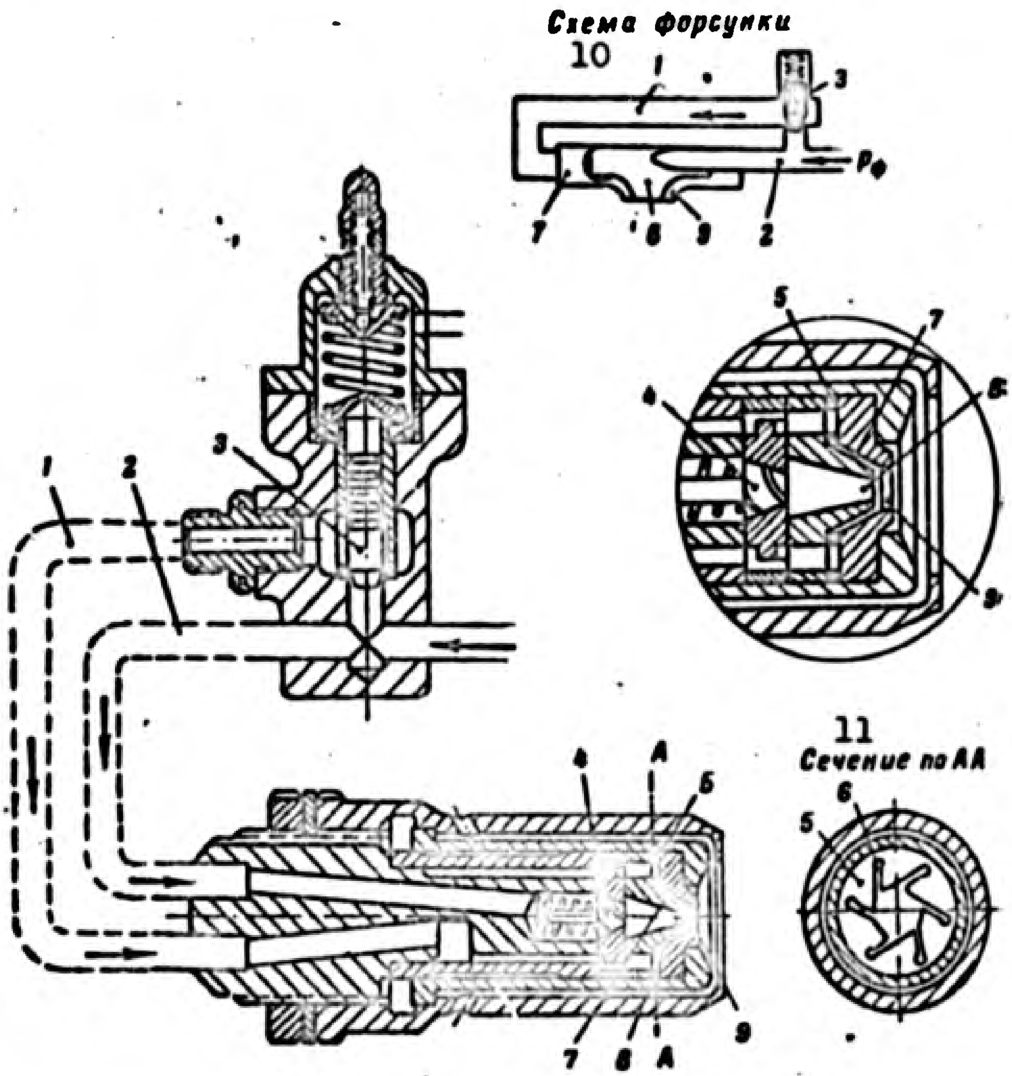

Fig. 213. Two-nozzle injector: 1) main manifold; 2) auxiliary manifold; 3) distributing valve; 4) conical swirler; 5) swirler disk; 6) swirler-disk channels; 7) swirl chamber; 8) auxiliary nozzle; 9) main nozzle; 10) diagram of injector; 11) section through AA.

A two-nozzle injector (Fig. 213) consists of two concentric fixed nozzles. The inside (auxiliary) nozzle is supplied directly from an auxiliary manifold 2 which is connected to the fuel pump. The fuel is swirled in the conical swirler 4, which has a screw thread, and enters the combustion chamber through the small-diameter auxiliary nozzle 8. The fuel flow rate through the auxiliary nozzle changes as the pressure ahead of the injector rises, as with a fixed injector (Fig. 214).

The main manifold 1 (see Fig. 213) is separated from the auxiliary manifold by the distributor 3. The valve spring is so adjusted that the valve opens at some specific pressure, equaling 10-15 kg/cm^2. With the distributor valve open, the fuel flows into the main manifold, through channels 6 in the swirler disk 5 to the swirl chamber 7, and through the large-diameter main nozzle 9 to the combustion chamber. As

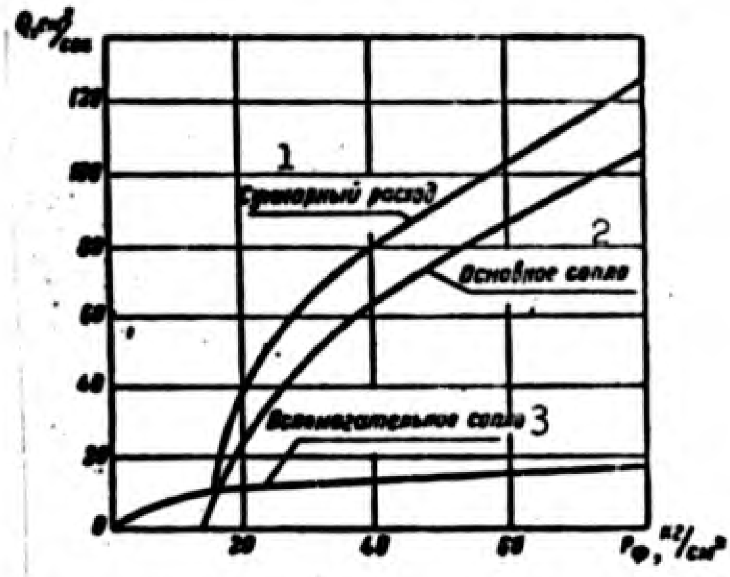

Fig. 214. Fuel flow rate Q as a function of pressure p_f ahead of two-nozzle injector. 1) Total flow rate; 2) main nozzle; 3) auxiliary nozzle.

a result, after the distributor valve opens, fuel is supplied to the combustion chamber through two nozzles. The fuel sprays mix to form one common spray. Figure 214 shows the curve for the total fuel flow rate through the injector. With a two-nozzle injector, it is possible to obtain a ratio of maximum to minimum flow rates of 15 to 20, with acceptable maximum fuel pressures ahead of the injector.

Two-nozzle injectors are used in the main combustion chambers of many types of engine.

<u>The duplex injector</u> (Fig. 215) has one nozzle 4 and two swirl chambers 1 and 3 separated by a diaphragm 2. With distributor valve 7 closed, fuel from the pump flows through the auxiliary manifold 8 and

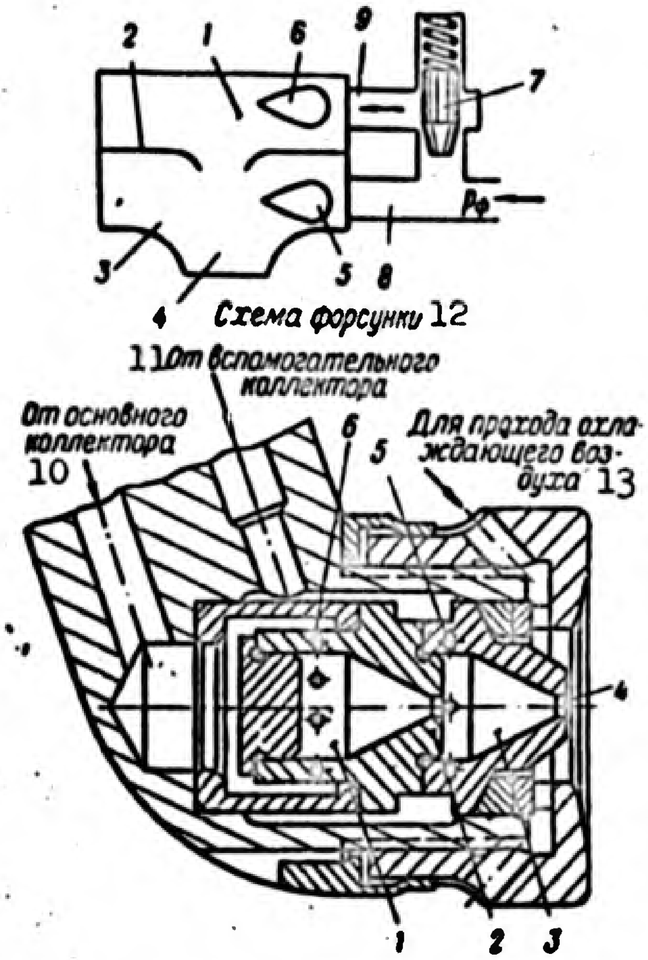

Fig. 215. Duplex injector: 1 and 3) swirl chambers; 2) diaphragm; 4) nozzle; 5 and 6) tangential channels; 7) distributor valve; 8) auxiliary manifold; 9) main manifold; 10) from main manifold; 11) from auxiliary manifold; 12) diagram of injector; 13) to cooling-air supply.

tangential channels 5 into the auxiliary swirl chamber and then to the combustion chamber. When the distributor valve opens, which happens, as in the case of two-nozzle injectors, at pressures of 10-15 kg/cm^2, fuel is also delivered through the main manifold 9, and then through the tangential channel 6 in swirl chamber 1 where they eddy out through a hole in diaphragm 2. The fuel streams are mixed in the chamber 3, and they then emerge as a single spray through nozzle 4. The characteristic curve for such injectors is similar to that shown in Fig. 214. With duplex injectors, a ratio of maximum flow rate to minimum flow

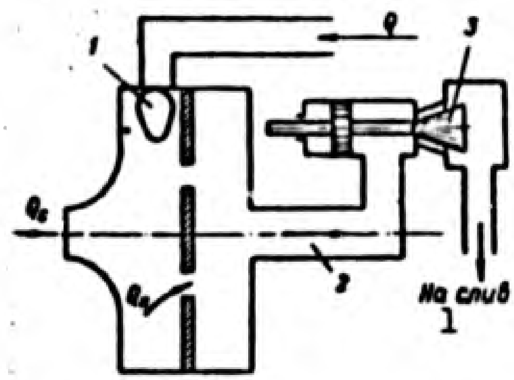

Fig. 216. Bypass injector: 1) tangential channel; 2) bypass channel; 3) distributor valve; Q_B) flow through nozzle; Q_p) bypassed fuel; Q) total fuel flow. 1)

rate of 20 to 30 is obtained with fuel pressures at the pumps not exceeding 50-60 kg/cm^2.

A considerably broader range of flow-rate variation may be obtained with a bypass injector of the sort shown in Fig. 216. Fuel is supplied by the pump at maximum throughput and maximum pressure; it flows through the tangential channels 1 into a swirl chamber. From the swirl chamber, the fuel can flow through the bypass channel 2 and distributor valve 3 back to the pump intake. With the distributor valve completely open, all of the fuel reaching the swirl chamber is returned. As the distributor valve closes, more and more of the fuel is sent to the combustion chamber, and when it is completely closed, all of the fuel reaching the injector is sprayed into the chamber. A feature of such injectors is the possibility of obtaining very low fuel flow rates through the nozzle. In a bypass injector at low flow rates, atomization quality is still satisfactory, as the nozzle operates at maximum differential pressure.

Owing to the possibility of obtaining very low fuel flow rates, the total range of flow-rate variation is 1.5 to 2 times greater than the flow-rate ranges available with other types of injectors.

Part Eight

AUTOMATIC CONTROL SYSTEMS

Chapter 17

GENERAL INFORMATION

1. AUTOMATION OF PRESENT-DAY GAS-TURBINE ENGINES

As we have already mentioned, the modern aircraft gas-turbine engine is provided with fairly complicated automatic control and regulating systems as well as with individual automatic devices. This makes control of the engine considerably easier and increases its operating reliability. Even during initial development of turbojet engines, it was necessary to connect automatic devices into the fuel-supply system. Manual control of a fuel system using a pump driven by the engine rotor through direct manipulation of the pump control devices proved to be impossible owing to the instability of TRD operation over a wide range of rotor speeds. The first attempt to remove this instability involved the incorporation of automatic regulators into the fuel-feed system to ensure steady delivery of fuel to the engine when the control lever was in a fixed position. But even with constant fuel feed, as flight altitude and speed change, engine operation as characterized by rotor speed does not remain constant, and in order to keep the engine from changing operating regime, a special correcting device was added to fuel systems of many engines that automatically change the fuel feed as a function of the air pressure at the compressor inlet; this device is called a <u>barostat</u> or barostatic regulator.

A characteristic feature of a barostat is the fact that it does not maintain the engine speed selected by the pilot with the required degree of accuracy. This made it necessary to connect an automatic overspeed governor into the fuel supply system; the need to maintain maximum speed with high accuracy is dictated by the need to ensure engine operating reliability together with the maximum possible thrust.

The additional need to ensure accurate maintenance of a given speed over the entire operating range of a TRD led to the replacement of the automatic overspeed governor on many engines with a universal speed regulator, and control of the engine in a given range was reduced to changing the setting of this regulator.

To avoid too high a fuel-feed rate to the engine during acceleration from one operating mode to another, in the initial stages of development of TRD, a maximum rate of control-lever advance was established. On the one hand, this complicated engine control and on the other did not sufficiently ensure that an optimum acceleration pattern could be chosen. The danger of engine failure owing to surging or overheating appeared. At present, all TRD are provided with automatic accelerators, which automatically limit the fuel feed rate during engine-rotor acceleration.

An automatic fuel starting device is introduced into the fuel-supply system for starting and for increased reliability. Automatic fuel-pressure limiters are used to prevent overloading of pumps and lines. In high-speed low-altitude flights at low temperature, an extreme engine overload is possible. In this connection, either an automatic limiter for the engine fuel supply is installed, or an air-pressure limiter is introduced at the compressor output. To prevent high-altitude flameouts, and extinction of the flame in an abrupt throttling down, an automatic device is provided that limits the minimum fuel pressure ahead of the injectors.

With constant rotor speed, in a TRD using a fixed-area nozzle, the temperature of the gas ahead of the nozzle diaphragm of the turbine, T^*_3 (see Fig. 1), may assume the maximum permissible value. In this case, it will become necessary to use an automatic limiting device for the temperature T^*_3, which will act on the fuel supply to the engine.

In connection with the substantial increase in the flight-speed range of aircraft using TRD, it is impossible to keep the rotor speed constant owing not only to the variation in T^*_3, but also owing to the substantial variation in compressor efficiency and the danger of surging. The speed-regulation system is therefore sometimes supplemented by an automatic speed-regulator setting corrector that operates on the basis of the temperature T^*_N of the decelerated air at the compressor intake, usually so as to maintain the maximum possible value of the so-called reference speed, which equals the ratio $n/\sqrt{T^*_N}$.

In addition to these automatic devices, the fuel-supply system is occasionally provided with automatic limiters for the dynamic pressure, flight-speed regulators, and devices for automatic ejection of fuel during firing [27].

Compressors of present-day single-rotor TRD are also provided with automatic devices in the systems used to control variable blades in guide-vane assemblies and in air bypass systems, which are used to eliminate surging and increase compressor efficiency where the engine operates in other than design modes.

Automatic devices for bypassing compressor air close or open the bypass ports with the aid of valves or bands, depending on the engine rotor speed or the degree to which the air pressure in the compressor increases. Automatic compressor-blade rotating devices change the angles of blades in guide and straightening vanes, in continuous or stepwise fashion, depending on compressor operating conditions [27].

The possibility of changing the exhaust-nozzle throat area permits both limiting and regulation of the temperature T^*_3, independent of speed, to increase economy of a TRD operating under other than design conditions and to produce maximum possible thrust in all flight regimes. In this case, T^*_3 is regulated by an automatic temperature control.

Two-rotor TRD have a speed regulator on one rotor and usually an automatic speed governor on the other rotor [27].

An engine with an afterburner is provided with an installation for automatic adjustment of the fuel feed rate to the afterburner, in addition to the nozzle-flap control mechanism.

Turboprop engines have, in addition to the automatic devices used with a TRD, automatic regulating devices for the angles at which the propeller blades are set, and various types of automatic governors that ensure reliable operation of propellers and the reduction gearing.

This far from complete list of automatic governors, regulators, and operating components of various GTD control systems shows clearly that the modern GTD is an engine that is automatic to a high degree. The quality of the automatic devices and their operating characteristics to a considerable degree determine engine reliability and the combat effectiveness of the entire flying craft.

2. SOME FUNDAMENTAL DEFINITIONS AND CONCEPTS OF AUTOMATIC DEVICES

The problem of studying automatic devices in general and automatic devices designed for gas-turbine engines in particular is simplified by the fact that the principles of design and operation involved are general in nature. Automatic devices are usually made up of separate elements which may be classified fairly simply as to function, operating principle, construction, form of energy used, type of working fluid used, dynamic and static properties, etc.

Automatic systems are classified on the basis of operating principle into open-loop and closed-loop systems.

An open-loop automatic system is characterized by the fact that the process of system operation does not depend directly on the results of its action. An example of such a system is the automatic control system for the exhaust-hozzle flaps; this system opens or closes the flaps when the afterburner is on or off. Here the initial source of the action is the force applied by the human controlling the engine [27].

An automatic closed-loop system is characterized by the fact that the process of system operation depends directly on the result of the action taken. Such systems include automatic computing devices and automatic control systems.

An automatic control system (SAR) can automatically, without human intervention, maintain a constant prescribed value of any controlled variable (parameter) or can vary it as desired.

In the first case, we have an automatic stabilization system. It is clear that, for example, a control system that maintains a prescribed engine speed constant is an automatic stabilization system.

If the system does not maintain the controlled variable constant, but changes it in accordance with some prescribed law, the system is called a program control system. The law governing the variation in the controlled variable may be given as a function of time (time-pattern control system) or as a function of some other quantity. Thus, for example, engine-rotor speed may be regulated in accordance with the prescribed law, depending on the temperature T^*_n at the compressor intake.

There are also the so-called automatic servosystems, in which a given controlled variable changes continuously and arbitrarily with time. The servosystem operates so that the controlled variable exactly follows the prescribed quantity. From this it follows that the servo-system differs from the program control system only in the way in which the controlled variable is specified.

An automatic control system consists of the system under control (the regulated object) and the regulator.

The controlled system is the object in which the process to be controlled occurs. Thus, for example, a TRD or TVD will be such a controlled system.

The variables characterizing the process taking place in the controlled system which are at the same time themselves regulated (for example, held constant or varied in accordance with a prescribed law), are called the controlled variables or controlled parameters (in an engine, for example, this will be the rotor speed, temperature T^*_3, degree of compressor temperature rise, oil pressure in lubrication system, etc.). The factor which forms the basis for acting on the process occurring in the controlled system so as to maintain the prescribed value of the controlled parameter is called the regulating factor (in a TRD, for example, the regulating factor is the fuel feed rate).

The process occurring in the controlled system may be affected as well by disturbances (for a TRD, these will be variations in ambient pressure and temperature, flight speed, a change in the geometric parameters of the engine blading, etc.). The task of the control system is to maintain the controlled parameter constant or to vary it in desired fashion in the presence of disturbances.

The regulator is an automatic device that acts through the regulating factor on the controlled system during the control process so as to make the controlled parameter equal to the prescribed value.

The regulator automatically eliminates the consequences of disturbances as they affect the prescribed value of the controlled parameter.

There are two fundamental principles of control: control based on the deviation of the controlled parameter from the prescribed value, and control based on the disturbance.

In control based on deviation the so-called sensing element of the regulator directly measures the controlled parameter, compares it with the prescribed value, and in the presence of a deviation either directly or through intermediate devices of the regulator causes the final control element to move so as to eliminate the deviation that has appeared. The deviation of the controlled parameter from the given value plays the role of the input signal to the regulator, while the movement of the final control element is the output signal. Thus, for example, in a speed regulator, the input signal is the departure of the speed from the given value, while the output signal is the motion of the fuel-system final control element that changes the rate at which fuel is fed to the engine.

It should be noted that the output signal of a regulator acts as the input signal for the controlled process, while the output signal from the controlled process is the input signal to the regulator. Thus, a control system operating on the principle of deviation is a closed-loop system and comes into play as soon as the controlled parameter departs from the prescribed value.

The principle of deviation is also called the principle of I.I. Polzunov; this scientist was the first (1765) to use this principle in an automatic water-level regulator for a steam boiler.

Although during the control process, the controlled parameter will deviate from the prescribed value, after the control process has terminated (under static conditions), such deviations may be reduced to arbitrarily small values. The high control accuracy yielded has brought the principle of deviation into wide use.

In control based on the disturbance, the regulator sensing element does not measure the controlled parameter, but the disturbances themselves; the regulator then acts directly or through intermediate devices on the final control element so as to compensate for the disturbances and maintain the prescribed value of the controlled parameter. Thus, in the case given the regulator input signal is a disturbing influence on the controlled system, and we have an open-loop control system. This improves control-system dynamic properties, but in practice, it is very difficult to provide the required static accuracy of control with such a system owing to the complex nature of a disturbance and possible variations in the parameters of the regulator itself. This principle is therefore used only where a very high degree of control accuracy is not demanded. It is also used in combination with the principle of deviation.

The value of the controlled parameter is specified either by the human controlling the engine (controlled system), or by a computer with the aid of the regulator adjustment mechanism.

All automatic control systems are classified as direct or indirect systems.

A direct automatic control system is a system in which the sensing element acts directly on the final control element with no additional source of energy and, consequently, with no intermediate elements.

An indirect automatic control system is a system in which the sensing element does not act directly on the final control element, but operates through special amplifying and conversion elements to which energy is supplied from an outside source. Such a system is employed, as a rule, where the energy of the sensing element is insufficient to drive the final control element or where it is necessary to obtain system dynamic and static properties that cannot be obtained with a direct system.

All automatic control systems may be classified on the basis of operation into continuous and intermittent systems.

A continuous system is a system in which a continuous process in all control-system elements corresponds to a continuous variation in the controlled variable.

An intermittent system is a system in which a continuous variation in the input signal corresponds to a discontinuous application of the output quantity in at least one control-system element. Intermittent systems include relay and pulse systems.

An automatic control system is characterized by its dynamic and static properties.

The dynamic properties are the system properties exhibited during the process of regulation and characterized by the magnitude and duration of the deviation of the controlled variable from the prescribed value.

The static errors of a system, conversely, appear after the control process has been concluded and are evaluated in terms of the deviation of the value of the controlled parameter from the prescribed value after the control process initiated by a disturbance has been concluded.

In order to obtain the required dynamic and static properties, we must not only design the sensing element and amplifying devices appropriately, but must also introduce additional special elements with suitable intercoupling.

Here we must concern ourselves with certain general considerations bearing directly on the subsequent discussion of automatic control systems for gas-turbine engines.

The state of an operating engine is called its operating regime. The operating regime of an engine is characterized by several parameters of the gasdynamical process taking place in the engine, which in turn determines the thermal and dynamic loads on engine structural elements.

Under given external conditions, the parameters of the gasdynamical process are determined by the dimensions, shape, and relative positions of the engine structural elements forming the air-gas flow area and, in addition, by the fuel supply rate (flow rate) and the size of the load taken off the engine rotor. The parameters of the gasdynamical process are interdependent. For a TRD in which the geometric parameters of the gas-air flow area remain unchanged (called a TRD with fixed geometry) and constant rotor load (due to the drive furnished for various units), the parameters of the gasdynamical process will be determined solely by the fuel feed rate G_t. In this case, an aircraft TRD represents a system with one degree of freedom; specification of one of the process parameters will determine all the rest. Regulation of the engine rotor regime reduces to controlling one of its parameters, which in this case is taken as the controlled parameter. Here the fuel feed rate acts as the controlling factor which is varied in order to vary the controlled parameter and the entire operating regime of the engine. An engine with one degree of freedom has one controlled parameter and one controlling factor.

A TRD with variable geometry (with rotatable guide vanes and compressor stator blades, rotatable turbine nozzle blades, variable-area exhaust nozzle, etc.) has several degrees of freedom, and the gasdynamical process in such a TRD will be determined not only by the fuel feed rate, but also by the variable geometric quantities (angles at which rotatable blades are set, nozzle throat area, etc.), which act as additional (in the case given, geometric) control factors. For a TVD, in addition, the angle at which the blades of the VISh [variable-pitch propeller] are set is a controlling factor.

In a variable-geometry engine, as the number of controlling factors goes up, the number of independent controlled parameters also increases. In controlling an engine, the specification of engine operating regime reduces to specification of several controlled parameters. In this case, the engine represents a system with many degrees of freedom, equal to the number of independent controlled parameters and, correspondingly, to the number of controlling factors. Here, naturally, engine control and design will be more complicated, but it then becomes possible to improve its characteristics.

When external conditions change, the engine is subject to the action of disturbances, which at a given value cause the controlling factors to change the controlled parameters, and together with them the entire complex of parameters characterizing the gasdynamical process and engine thermal and dynamic loads. It thus becomes necessary to provide for engine regulation (manually or automatically) in accordance with one (with one controlling factor) or several parameters (corresponding to the number of controlling factors). The uncontrolled parameters take on specific values that depend on the controlled parameters and the external conditions.

Since with a variation in external conditions, the relationship among the engine operating-regime parameters will vary, for a prescribed value of a controlled parameter, the remaining parameters may take on values that are impermissible (for reasons of strength, engine operating reliability, process stability, etc.). It may thus become necessary to limit one or several of these parameters. When any of the engine process parameters are limited, the value of the controlled parameter will vary owing to the relationships among the engine parameters. Parameters on which a limitation is imposed are called restricted parameters.

For each actual engine type, the controlled parameters are selected and a control program specified; by the control program, we mean the law governing the variation in the controlled parameters under a variation in external conditions, or where the amount of thrust or power developed by the engine is to be controlled (throttling control program). In practice, it sometimes becomes necessary to specify a program for varying the controlled parameter as a function of time (time programing) or as a function of some other parameter (parameter programing).

With the engine running wide open, the control program is considered to be optimum if under given external conditions the engine develops maximum thrust. A throttling control program is considered optimum if when this program is operative the minimum possible specific fuel consumption is realized.

The control program in many ways determines the efficiency of engine operation. Control programs are selected on the basis of careful study of engine characteristics and construction.

Where there is automatic regulation of some engine parameter, a regulator is always installed. The regulator and engine form a control system. An engine may have several automatic-control systems, corresponding to the number of controlled parameters (controlling factors). Automatic-control system nomenclature is based on the controlled parameter (automatic engine-speed control system, control system for temperature ahead of turbine nozzle-box assembly, etc.) or on the controlling factor (fuel feed-rate control system, exhaust-nozzle control system, etc.).

An automatic control system is designed with allowance for the static-accuracy requirements for keeping to the given control program with acceptable dynamic properties. In the most general case the creation of some given automatic control system involves the following basic stages:

– a study of the static characteristics of the engine (exhibited in steady regimes), making it possible to select controlled parameters for the corresponding controlling factors, and to designate the control program;

– determination of the dynamic engine properties (exhibited in transient regimes), or, in other words, study of the engine as a controlled process;

– analysis of possible methods of implementing the TRD control program chosen, with given dynamic properties;

– development of tactical and technical specifications for the control systems pertaining to statical and dynamical control accuracy, operating reliability, size, weight, structural simplicity, manufacturing technology, etc.;

– synthesis of the control system, investigation of the statical and dynamical properties of system elements and of the system as a whole;

– design and manufacture of control-system units;

– carrying out of experimental test-stand and flight studies and final development of the control system.

At the present time, the creation of new control systems is made somewhat easier by the existence of a large body of practical experience in the design, manufacture, and operation of many gas-turbine engines manufactured on a production basis. The control-system designer has at his disposal excellent methods from the modern theory of automatic control and from experimental studies. In the majority of cases, difficulties in control-system design are caused by a lack of knowledge of the actual statical and dynamical properties of newly designed gas-turbine engines, which have a broad range of operating regimes.

Chapter 18

CONTROL PROGRAMS FOR TURBOJET ENGINES. REQUIRED AND AVAILABLE FUEL FLOW RATES. OPERATING STABILITY OF TURBOJET ENGINES

1. CONTROL PROGRAMS FOR TURBOJET ENGINES

Control programs for TRD [turbojet engines] determine the nature of the variation in the controlled parameters under various engine operating regimes and aircraft flight regimes so as to produce the desired values of thrust and specific fuel consumption.

A control program must provide:

– the possibility of very simple technical implementation of the selected program by an automatic control system;

– the possibility of the most efficient utilization of the engine on the basis of the gas temperature ahead of the turbine nozzle-box assembly and engine-rotor speed, with a simultaneous guarantee of reliable operation in terms of the conditions governing the thermal and dynamic loads on parts;

– surge-free operation of the engine under substantial variations in external conditions or rapid transition from one engine operating regime to another.

There are two basic types of control programs for single-rotor and two-rotor TRD:

– control programs that do not require a change in the geometry of the engine air-gas flow area;

– control programs whose implementation requires variation of the geometry of the engine air-gas flow area.

Programs of the first type which do not require changes in the flow areas of engine flow passages permit both the engine and its control system to be structurally simpler.

The utilization of programs of the second type, whose implementation requires variation in flow areas of the flow passage, for example, in the turbine nozzle-box assembly area, the nozzle exhaust area, etc., inevitably entail structural complication of the engine.

Control programs for single-rotor TRD with fixed flow-passage geometry. In single-rotor TRD with fixed flow-passage geometry, when the external conditions remain constant, i.e., the flight altitude and speed remain the same, all engine parameters are interrelated and specified when one of them is given, such as the engine-rotor speed, gas temperature ahead of the turbine nozzle-box assembly, engine thrust, etc.

The parameter that permits the engine to be controlled most simply from the technical viewpoint is chosen as the controlled parameter. This condition is best satisfied by the engine rotor speed, which is explained by the following reasons.

The rotor speed uniquely determines the engine operating regime and characterizes its dynamic loading. Very accurate speed measurements may be made with the aid of fast-response instruments.

It is difficult to measure, for example, the gas temperature ahead of the turbine nozzle-box assembly with the required accuracy owing to the nonuniformity of the temperature field ahead of the turbine and the lack of fast-responding measuring devices that will operate reliably at high temperatures. Inaccurate temperature measurement may lead to large variations in developed engine thrust. In addition, under given external conditions, the gas temperature ahead of the turbine nozzle-box assembly, as we shall show, does not uniquely determine the engine operating regime or the thrust developed by it.

The only possible controlling factor for an engine with fixed geometry is the fuel-feed rate.

Figure 217 shows the way in which the fuel feed rate, gas temperature ahead of the turbine nozzle-box assembly, thrust, and specific fuel consumption depend on the rotor speed of an engine with fixed flow-passage geometry under constant external conditions. To each value of the controlling factor there correspond specific values of the controlled parameter – the rotor speed – as well as of the gas temperature ahead of the turbine nozzle-box assembly, specific fuel consumption, and thrust.

As we can see from the figure, a prescribed gas temperature ahead of the nozzle-box assembly, for example, T^*_{31}, will not uniquely determine the remaining parameters since it corresponds to operating regimes at points a and c with different speeds.

The thrust will drop below its maximum value owing to a simultaneous reduction in both the gas temperature ahead of the turbine nozzle-box assembly and the speed. This aids in reducing the thermal and dynamic loads on engine parts in low-thrust regimes, and enables the engine to operate over long periods of time.

A drawback in engines using fixed flow-passage geometry is the impossibility in most cases of obtaining a prescribed thrust with an optimum relationship among the gasdynamical parameters of the engine, which corresponds to the minimum possible specific fuel consumption.

As an example, we have shown in Fig. 217 (by the dashed curves) the possible relationships between thrust and specific fuel consumption that may be obtained when the nozzle throat area is varied, provided the optimum relationship among gasdynamical parameters is maintained. In such regimes, it is possible to gain a very substantial advantage with respect to specific fuel consumption ΔC_p over an engine using fixed geometry.

When the external conditions change, the relationship among engine parameters will vary. For the best engine efficiency with respect to thrust and to prevent dynamic overloading of parts, it is desirable to keep engine rotor speed constant under all flight conditions.

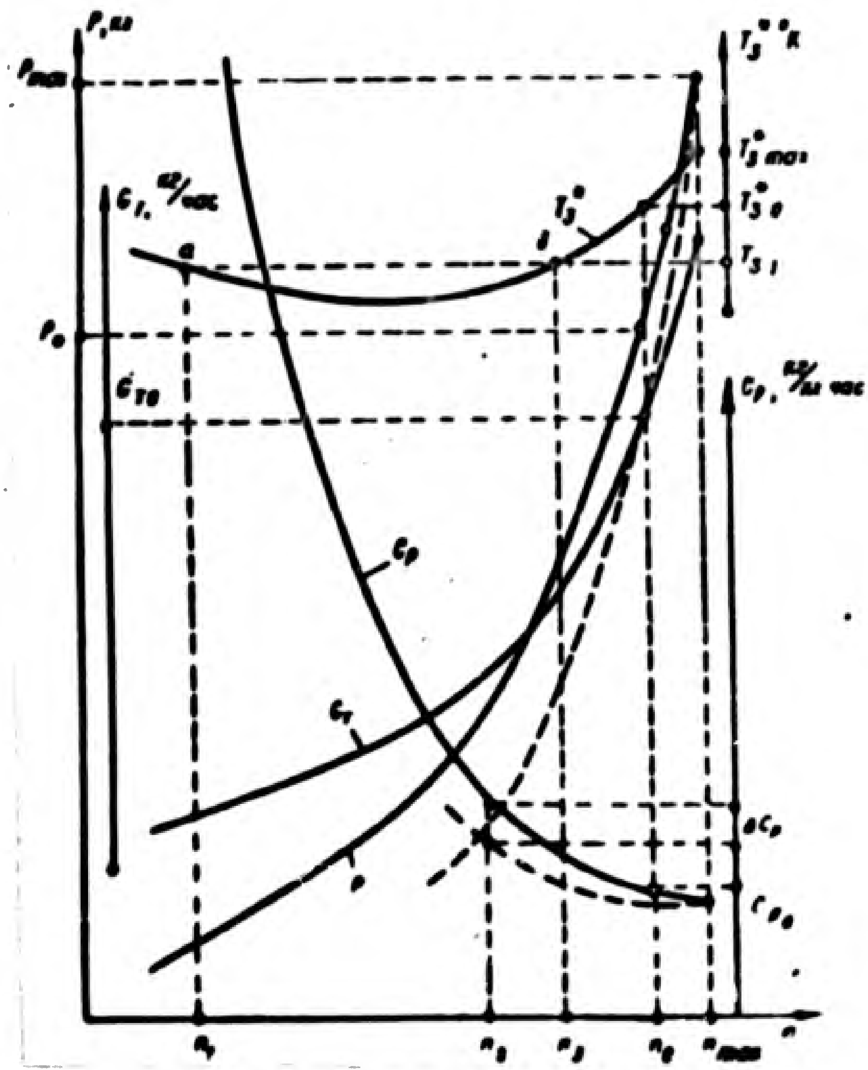

Fig. 217. Thrust P, specific fuel consumption C_p, fuel flow rate G_t, and gas temperature T^*_3 ahead of turbine nozzle-box assembly as functions of rotor speed n of TRD with fixed flow-passage geometry, constant external conditions.

If the rotor speed drops when the external conditions vary, the result will be that the engine will not operate at maximum thrust efficiency. On the other hand, an increase in the speed, especially with the engine running wide open or nearly wide open, may lead to impermissible (from the viewpoint of strength) increases in dynamic loads on parts.

The practical possibility of using a control program with constant engine rotor speed is determined by engine efficiency with respect to the gas temperature ahead of the turbine nozzle-box assembly, the degree to which absence of engine thermal-stress overloading is guaranteed, and the degree to which surge operation is precluded.

The variation in gas temperature ahead of the turbine nozzle-box assembly is affected in present-day engines only by the stagnation temperature at the engine intake. The stagnation temperature at the engine intake is determined by the ambient temperature, which depends on the flight altitude as well as on the flight Mach number, which depends on the flight speed. Thus, variations in flight speed and altitude, which affect the stagnation temperature at the engine intake, cause the gas temperature ahead of the nozzle-box assembly to vary.

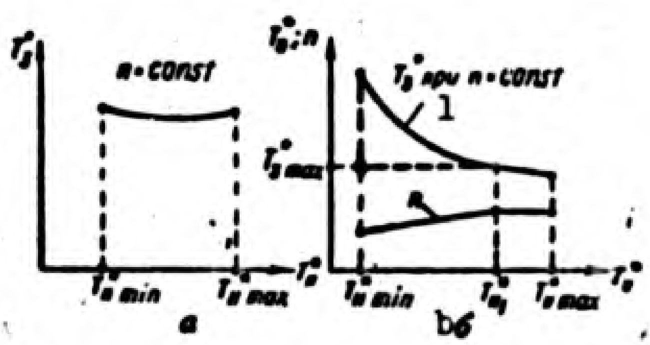

Fig. 218. Gas temperature T^*_3 ahead of turbine nozzle-box assembly and rotor speed n as functions of stagnation temperature T^*_N at engine intake. T^*_{3max}) Maximum permissible gas temperature; T^*_{Nmax} and T^*_{Nmin}) maximum and minimum possible air temperatures at engine intake, respectively; T^*_{N1}) value of T^*_N below which speed is restricted. 1) For.

At constant flight altitude, the stagnation temperature at the engine intake will rise with increasing flight speed. At constant flight speed, the stagnation temperature at the engine intake will drop with increasing flight altitude to a height H = 11 km, and then will remain constant.

The possible variation in the relationship between gas temperature ahead of the turbine nozzle-box assembly and the stagnation temperature at the engine intake is shown in Fig. 218 for present-day engines. In the first case (Fig. 218a), a change in external conditions with constant engine speed leads to a very slight variation in the gas temperature ahead of the turbine nozzle-box assembly. In practice, this means that it is possible to hold rotor speed constant for all flight speeds and altitudes with engines characterized by such a variation in gas temperature ahead of the turbine nozzle-box assembly under various flight conditions. Here the engine will be operating as efficiently as possible with respect to both speed and gas temperature ahead of turbine nozzle-box assembly.

In the second case, shown in Fig. 218b, when the speed is held constant, we observe a quite considerable variation in gas temperature ahead of the turbine nozzle-box assembly with variations in flight regime. In certain flight regimes, the gas temperature ahead of the turbine nozzle-box assembly may exceed the maximum permissible value allowed by mechanical-element strength conditions. This indicates that for such engines it is not possible to use a control program requiring constant rotor speed in all flight regimes.

In order to keep the gas temperature ahead of the nozzle-box assembly from increasing above the maximum permissible value, it is necessary to reduce the fuel-feed rate. A reduction in fuel-feed rate leads to a drop in gas temperature and at the same time to reduced speed.

If it is desirable to prevent the gas temperature ahead of the turbine nozzle-box assembly from exceeding a maximum permissible value, as shown by the dashed line in Fig. 218b, it is necessary in these flight regimes to decrease rotor speed. As we can see from the figure, rotor speed is related in definite fashion to the stagnation temperature at the engine intake.

As a consequence, for certain engines the conditions ensuring reliable operation and full efficiency with respect to the gas temperature ahead of the turbine nozzle-box assembly, it is necessary to use a combination control program: in flight regimes corresponding to high stagnation temperatures at the engine intake, the rotor speed is held constant and the gas temperature ahead of the turbine nozzle-box assembly will vary slightly, and in the remaining flight regimes, the gas temperature may be held constant while the rotor speed will drop.

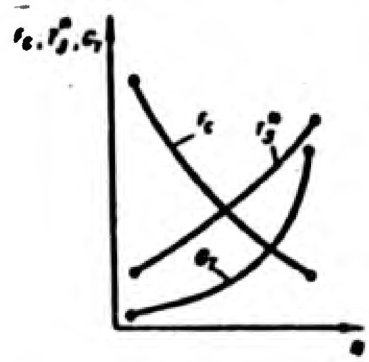

Fig. 219. Nozzle throat area F_s, gas temperature T^*_3 ahead of turbine nozzle-box assembly, and fuel flow rate G_t as functions of rotor speed n with optimum TRD control program.

The utilization of a combination control program complicates the automatic control system for the engine.

Control programs for single-rotor TRD with variable flow-passage geometry. Regulation with the aid of just one controlling factor – the fuel flow rate – as we have already mentioned in many cases does not permit full utilization of engine potential. Additional control factors may be gained by varying the areas of flow-passage cross sections, for example, the turbine nozzle-assembly area and the exhaust-nozzle throat area. With a convergent exhaust nozzle, the throat area is the same as the exhaust area.

The least complication is introduced into engine construction and the control system when we vary the exhaust-nozzle throat area, and this is most frequently done in practice.

Nozzle throat-area control provides the following advantages:

– optimum or nearly optimum variation in specific fuel consumption under changing flight regimes;

– reduced power drawn by engine starter and reduced gas temperature ahead of turbine nozzle-box assembly during starting and in operation at low throttle settings;

– maximum engine efficiency with respect to gas temperature ahead of turbine nozzle-box assembly in all flight regimes.

Figure 219 shows the variations in fuel consumption, gas temperature ahead of turbine nozzle-box assembly, and nozzle throat area with speed, corresponding to an optimum control program with external conditions remaining unchanged.

As we can see from the figure, a quite definite nozzle throat area F_s corresponds to each speed. When the external conditions change, the optimum control program will also change.

In the majority of designs in use, simpler control programs have been selected.

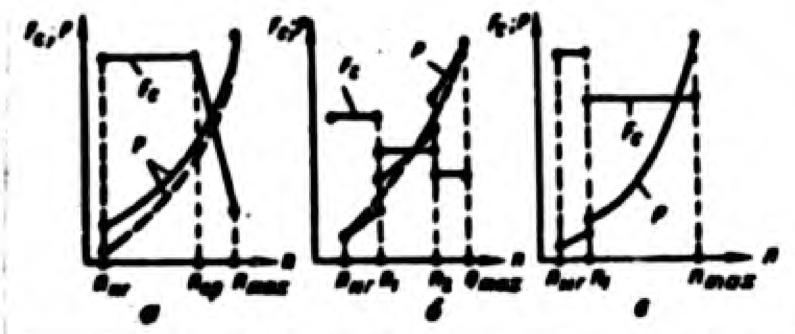

Fig. 220. Thrust P and nozzle throat area F_s as functions of rotor speed n with different control programs (the variation in thrust, following an optimum law, is shown by the dashed line). n_{kr}) Rotor speed at cruising speed; n_{mg} and n_{max}) rotor speed at low throttle setting and full throttle, respectively.

Special cases of variable-area nozzle application are shown in Fig. 220.

For the thrust to vary in accordance with a nearly optimum law, the control program shown in Fig. 220a is used in the speed range from cruising to maximum speeds. In this case, the nozzle throat area varies only over this speed range.

As is shown in Fig. 220b, a stepwise change in nozzle area can provide a nearly optimum relationship of parameters over the entire speed range.

A stepwise change in nozzle area may be used, for example, to ease engine starting conditions (Fig. 220c). The nozzle area increases in regimes close to idle running speed. This increases the pressure drop at the turbine, eases engine starting, and at the same time somewhat reduces the gas temperature ahead of the turbine in steady operation at low throttle settings. In all other regimes, the nozzle area remains unchanged and corresponds to the design wide-open regime.

As we have already shown, where a combination control program is used, in many flight regimes engines with fixed flow-passage geometry require that the speed be reduced in order to limit the rise in gas temperature ahead of the turbine nozzle-box assembly. Here the engine operates with considerable inefficiency with respect to speed.

Engine speed inefficiency may be reduced by changing the nozzle area while the gas temperature ahead of the turbine nozzle-box assembly is held constant.

Features of control programs for two-rotor TRD [2, 27]. A two-rotor TRD has high- and low-pressure compressors each of which can turn at its own speed. With the flow-passage geometry fixed, the controlling factor in a two-rotor engine is the fuel feed rate alone. It is thus possible to control just one parameter, while the others will be established by the interaction of the engine process parameters.

Just as with the single-rotor TRD, the controlled parameter is not the gas temperature ahead of the turbine nozzle-box assembly, but the rotor speed of the high- or low-pressure compressor.

If the speed of the low-pressure rotor is taken as the controlled

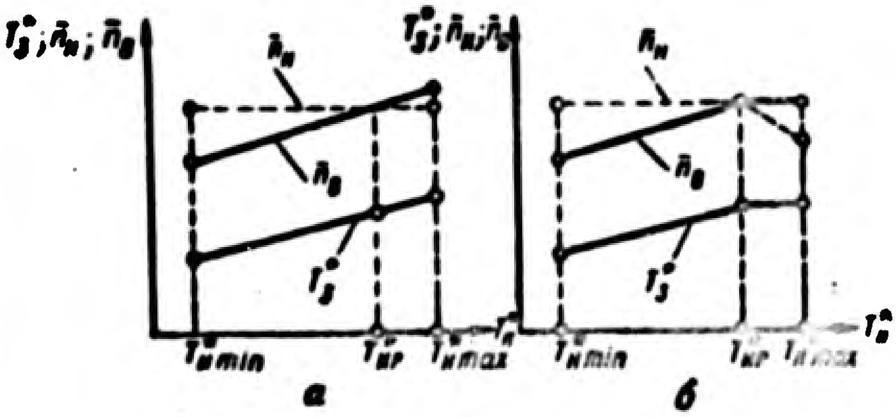

Fig. 221. Gas temperature T^*_3, relative speeds of high-pressure rotor $\overline{n}_n = n_n/n_{n\ max}$, and of high-pressure rotor $\overline{n}_v = n_v/n_{v\ max}$ as functions of air temperature T^*_N. a) With $\overline{n}_n$ constant; b) with the restriction $\overline{n}_v = 1$; T^*_{Nr}) value of temperature T^*_N for which $\overline{n}_n = \overline{n}_v = 1$.

parameter and is maintained constant under changes in flight regime, the gas temperature ahead of the turbine nozzle-box assembly and the high-pressure rotor speed will not be constant.

If the flight speed increases or flight altitude decreases (where H < 11 km), which corresponds to an increase in the stagnation temperature at the engine intake, there will be an increase in the angle of attack at the blades of the low-pressure compressor stages, and more driving power will be needed. Consequently, in order to hold the speed of the low-pressure rotor constant, it is necessary to increase the gas temperature ahead of the turbine nozzle-box assembly (Fig. 221a). At the same time, there will be an increase in the speed of the high-pressure compressor rotor.

The speed of the high-pressure rotor and the gas temperature ahead of the turbine nozzle-box assembly may rise above permissible limits. Thus, when, for example, the maximum permissible rotor speed is reached for the high-pressure compressor, the rate of fuel flow to the engine is limited, resulting in a drop in the gas temperature ahead of the turbine nozzle-box assembly and a reduction in the speed of the low-pressure rotor (Fig. 221b).

If the speed of the high-pressure compressor rotor remains constant under a change in flight regime, this will cause a change in the gas temperature ahead of the turbine nozzle-box assembly and in the speed of the low-pressure rotor.

When the stagnation temperature increases at the engine intake, the temperature of the air at the high-pressure compressor intake will rise, and if its speed is held constant, the drive power required will decrease and, consequently, so will the gas temperature ahead of the turbine nozzle-box assembly, which will in turn cause a drop in low-pressure compressor rotor speed (Fig. 222a). In other words, at low stagnation temperatures at the engine intake; the low-pressure compressor rotor speed and the gas temperature may rise above permissible values.

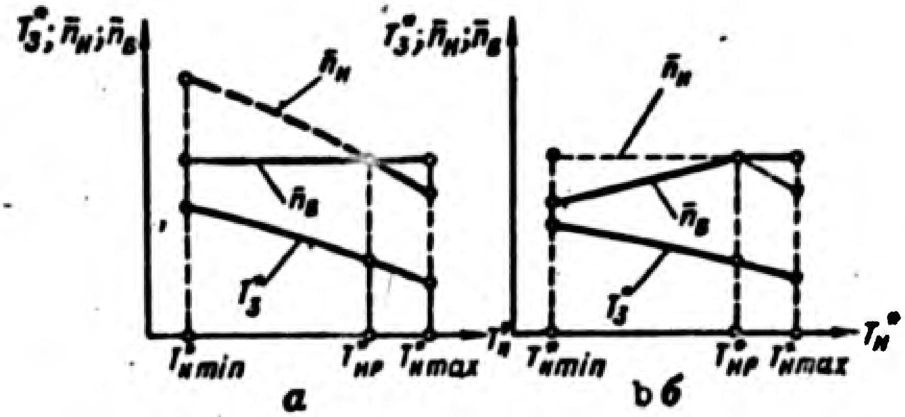

Fig. 222. Gas temperature T^*_3 and relative speeds of low-pressure rotor $\bar{n}_n$ and high-pressure rotor $\bar{n}_v$ as functions of temperature T^*_N. a) With $\bar{n}_v$ constant; b) with restriction $\bar{n}_n = 1$.

If the low-pressure compressor rotor speed is restricted at low air temperatures by reducing the fuel flow rate, the high-pressure compressor rotor speed and gas temperature will change as shown in Fig. 222b.

Thus, where such control programs are in use, full engine efficiency with respect to gas temperature and speed will not be developed in a two-rotor engine in individual operating regimes.

By using a variable-area nozzle, it is possible to obtain an optimum relationship between the speeds of both rotors when the flight regime changes, i.e., the engine can be used more efficiently with respect to speed. Here the speed of the high-pressure rotor is usually controlled by the fuel feed rate, and the speed of the low-pressure rotor by a change in nozzle-area variation, since variation of the pressure drop in turbines primarily affects the variation in turbine speed for the low-pressure rotor. The gas temperature ahead of the nozzle-box assembly is not regulated at all, and its increase may be restricted by changing the fuel feed rate for different regimes.

Fully efficient utilization of an engine with respect to speed and gas temperature is possible only by the introduction of a third controlling factor, for example, by varying the area of the turbine nozzle-box assembly or the angular setting of the compressor guide-vane assembly, which is not done in practice owing to the considerable complication in engine structure as the control system that results.

2. REQUIRED FUEL FLOW RATES

In the most common TRD control program, i.e., where the geometry of the flow passage remains fixed, the required engine operating regime (thrust, specific fuel consumption) is established in accordance with speed by changing the fuel feed rate. Figure 223 shows the curve of required engine fuel feed rates where a fixed-area nozzle is used; the flight regime is constant. This curve is the line of steady engine operating regimes.

Steady engine operation is the name given to the regime in which the rotor speed is constant. Here the torque developed by the engine turbine equals the torque absorbed by the compressor, and the rate of fuel flow to the combustion chamber of the engine equals the fuel flow rate required for operation at the given speed.

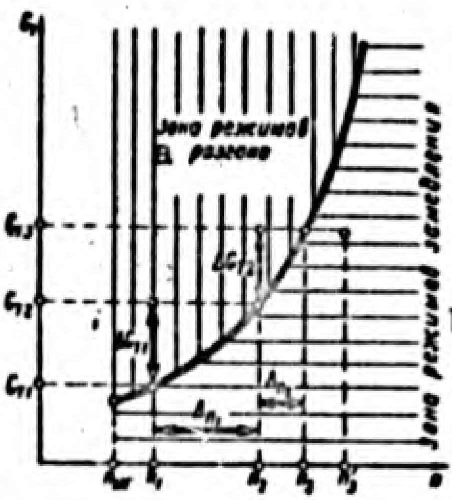

Fig. 223. Influence of fuel feed rate on TRD rotor speed. G_t) Fuel feed rate; n) rotor speed; $\Delta G_{t1} = \Delta G_{t2}$) fuel feed-rate increments; Δn_1 and Δn_2) speed increments. a) Acceleration zone; b) deceleration zone.

Thus, for example, for a speed n_1 and fuel flow rate G_{t1} (Fig. 223), the engine will be in steady operation, with the turbine torque and compressor torque equal. If at this same speed, the fuel flow rate is increased by an amount ΔG_{t1} in comparison with that needed, the increase in gas temperature ahead of the turbine nozzle-box assembly will result in an increase in turbine power and, consequently, the turbine torque will become larger than the compressor torque. As a result, the engine will go into unsteady operation. The excess turbine torque will cause the engine rotor to accelerate. The compressor torque will then increase. Finally, at some speed n_2, the turbine torque will become equal to the compressor torque, and the fuel flow rate G_{t2} will correspond to a new steady operating regime. The higher the excess fuel flow rate at a given speed, the greater will be the excess torque at the turbine and the more rapidly the engine rotor will accelerate to the new steady speed, which will also be higher.

If at speed n'_3 the fuel flow rate is G_{t3}, i.e., it is less than needed for a given engine operating regime, the gas temperature ahead of the nozzle-box assembly will drop, resulting in decreased turbine torque, which in this case will become less than the compressor torque. This causes the engine rotor to decelerate, reducing the torque needed to turn the compressor and, finally, at a speed n_3 below the initial speed, a new engine operating regime will be established, in which the fuel flow rate will equal the required rate.

Consequently, the curve of required fuel flow rates represents a boundary dividing the engine-rotor acceleration zone from the engine-rotor deceleration zone. If at a given rotor speed the fuel flow rate lies above the line of steady operation, the engine rotor will speed up. If the fuel flow rate lies below the steady-operation line, the rotor will slow down. This property of the required fuel flow-rate curve is normally used in analyzing the stability of an engine-operation regime.

Let, for example, the fuel flow rate increase by an amount ΔG_{t1} in a steady operating regime with rotor speed n_1. Then, as we have already discussed, the engine will accelerate to a new steady regime at speed n_2, i.e., the speed will increase by an amount Δn_1. As we can see from Fig. 223, if the fuel supply rate is increased by ΔG_{t2} in a steady regime for which the speed n_2 is greater than n_1, there will be a further acceleration and engine speed will increase by Δn_2. The quantity Δn_2 will be less than Δn_1. It follows from this that an identical excess fuel supply rate will cause greater engine-rotor acceleration at low speeds than at high speeds. This also means that in order to increase rotor speed by a given amount, less of an excess fuel supply rate is needed at low speeds than at high speeds.

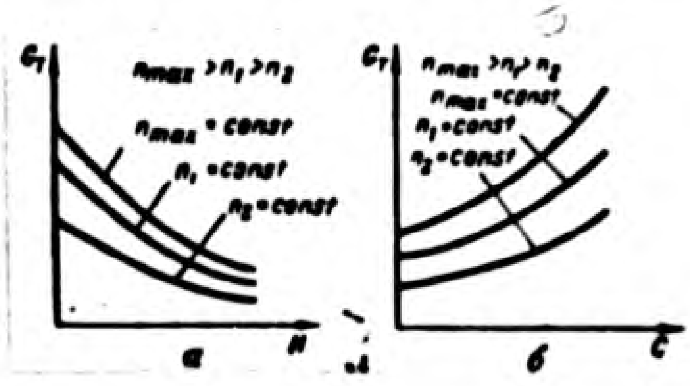

Fig. 224. Required fuel flow rate G_t at various rotor speeds n. a) As a function of flight altitude H; b) as a function of flight speed C.

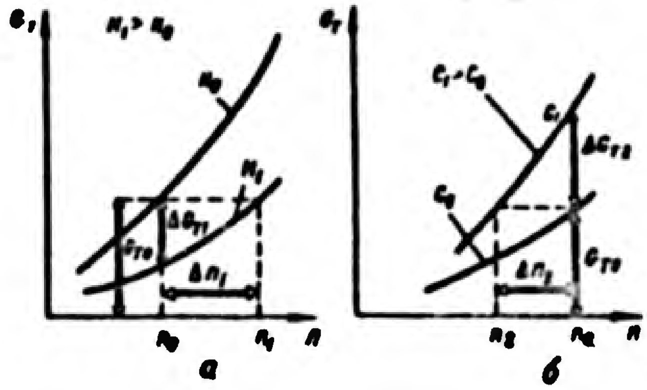

Fig. 225. Required fuel flow rate G_t as a function of rotor speed $\underline{n}$.
a) At various flight altitudes H;
b) at various flight speeds C.

To keep the rotor speed constant with increasing flight altitude, it is necessary to reduce the fuel flow rate, while at increasing flight speeds, the feed rate must be increased in accordance with the reduction in the weight flow rate of air. The variation in the required fuel flow-rate curves as a function of flight altitude and speed for various constant engine-rotor speeds is shown in Fig. 224. Using these curves, it is possible to plot required fuel flow-rate curves as a function of rotor speed at various flight altitudes and speeds (Fig. 225).

Examining the required fuel flow-rate curves for various flight regimes, we may conclude that with the fuel flow rate unchanged, an increase or decrease in flight speed or altitude will lead to a reduction in engine rotor speed.

As we know, control programs for the majority of engines provide for maintaining rotor speed constant under variations in flight altitude and speed. It is therefore necessary to vary the fuel feed rate in order to hold a given rotor speed under changing flight conditions.

If at some speed n_0 (Fig. 225a), the flight altitude increases from H_0 to H_1, then in order to keep the rotor speed n_0 unchanged, the fuel feed rate must be reduced by an amount ΔG_{t1}. If the fuel feed rate G_{t0} were held constant, this would lead to an increase in rotor speed of Δn_1. Similarly, as we can see from looking at Fig. 225b, in order to hold rotor speed constant while the flight speed changes from C_0 to C_1, we must increase the fuel feed rate by an amount ΔG_{t2}. If the fuel feed rate G_{t0} were held constant with increasing flight speed, there would be an accompanying drop Δn_2 in rotor speed.

It should be noted that at high rotor speeds, a change in flight speed and flight altitude by exactly the same amount would make it necessary to change fuel feed rate by more than would be the case at lower rotor speeds.

3. AVAILABLE FUEL FEED RATES

As we have already mentioned, gear-driven piston and gear pumps are used, in the main, to supply fuel to combustion chambers.

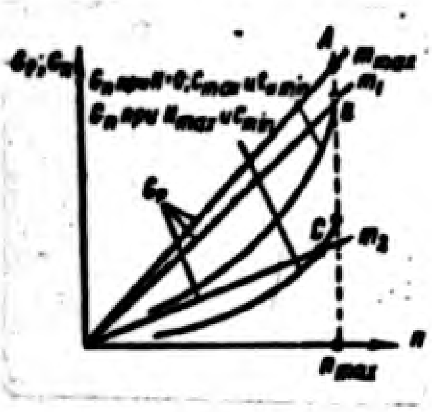

Fig. 226. Available fuel flow rate G_r and required fuel flow rate G_p as functions of rotor speed n with piston pump installed on engine (m_{max} is the maximum angle at which the inclined disk can be located).

Piston pumps are variable-throughput pumps, i.e., depending on the position of the inclined disk they provide various fuel feed rates at constant engine-rotor speed and, consequently, at constant pump-rotor speed.

Figure 226 shows characteristic curves for a piston pump. The characteristic represents the relationship between pump throughput and engine-rotor speed at various positions of the inclined disk. The position of the inclined disk is indicated arbitrarily on the figure by the letter m. The same figure gives two curves showing the fuel feed rate required by the engine on which this pump is installed: at maximum speed in flight near the ground and at minimum speed in flight at maximum altitude. The regimes selected correspond to cases of maximum and minimum fuel flow rate requirements of the given engine.

At the maximum angle of the inclined disk and maximum rotor speed, the available fuel flow rate provided by the pump (point A on the characteristic curve), is higher than the fuel flow rate required, which corresponds to point B. The excess fuel represents spare pump throughput.

In order to provide for a given operating regime, for example, maximum speed, it is necessary to satisfy the relationship equating the available and required fuel flow rates. Then in order to ensure the operating regime that corresponds to point B on the required flow-rate characteristic, the inclined disk should be set to position m_1, while in order to provide the regime corresponding to point C on the required flow-rate curve, the inclined disk should be set to position m_2.

It follows from this that when a variable-throughput piston fuel pump is installed on an engine, any engine operating regime may be obtained by setting the inclined disk to a position such that at the given engine-rotor speed, the fuel pump has an available throughput matching that required by the engine. The role of the automatic regulator installed on the engine reduces, in the case of a piston pump, to setting the inclined disk to the required position.

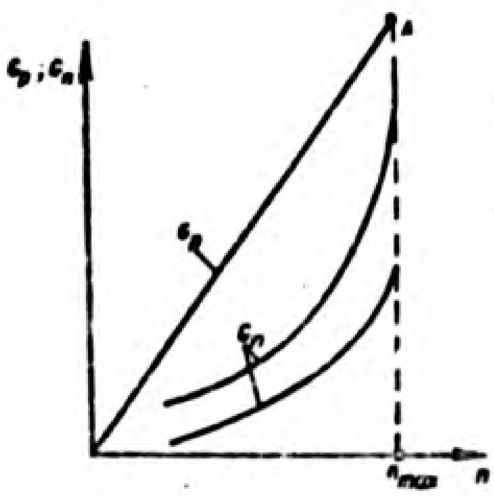

Fig. 227. Available fuel flow rate G_r from gear pump and required fuel flow rate G_p as functions of rotor speed $\underline{n}$.

The throughput of a gear pump is constant at constant gear speed. Figure 227 shows the way in which gear-pump throughput depends on engine rotor speed. As in the case of the piston pump, the same figure gives two

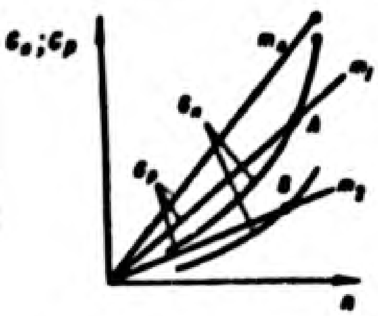

Fig. 228. Fuel flow rates G_r available from gear pump with bypass valve and required fuel flow rates G_p as functions of rotor speed n.

curves for the required fuel flow rate for the same flight regimes. At point A on the pump characteristic curve there is also the required extra throughput for maximum speed.

As we can see, in all engine operating regimes and all flight regimes, a gear fuel pump has an available throughput greater than that required, i.e., an excess fuel flow rate is available. The available and required fuel flow rates are matched with the aid of special bypass devices.

The basic fuel-system diagram (see Fig. 200) shows the way in which the bypass valve is connected into a fuel-supply system using a gear pump. The bypass valve is connected in parallel to the injectors at the pressurized line. Depending on the position of the valve, a greater or smaller fraction of the fuel is bypassed from the pressurized line to the pump intake side. As a result, at constant pump throughput, it is possible to supply various quantities of fuel to the engine, depending on the position of the valve which, as for the inclined disk, can be characterized by the quantity m.

Characteristic curves for a gear pump with a bypass valve in various positions are shown in Fig. 228. To provide a steady engine operating regime at points A and B, respectively, on the required fuel flow-rate curves, it is necessary to set the bypass valve to the positions m_1 and m_2. Then for position m_1, the amount of fuel bypassed to the intake side will be less than for position m_2.

The role of an automatic regulator installed on an engine using a gear fuel pump with a bypass valve reduces to setting the bypass valve to the position for which the available fuel flow rate equals the required flow rate.

A centrifugal pump is also characterized by the need to set a bypass valve so as to match the required and available fuel flow rates.

4. OPERATING STABILITY OF TRD USING FUEL PUMP WITHOUT AUTOMATIC REGULATOR

Let us consider the reasons for which a turbojet engine cannot operate without an automatic speed regulator. To do this, we analyze the operating stability of a TRD using a gear-driven fuel pump without automatic regulator.

The engine is required to operate stably in all operating regimes from idle running to maximum speed and in all aircraft flight speeds and altitudes.

By stable operation, we mean the ability of an engine under given external conditions and constant throttle setting to maintain a given rotor speed in the course of time.

Owing, for example, to pulsating delivery of fuel by the pump, intermittent variations in the parameters of the air at the engine intake, or other factors, there may be random deviations in speed from the prescribed value. We agree to say that a stably operating engine has positive self-regulation. After elimination of the factor responsible for a random deviation, the speed will again return to the prescribed value.

An unstable engine, which has negative self-regulation, possesses the property that following a random deviation, the rotor speed will not return to the prescribed value in the course of time, but will increase or decrease.

Finally, if under a random deviation in rotor speed, the speed takes on any value close to that prescribed, the engine will have zero self-regulation in the given regime.

Engine self-regulation, i.e., its ability to return to the initial

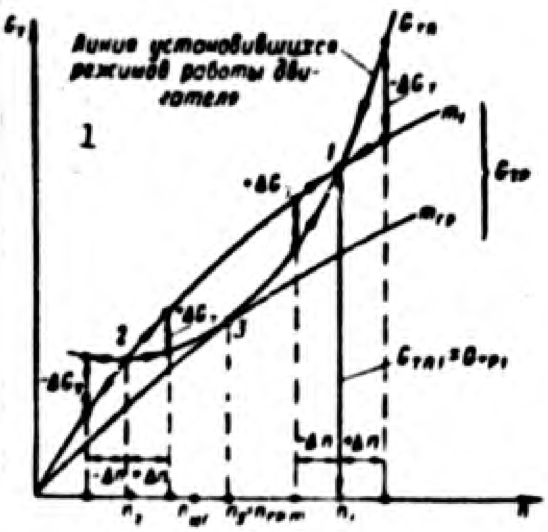

Fig. 229. Operating stability of TRD without regulator. 1) Line of steady engine operating regimes.

operating regime, depends on the nature of the relationship between the curves representing the required and available fuel flow rates with respect to the engine-rotor speed.

Let us consider the operation of an engine and fuel pump with rotor speed n_1 (Fig. 229). To operate the engine at speed n_1, it is necessary to set the final control element of the pump to position m_1, so that the available fuel flow rate G_{tr1} equals the required flow rate G_{tp1}.

An arbitrary decrease in rotor speed by a small amount Δn (section Δn in Fig. 229), leads to a situation in which the available rate of supply of fuel to the engine does not remain constant, but diminishes. The required fuel-flow rate decreases still more, however, and then the engine-rotor speed will return to the given value n_1 owing to the excess torque that appears owing to the excess fuel flow rate $+\Delta G_t$.

Similarly, any random increase in speed by an amount Δn (section $+\Delta n$ in Fig. 229), will lead to an increase in the available rate of fuel delivery to the engine. The required fuel flow rate for operation in a regime at speed $n_1 + \Delta n$ increases still more rapidly. Thus in this case the engine rotor speed will again return to the given value n_1 owing to the excess torque due to the rotor resistance to rotation. Thus, the engine operating regime corresponding to point 1 (Fig. 229), is stable and the engine possesses positive self-regulation.

In like manner, we can see that at rotor speed n_2, the engine operates unstably, with negative self-regulation. Thus, for example, when the final control element remains in the same position m_1, a small decrease in speed by an amount $-\Delta n$ will stop the engine, while a small increase by $+\Delta n$ will cause the engine rotor to accelerate from speed n_2 to n_1.

On the required fuel-flow-rate curve, we can find the speed n_3 that corresponds to zero engine self-regulation. At point 3 on the available fuel-flow-rate curve (with the final control element in position m_{grm}), there is contact with the curve for the required fuel flow rate. When the rotor speed varies slightly from the prescribed value n_3, there will be no excess torque at the rotor and, consequently, the speed will not return to the prescribed value. The speed n_3 corresponding to zero self-regulation is called the limiting speed for the final-control-element position and is represented by n_{grm}. At a speed below n_{grm}, a motor with a fuel pump not using a regulator cannot operate stably, i.e., it has negative self-regulation. To the right of point 3, there is a region of positive self-regulation, i.e., a region of stable engine operating regimes.

Thus, the speed n_{grm} divides the line of steady operating regimes into two sections – stable operating regimes for an engine without regulator for which the final control element is in a fixed position and unstable operating regimes.

The n_{grm} depends on the characteristics of turbine and compressor, and when the engine is running parked on the ground, the speed will range from $(0.4\text{-}0.65)n_{max}$, where the lower values apply to TRD using a centrifugal compressor. With the engine running idle, the speed will lie in the range $(0.2\text{-}0.4)n_{max}$, with the lower values also applying to an engine with a centrifugal compressor. Thus present-day engines in the speed range of n_{grm} to n_{mg} are characterized by negative self-regulation, so that stable engine operation cannot be used unless special regulators are employed.

As a consequence, in order to obtain stable operation of a TRD over this rotor-speed range, it is absolutely necessary to install an automatic regulator on the engine. In the speed range from n_{grm} to n_{max}, it is in principle possible to have stable engine operation where no special regulators are used, but only for the case in which external conditions remain unchanged, i.e., where flight speed and height remain constant.

5. EFFECT OF FLIGHT SPEED AND ALTITUDE ON OPERATING STABILITY AND ROTOR SPEED OF TRD WITHOUT REGULATOR

As the flight altitude and speed vary, there is a change in the way in which the required fuel-flow-rate curves change as a function of rotor speed. The characteristic curves for a fuel pump remain nearly unchanged in this case. Figure 230 shows required fuel-flow-rate curves for altitudes $H_0 < H_1 < H_2$, which show that as the flight altitude increases, the limiting speed n_{grm} increases.

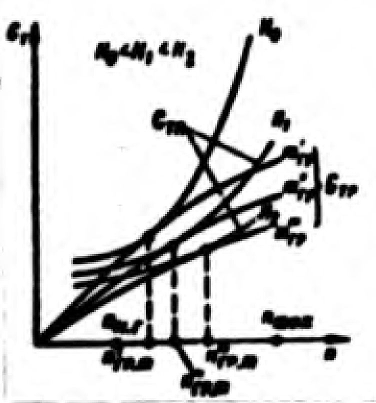

Fig. 230. Effect of flight altitude on stability region of TRD.

Thus, a TRD without regulator, operating stably under conditions at the ground at a rotor speed of let us say n_0, may become unstable at some altitude. As a consequence, as flight altitude increases, the region of stable engine operation with respect to rotor speed is narrowed.

As flight speed increases, for example, from V_0 to V_1 (Fig. 231), the region of stable operating regimes expands slightly.

Let us now look at the effect of flight height and speed on the rotor speed of a TRD.

It was shown earlier that in engine operating regimes above cruising speed, in order to hold rotor speed constant it is necessary to

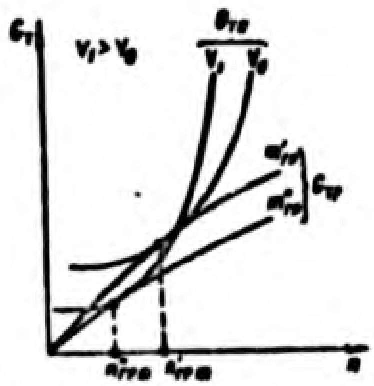

Fig. 231. Effect of flight speed on stability region of TRD.

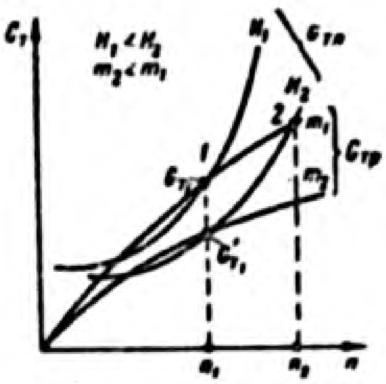

Fig. 232. Effect of flight altitude on rotor speed of TRD.

reduce the rate at which fuel is supplied to the engine as flight altitude increases, and to increase the feed rate with increasing flight speed, in accordance with the variation in the weight flow rate of air.

Figure 232 gives required fuel-flow-rate curves for altitudes H_1 and H_2. For operation at speed n_1 at height H_1, the required fuel flow rate is G_{t1}, and the final control element should be in position m_1. At a greater altitude H_2, for the same speed n_1, it is necessary to supply less fuel to the engine, i.e., the flow rate should not be G_{t1}, but G'_{t1}. To do this, the final control element should be moved from position m_1 to position m_2, with $m_2 < m_1$. If the final control element is not reset in this manner, at a height $H_2 > H_1$, the rotor speed will increase to a value n_2 determined by the point of intersection 2 of the curves for the available and required fuel flow rates at height H_2. Thus, with increasing flight altitude and a fixed position of the final control element, the rotor speed for a TRD without a regulator will rise.

Using a similar argument, we can see that where the final control element remains in the position needed for operation at a given speed, an increase in flight speed will lead to a decrease in rotor speed. As a consequence, in order to keep rotor speed constant while flight speed and altitude vary, it is necessary to change the available rate at which fuel can be fed to the engine until it reaches the required values by resetting the final control element.

Modern TRD carry automatic regulators that automatically maintain a given rotor speed for all flight speeds and altitudes.

Speed regulators are subject to several requirements pertaining to the quality of the control process. While the engine rotor is accelerating, the regulator should not permit the maximum rotor speed to be exceeded by more than 2-3% of the maximum value. If this is not the case, it is possible for turbine or compressor parts to fail. The accuracy with which it is necessary to maintain the maximum speed as altitude and flight speed vary is high (within 0.3-0.5%), in view of the great effect that rotor speed has on the strength safety factor for turbine blades and on the magnitude of engine thrust. The prescribed speed should be restored as rapidly as possible during the control process.

The basic component parts of a regulator are a sensing element to measure rotor speed and an amplifying device (servomotor), which is designed to set the final control element to the required position. The desired speed is set with the aid of a special regulator adjusting mechanism which is connected to the engine control in the cockpit.

Chapter 19

SENSING ELEMENTS AND SERVOMOTORS FOR SPEED REGULATORS

1. BASIC TYPES OF SENSING ELEMENTS FOR REGULATORS WORKING ON THE PRINCIPLE OF DEVIATION

The sensing element (detector) of a speed regulator is one of the most important parts of the regulator. It is designed to measure the rotor speed, and where this deviates from the prescribed value, to actuate the regulator so as to eliminate this deviation.

At present, centrifugal-pendulum and hydrocentrifugal sensing elements are in very common use.

Figure 233 shows the centrifugal pendulum of a sensing element. The centrifugal weights 1, which can turn about shafts 2, are rotated with the aid of shaft 3 which is linked kinematically with the engine rotor. The centrifugal forces that are then produced by the weights load valve 4 with a force C, transmitted with the aid of the pivoting pins 5; this force is directed along the axis of the sensing element. In the opposite direction, the valve is loaded by force F due to spring 6, which acts on the valve through the ball bearing 7.

The spring force depends on the position of coupling 8 of the adjusting mechanism and the position of the valve. By moving coupling 8, it is possible to vary the initial force of the spring in order to adjust the sensing element for the desired speed. All other conditions being equal, the position of the valve is determined by the magnitude of the force C, which depends on the speed of the sensing-element shaft.

If the sensing-element shaft is at rest, the centrifugal force C will equal zero, and the valve will be held at the very top position by the initial spring force, as shown in Fig. 233. As the engine-rotor speed increases, the centrifugal force C due to the weights will increase. The valve will not move downward, however, until the centrifugal force C due to the weights exceeds the initial spring force. Let the speed corresponding to the point at which the valve begins to move be n_{nach}. When $n > n_{nach}$, the downward motion of the valve compresses the spring, increasing the spring force. To each steady spring force there will correspond a definite value of the force C and an equal value of the opposing spring force F, and also a definite position of the valve, specified by the coordinate y. The valve can act directly as the final control element or as a controlling element for an intermediate mechanism (for example, a hydraulic servomotor, as shown in Fig. 233).

The slide valve takes up some position, specified by the coordinate y_0, as the initial (neutral) position. We let the rotor speed corresponding to this valve position be n_0, and call it the equilibrium speed. In this position, the valve has no effect on the position of the final regulating element. Thus, for example, where the valve acts as a hydraulic servomotor, in the equilibrium position, the valve flanges cover the working-fluid inlet and outlet channels in the servomotor.

It is quite clear that as the initial spring force F_{predv} increases, the rotor speed at which the valve will be in the neutral position will increase.

The centrifugal force C produced by the weights and applied to the axis of the sensing element, will depend on the speed and position of the weights, as determined by the magnitude of y (see Fig. 233), as well as on the structural parameters of the sensing element (its dimensions, the weight and shape of the weights, and the gear ratio between the engine rotor and the sensing-element shaft).

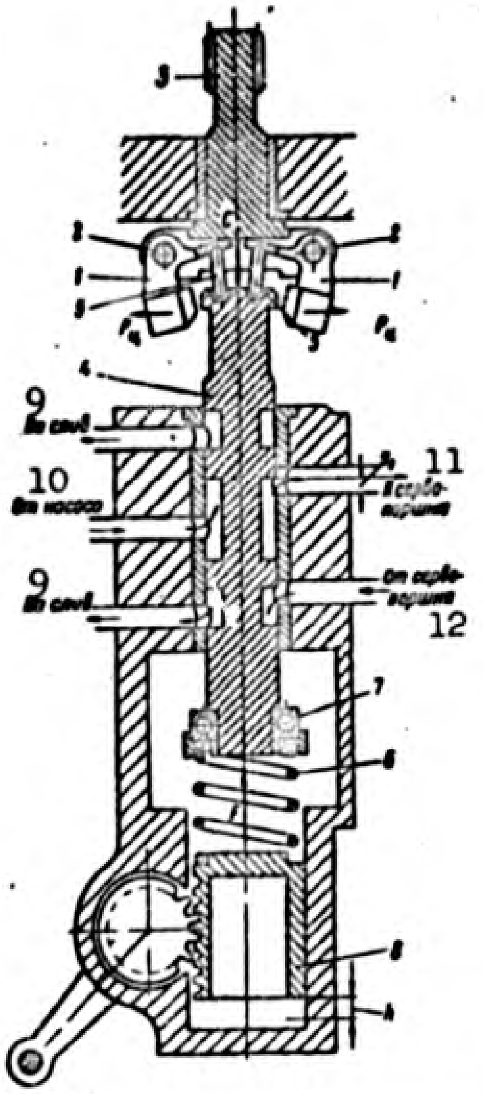

Fig. 233. Basic diagram of centrifugal-pendulum sensing element. 1) Centrifugal weights; 2) pivots; 3) shaft; 4) valve; 5) pivoting pins; 6) spring; 7) ball bearing; 8) coupling; C) centrifugal force due to weights, acting along axis of sensing element; F) spring force; P_{ts}) centrifugal forces due to weights; h) coordinate of adjusting-coupling position; y_0) coordinate corresponding to neutral position of valve; 9) to drain; 10) from pump; 11) to servomotor piston; 12) from servomotor piston.

Figures 234 and 235 show the approximate variation in the force C as a function of n and of y for given sensing-element structural parameters.

The spring force F depends on the position of the valve, as determined by the coordinate y, position h of the adjusting-mechanism coupling (see Fig. 233), and the spring stiffness k. This function is shown in Fig. 236.

Figure 237 shows a diagram of a hydrocentrifugal sensing element. Its drive shaft, linked kinematically with the engine rotor, actuates impeller 1, to which the working fluid (oil or fuel) is supplied. The pressure of the working fluid as it leaves the impeller, which is proportional to the density of the fluid and the square of the speed, is applied to the bellows element 2. The force C due to the pressure of the working fluid, appearing at the bellows, is balanced by the spring tension force F.

As we can see from Fig. 238, the force C does not depend on y. In comparison with a centrifugal-pendulum sensing element, the

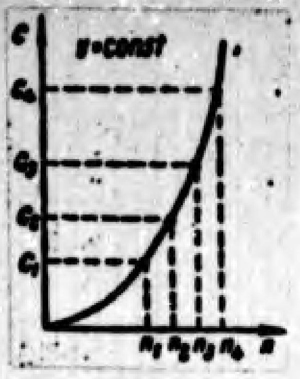

Fig. 234. Centrifugal force C due to weights as a function of speed n.

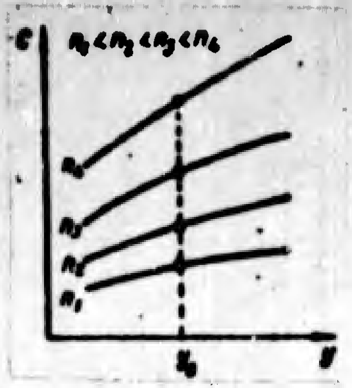

Fig. 235. Centrifugal force C due to weights as a function of position of sensing-element rod (coordinate y).

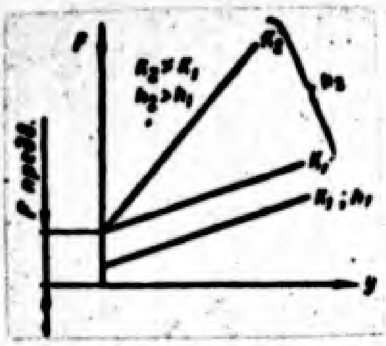

Fig. 236. Curves for sensing-element spring with various stiffnesses k and various positions of the adjusting mechanism (coordinate h).

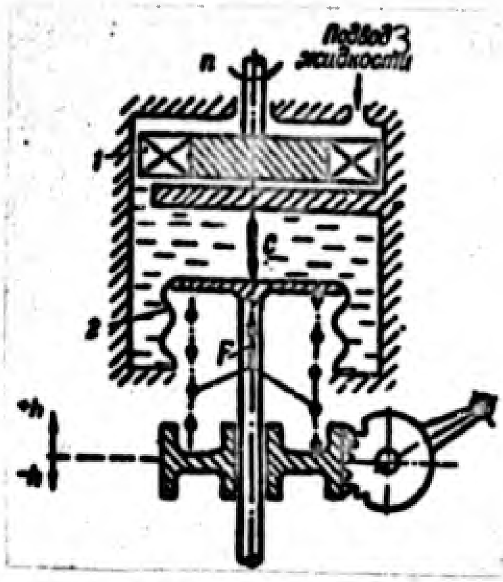

Fig. 237. Basic diagram of hydrocentrifugal sensing element. 1) Impeller; 2) bellows; C) force due to pressure of fluid on bellows; F) spring tension force; 3) fluid inlet.

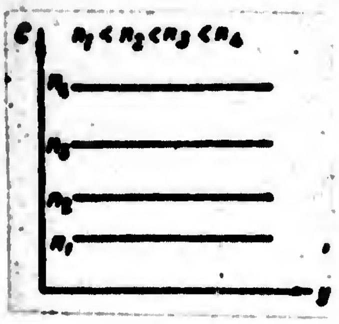

Fig. 238. Force due to fluid pressure as a function of position of rod of hydrocentrifugal sensing element for various rotor speeds.

hydrocentrifugal sensing element has the disadvantage that the equilibrium speed varies with variation in the working-fluid density when the initial spring force remains constant. A 1% decrease in the weight density in comparison with the initial value will lead to an increase of about 0.5% in the equilib-

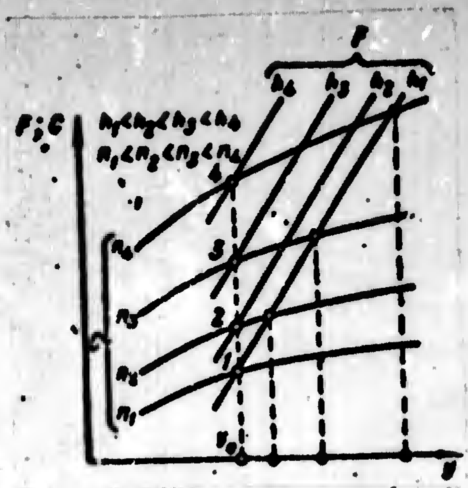

Fig. 239. Set of characteristic curves for centrifugal sensing element.

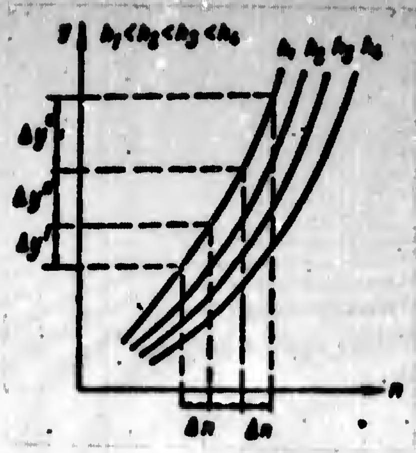

Fig. 240. Static characteristics of sensing element.

rium speed. It is thus necessary to readjust the regulator in going from winter to summer flying conditions and back again. Regulators using hydrocentrifugal sensing elements offer many advantages, however, as we shall see below.

If we have the curve showing the way in which C and F vary, we can construct the equilibrium (static) characteristic curve of the sensing element (calibration curve), which shows the relationship between the position coordinate y of the rod and the speed n for a given value h of initial spring force in steady (static, equilibrium) operation of the sensing element. To construct the calibration curve, we must combine the sets of curves for C and F (Fig. 239), and obtain the values of y and n for a given value of h at the points of intersection of the C and F curves.

It is clear from the calibration curves for various values of h (Fig. 240), that there is a specific rod displacement equaling Δy corresponding to each deviation Δn of the speed from the prescribed value.

2. FUNDAMENTAL PROPERTIES OF CENTRIFUGAL SENSING ELEMENTS

Stability of sensing element. By sensing-element stability at a given speed, we mean its ability to maintain a given position in the course of time, i.e., the ability of the sensing element to return to this position under random deviations from the prescribed position, following elimination of the factor producing the deviation.

Factors causing random deviations of the sensing element from the prescribed position may be: jolting of the engine, gravitational forces produced by sensing-element moving parts when it changes position in flight, etc.

Sensing-element stability depends on the mutual variation of the curves for the centrifugal force C (Fig. 241) and the spring force F, depending on rod position (coordinate y) at constant sensing-element shaft speed.

The figure shows typical possible cases of mutual variation in these curves.

Thus, for example, if the sensing element is in steady operation, the spring force and centrifugal force will be equal, corresponding to the points of intersection of the F and C curves in Fig. 241a, b, and c. The stability of the sensing element will be different in each case, however.

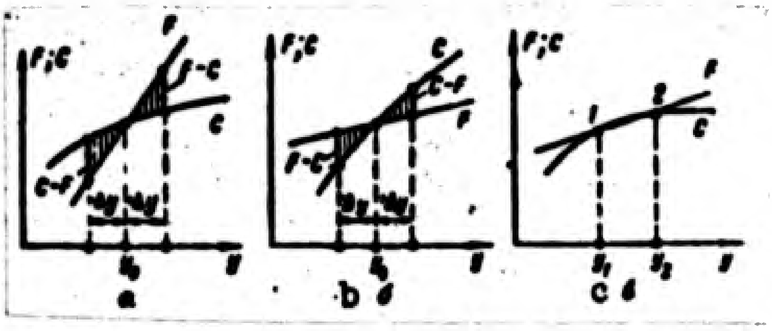

Fig. 241. The problem of sensing-element stability.

If the slope of the F curve is greater at the x axis than the slope of the C curve, the sensing element will be stable (Fig. 241a), as we can see without difficulty. When the sensing element departs from the equilibrium position, an excess restoring force F – C appears,

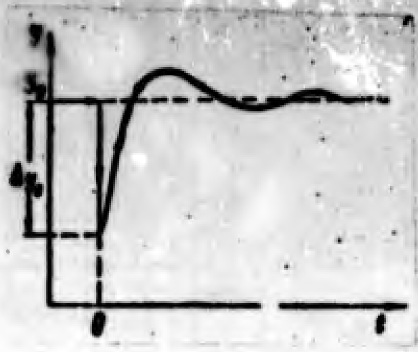

Fig. 242. Graph of a transient in a centrifugal sensing element with instantaneous readjustment.

which returns the element to the initial equilibrium position. The greater the slope of the F curve with respect to the C curve, the more stable the sensing element.

If the slope of the F curve is less than that of the C curve, the element equilibrium position will be unstable, since under a random deviation from equilibrium position there will appear an excess force C – F (Fig. 241b), tending to force the element still further from the equilibrium position and to establish it in one of the extreme positions.

And, finally, if the slopes of the F and C curves are identical on some section, the equilibrium position of the sensing element will be neutral, as no excess force will appear during random deviations from the equilibrium position (Fig. 241c).

Only stable sensing elements are suitable as speed regulators for TRD rotors.

The sensing element should not only be stable under random deviations from a prescribed position. Its equilibrium may also be upset when the adjustment mechanism is moved or when the shaft speed departs from a prescribed value. In this case, the sensing element should go to a new equilibrium position.

A transient process in a sensing element will always be oscillatory in nature owing to the presence of the inertial forces due to its masses (the weights, slide valve, etc.).

Figure 242 gives an example of a curve of slide-valve motion in the presence of a transient due to a random deviation of the valve from the equilibrium position by an amount Δy_0. The curve shows that in the course of time, the valve of a stable sensing element will return to the initial equilibrium position after several oscillations.

It is desirable for the natural resonant frequency of the sensing element to be as high as possible, and for the oscillations to be damped as intensively as possible. If this is not the case, the quality of the automatic control system may be severely impaired. In order to increase the natural resonant frequency, the masses of the weights, valve, etc. are made as small as possible, and the reduction in the weight centrifugal force owing to the reduction in mass is compensated by increasing the speed of rotation. The natural resonant frequency is also increased by an increase (acceptable) in spring stiffness.

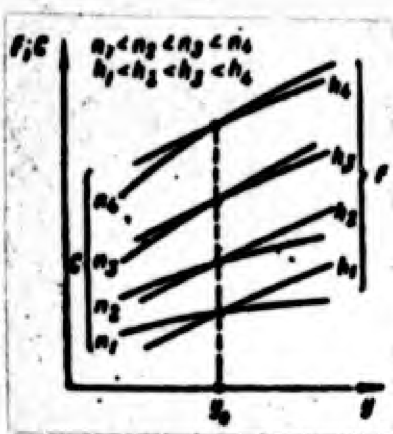

Fig. 243. The problem of stability of a centrifugal-pendulum sensing element at various speeds.

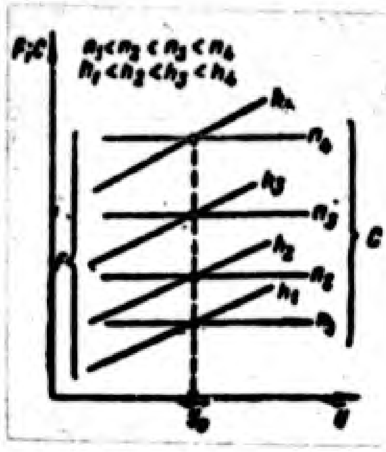

Fig. 244. The problem of stability of a hydrocentrifugal sensing element at various speeds.

In order to increase oscillation attenuation, the centrifugal weights are sometimes placed within a cavity filled with the working regulator fluid.

As the speed increases, the stability of a centrifugal-pendulum sensing element drops, since with increasing n, the slope of the C curves (depending on y) become steeper (increase) with respect to the x axis, and at some value of n may become greater than the slope of the spring-force curves (Fig. 243). In other words, the sensing element, which is stable at low speeds n_1 and n_2, may lose stability at high speeds n_3 or n_4. This hampers the design of a centrifugal regulator that will operate stably over a wide range of speed variation.

The stability of a hydrocentrifugal sensing element does not depend on the speed, since the force C does not depend on the coordinate y, and at all speeds, the slope of the F lines will be greater than the slope of the C curves (Fig. 244). As a result, the range of automatic operation for a regulator using a hydrocentrifugal sensing element is somewhat greater than that of a centrifugal-pendulum element.

Nonuniform operation of a sensing element. As we have already pointed out, a sensing element converts a speed deviation from a prescribed value into displacement of a rod connected to a control valve or directly to the regulator final control element.

A centrifugal sensing element has the property that for exactly the same departure of the speed from a prescribed value, the displacement of the rod will be less at low speeds than at high speeds. This nonuniform displacement of the rod at low and high speeds is undesirable, as it affects the process of regulating the TRD rotor speed.

The factor responsible for this nonuniform operation of the sensing element consists in the fact that the relative centrifugal force C rises sharply as the speed increases, while the spring force depends solely on the amount of spring deformation (i.e., on the position of the rod – the coordinate y).

Thus an identical increase in speed will cause a greater increase in the centrifugal force C at high speeds than at low speeds (see Fig. 234), while since the centrifugal force C is balanced by the spring force, at higher speeds, equilibrium of the sensing element will occur at greater rod displacements than at low speeds. Conversely, an identical displacement of the slide-valve rod during the control process will occur: at low speeds, in the presence of a larger departure of the speed from the prescribed value than at higher speeds.

This nonuniform operation of a centrifugal sensing element may be seen clearly from the calibration curve (see Fig. 240).

Element sensitivity at low speeds may be increased by reducing spring stiffness. In this case, however, at high speeds, the sensing element may become unstable, since the slope of the F lines will be less than the slope of the C curves or else small random departures of the speed from the prescribed value will lead to very large displacements of the valve rod and, in the final analysis, to an oscillatory transient.

Thus in regulator designs in use, spring stiffness is so chosen as to provide adequate sensitivity and stability of the sensing element at high speeds in the region of operating regimes. At low speeds, spring stiffness proves relatively high and as we have mentioned, greater speed departures from the prescribed values are required for small valve-rod displacements, i.e., during the control process, this element may not provide the required quality of regulation at low speeds.

This is one of the reasons that centrifugal regulators in service have an automatic-operation range that is limited so that the regulator comes into service not at idle-running speed, but at a higher speed called the automatic governor-operation cut-in speed (NAR speed).

The automatic-operation range of a regulator using a hydrocentrifugal sensing element is somewhat wider than for a centrifugal-element regulator, since the centrifugal force does not depend on the displacement of the valve rod and the stability of a hydrocentrifugal element does not depend on speed. This type of sensing element also exhibits nonuniform operation, which is compensated somewhat by the use of a less stiff spring.

Sensing-element zone of insensitivity. Looking at the calibration curves of Fig. 240, we can see that even where there is a very slight variation in the speed, the valve rod should move. Actually, this is not the case owing to the presence of dry-friction forces in a real sensing element. Thus the valve rod will not move within a certain range of speed variation, i.e., it will not react to a change in speed. In the presence of dry-friction forces, the sensing element will not react to a speed change as long as the change ΔC in the centrifugal force remains less than the dry-friction force. Thus the dry friction forces reduce element sensitivity to speed changes. The insensitivity zone $\Delta n_n = n_2 - n_1$ in the n_0 regime can be seen on the static characteristic curve (Fig. 245).

The dry-friction forces have more effect at low speeds than at high speeds, i.e., as the speed decreases, the zone of insensitivity increases. In fact, in order to overcome the dry-friction force, a centrifugal-force increment $\Delta C \geq F_{tr}$ is needed (Fig. 246); at high speeds such as n_2, this increment ΔC is reached when the speed departs from the prescribed value (n_2) by an amount Δn_2, while at a low speed n_1, the same increment ΔC is reached with a speed departure from the prescribed value by an amount Δn_1 which is larger than Δn_2. This is one of the reasons that the automatic-operation range of a speed governor is restricted.

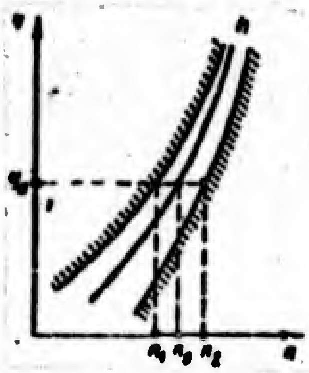

Fig. 245. Insensitivity zone of sensing element with respect to rotor speed.

In order to reduce the effect of the friction forces, the centrifugal weights are driven at high speed; as a result, the required centrifugal-force increment and actuation of the valve occur at low departures of the speed from the prescribed value. In addition, ball bearings are used for the centrifugal weights. In order to reduce fric-

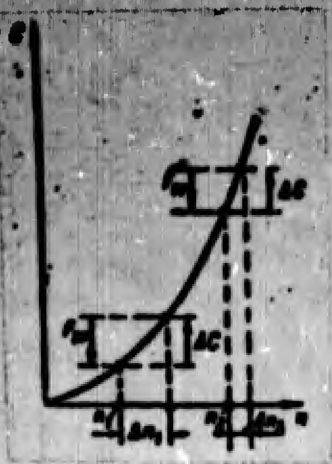

Fig. 246. Effect of dry-friction forces on sensing-element sensitivity.

tion forces at the bushing, the valve slide is also rotated at the same speed as the centrifugal weights.

Pivoting pins are installed where the weights and valve slide come into contact. In regulator designs in service, the dry-friction forces appearing at the sensing-element axis do not exceed 10-15 g.

The magnitudes of dry-friction forces are heavily affected by the nature of the fluid used as the regulator working fluid. By using oil in place of kerosene, we can reduce the danger of the appearance of dry-friction forces; this leads, however, to structural complications in the regulator owing to the need to use a special oil system, isolated from the regulator fuel chambers.

Friction forces are lower in hydrocentrifugal sensing elements owing to the absence of centrifugal weights.

3. SENSING ELEMENTS FOR REGULATORS PERFORMING THE CONTROL FUNCTION ON THE BASIS OF THE DISTURBANCE

As we have already mentioned, in automatic-control systems using the principle of control in accordance with the disturbance (i.e., the compensation principle), the regulator sensing elements record the change in the factors responsible for a variation in engine-rotor speed, and adjust the rate at which fuel is fed to the engine in accordance with the selected control program.

As a rule, sealed bellows capsules from which the air has been exhausted (aneroid elements) are used as sensing elements for the total air pressure at the engine intake. The capsules are placed within a sealed housing within which the pressure to be measured appears. In order to increase the stroke of the regulator rod, several capsules are joined into a common unit (Fig. 247).

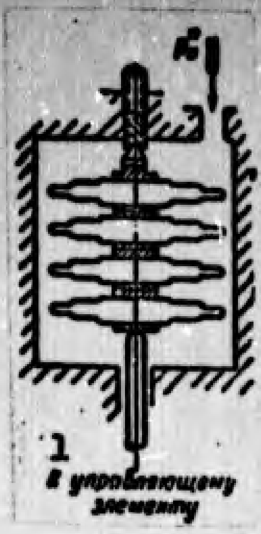

Fig. 247. Basic diagram of sensing element for total air pressure at engine intake. 1) To control element.

Elements for sensing the difference (drop) between the total pressure and atmospheric pressure use corrugated tubes (bellows). The total pressure appears within the tube, while the surrounding atmospheric pressure is applied directly to the outside surface of the bellows (Fig. 248).

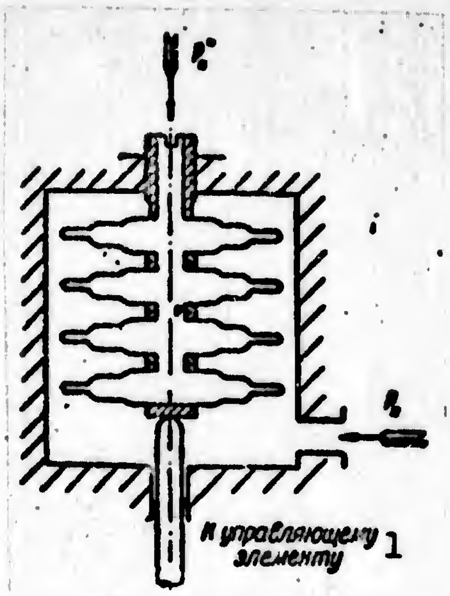

Fig. 248. Basic circuit of sensing element for air pressure drop. 1) To control element.

Sensing elements for the stagnation temperature of the air at the engine intake use the following types of thermometers: hydraulic, vapor-liquid, dilatometer, electrical, etc.

Operation of a hydraulic thermometer is based on the ability of a liquid to change its volume when the temperature varies.

Figure 249 shows a hydraulic thermometer using a bellows. The sealed chamber 1 contains a bellows 2, one end of which is welded to the chamber, with the free end attached to the control-element rod. The space between the bellows and the chamber wall is filled with a fluid (kerosene, oil, etc.). When the temperature increases, the liquid expands, compressing the bellows, and thus moving the rod.

In a vapor-liquid sensing element, the volume between the bellows and chamber wall is filled with a liquid having a low boiling point (ethyl chloride, methyl chloride, acetone, etc.). Boiling of the liquid causes a saturated vapor to form over the surface of the liquid; the

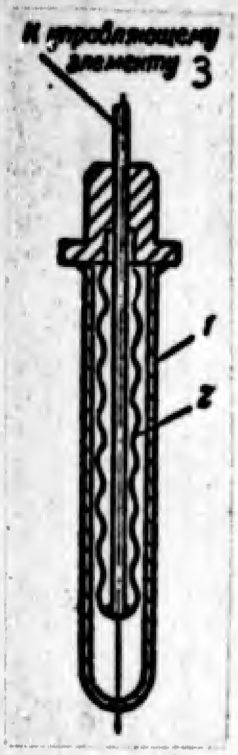

Fig. 249. Diagram of hydraulic thermometer with bellows. 1) Sealed chamber; 2) bellows; 3) to control element.

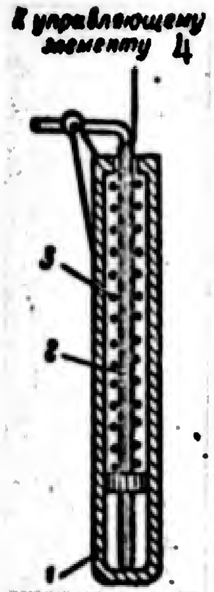

Fig. 250. Dilatometer temperature-sensing element. 1) Metal tube; 2) rod; 3) spring; 4) to control element.

vapor pressure depends on the temperature of the liquid. A change in air temperature causes a variation in the temperature of the liquid, results in a change in the saturated vapor pressure, and, finally, causes the rod of the sensing element to move, producing a change in the rate at which fuel is fed to the engine.

Operation of a dilatometer sensing element is based on the ability of solids to change their linear dimensions under temperature variations. The element consists of a metal receiving tube 1 and a rod 2 pressed against the end of the tube by a spring 3 (Fig. 250). An increase in the temperatures of the tube and the rod, which are made from materials with different coefficients of linear expansion, causes the rod to move relative to the tube by an amount proportional to the variation in temperature. This displacement is transmitted to the regulator control element. The tube is made from aluminum, which has a large coefficient of linear expansion, and the rod from invar, whose coefficient of linear expansion is very small. Such a combination raises element sensitivity and reduces the response-time lag caused by the air layer between rod and tube.

4. SERVOMOTORS USED WITH SPEED REGULATORS

Speed-regulator designs in use widely employ hydraulic servomotors, which act as amplifying or programed devices.

Amplifying servomotors are used to move the final control element to a new position, since the forces required for this purpose may reach high values, considerably exceeding those developed by the sensing element. Servomotors of this group "amplify" signals received from the regulator sensing elements.

Programed servomotors are made in accordance with the type of servo system in which the actuating element duplicates the motion of the control element in accordance with a specified temporal or kinematic program. They are used in regulator-adjusting mechanisms to delay readjustment of the regulator to a higher speed when the control lever is moved rapidly, and also to move the final control element, which duplicates the motion of the control element connected to the sensing element in accordance with a specified kinematic program. For this purpose, proportional feedback, which may be hydraulic or mechanical, is introduced between the actuating and control elements.

The servomotor consists of an actuating element connected to the final-control element or regulator adjusting mechanism, and a control element connected to the sensing element or regulator-adjusting lever.

A servopiston is most frequently used as the actuating element, while the elements by which the sensing element controls operation of the actuating device are slide valves and other types of valves.

Oil or kerosene may be used as the working fluid. Oil reduces the dry-friction forces and protects regulator parts against corrosion. The utilization of oil complicates system construction, however, as this makes it necessary to use a special oil pump with reduction valve. In addition, the viscosity of the oil, which affects regulator operation, depends heavily on the temperature.

Where kerosene is used as the working fluid, system construction is simplified owing to the lack of a pump; another advantage of a kerosene system is the fact that even at very low temperatures, kerosene has low viscosity. In addition to these advantages a kerosene system has the following drawbacks: kerosene does not protect parts against corrosion, especially when it contains an admixture of water and, in addition, the slight lubricating power of kerosene may cause dry friction to appear.

Let us consider the fundamental types and properties of servomotors used for speed regulators.

Hydraulic servomotor with slide valve and double-acting servopiston. In this servomotor, the control element is a slide valve 1 (Fig. 251), which is used to supply and withdraw the working fluid (fuel or oil) to or from the servopiston.

The servopiston moves to either side owing to the energy of the fluid. The servopiston spring plays an auxiliary role, i.e., it determines the initial position of the servopiston when the engine is not operating, that is, it holds the regulating element in the position in which the fuel feed rate is at a maximum.

In the neutral position (see Fig. 251), the slide-valve rings in this case completely cover (shut off) the channels through which liquid is supplied to the servopiston cylinder, and thus this is called a cutoff valve.

When the slide valve moves out of the neutral position, the working-fluid inlet is opened to one of the servomotor chambers, while the other is connected to a drain. Motion of the servopiston and, consequently, a variation in fuel feed rate will occur until the slide valve returns to the neutral position. Thus, displacement of the valve leads to a change in the position of the servopiston and the final control element.

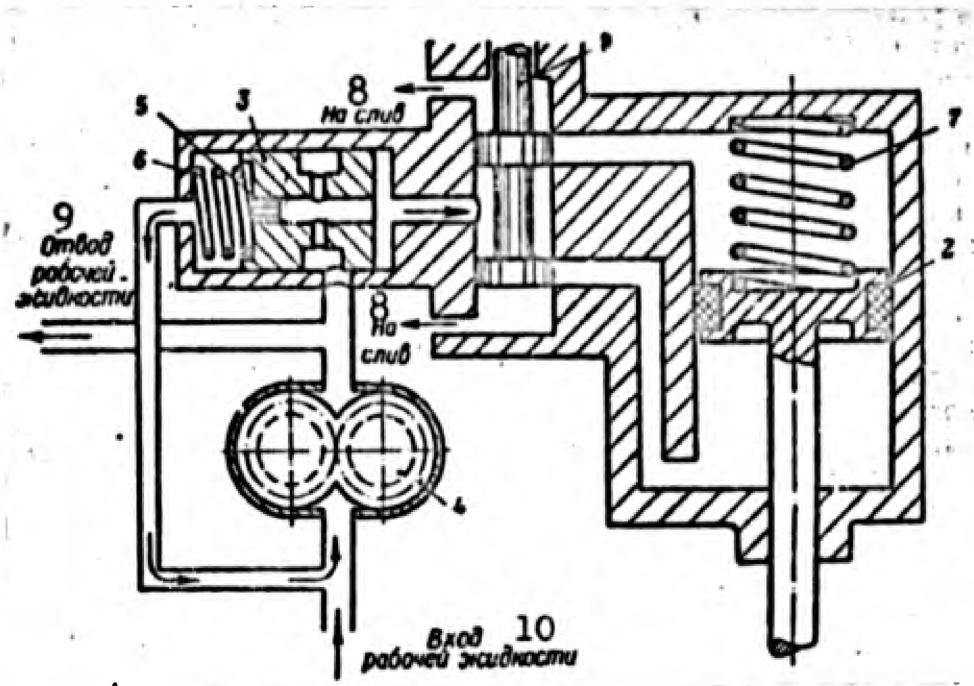

Fig. 251. Hydraulic servomotor with cutoff valve and double-acting servopiston. 1) Slide valve; 2) servopiston; 3) constant-pressure valve; 4) pump; 5) discharge nozzle; 6) valve spring; 7) servopiston spring; 8) to drain; 9) working-fluid outlet; 10) working-fluid inlet.

For proper functioning of the control system, the servopiston is connected to the final control element so that as the speed increases, the final control element moves to reduce the fuel feed rate.

At all equilibrium speeds, the slide valve of the speed-regulator system occupies a single unique neutral position, while to each equilibrium speed there corresponds a particular position of the servopiston. In other words, with the identical slide-valve neutral position, the servopiston may occupy various positions depending on the altitude, flight speed, and other factors. Such a servomotor is called a floating servomotor.

A servomotor with a control slide valve has the following characteristic property: displacement of the servopiston during the control process is larger the greater the deviation of the slide valve from the neutral position and the greater the time during which the slide valve remains away from the neutral position, i.e., the greater the time-cross section product for admission of the working fluid to the servomotor.

The response of the servomotor, i.e., the amount of servopiston displacement over a particular time interval depends on the servomotor structural characteristics: piston area, width of the port opened by the valve, rate at which the working fluid flows through the port and the stiffness of the servomotor spring. As piston area decreases and port width and the rate at which the working fluid flows increase, servomotor response improves.

The required servomotor response is provided by using a relatively high fuel pressure (10-25 kg/cm^2) in the chamber between slide-valve rings; this provides the required rate of working-fluid flow through the port; a piston with fairly small area is also used for the same purpose.

The possibility of increasing the pressure of the working fluid (fuel) in the chamber between slide rings is limited by the fact that in operation at high altitudes, the fuel pressure in the system will drop.

Since the servopiston should develop a quite definite force in order to move the final control element, the dimensions of the piston and the working-fluid pressure should be so matched that the required force can be obtained at the piston. This places a limitation on the choice of minimum servomotor piston dimensions.

In order to provide a constant response time for the servomotor in all engine operating regimes, the pressure of the working fluid in the cavity between slide-valve rings is held constant with the aid of the constant-pressure valve 3 (see Fig. 251).

The valve operates by throttling down the pressure of the working fluid supplied by pump 4 and flowing through the valve nozzle 5. If the pressure in the chamber between valve rings increases and becomes higher than the pressure set by means of the tension on spring 6 of

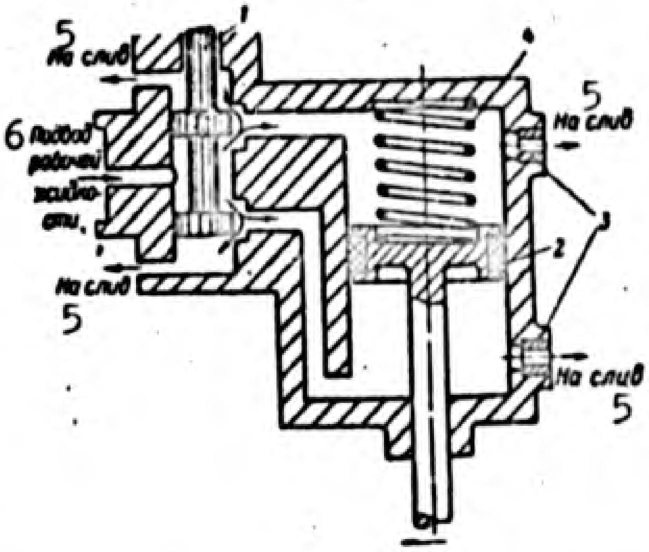

Fig. 252. Hydraulic servomotor with continuous-flow slide valve and double-acting servopiston. 1) Slide valve; 2) servopiston; 3) discharge nozzles; 4) servopiston spring; 5) to drain; 6) working-fluid inlet.

the valve, the valve will move to the left so as to decrease the flow-passage area, restoring the predetermined pressure in the chamber between valve rings. Conversely, when the pressure decreases, the valve moves to the right thus restoring the pressure in this chamber.

The presence of the spring increases the servomotor response time, since more force on the part of the working fluid supplied is needed to move the piston.

In present-day regulator designs, cutoff valves are used infrequently. Usually, when the slide valve is in the neutral position, its rings do not completely cover the fluid inlet channels to the servopiston cylinder, and fluid flows continuously from the intake to the drain through the gaps between the slide-valve rings and the ports, as well as through the nozzles 3, provided specially for this purpose (Fig. 252). Owing to the continuous flow of liquid through the slide valve, the liquid cannot solidify in the servomotor piston during extended periods of operation in steady regimes at low temperatures, and the dry-friction forces between the rotating slide valve and the sleeve are reduced.

In the neutral position of the slide valve, the gaps between its rings and the liquid intake ports are larger than the gaps on the liquid-outlet side, and this compensates for the leakage of liquid from the servomotor cavity through the nozzles 3 and the gaps between the piston rod and the housing.

The presence of spring 4 in a servomotor with a constant-flow slide valve makes it a static device, i.e., to each position of the piston there corresponds a quite definite position of the slide valve. Actually, in order to keep the piston in the upper position, a higher working-fluid pressure is needed under the piston, with less pressure above it, so as to correspond with the increasing spring force; this can happen only when the slide-valve position is shifted further downward. As a consequence, there is a proportional relationship between the positions of the piston and the slide valve. This fact affects the precision with which the speed of the engine rotor is controlled.

Among the drawbacks to hydraulic servomotors with a control slide valve, we must include the possibility for the appearance of dry friction between the slide valve and sleeve and the substantial mass of the slide valve. These drawbacks are less apparent in a servomotor using a valve-type control element.

Hydraulic servomotor using valve-type control element. In this servomotor, the position of the servopiston 4 (Fig. 253) is controlled by a valve 3, whose position is set by the sensing element. The flow of working fluid into cavity A of the servomotor is restricted by nozzle 1, and the flow to the drain by nozzle 2 and control valve 3. The pressure in cavity A depends on the amount of working fluid discharged through nozzle 2, while the pressure in cavity B equals the pressure of the liquid after the pump.

The piston is acted on by the force due to the liquid pressure

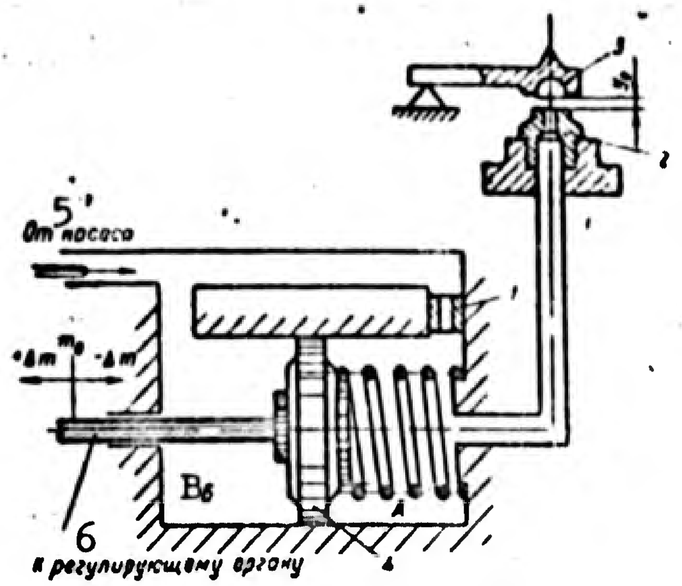

Fig. 253. Hydraulic servomotor with valve-type control element. 1 and 2) Nozzles; 3) control valve; 4) servopiston; A and B) cavities; 5) from pump; 6) to final control element.

from cavities A and B and the resultant force due to the spring and the action of the final control element. When the piston is stationary, the sum of the forces acting on it will equal zero and the volume of cavity A will not vary, i.e., the rate at which liquid flows through nozzle 2 equals the rate at which liquid flows in through nozzle 1.

Let us find the connection between the position of control valve 3 and the position of servopiston 4. If the valve closes nozzle 2, the liquid pressures in cavities A and B will be equal, and since the effective area of the piston on the side of cavity A is larger than on the other side, the forces due to fuel pressure and the spring will move the piston to the extreme left-hand position.

If valve 3 rises slightly, piston equilibrium will be upset, since the flow rate through nozzle 2 will increase, the pressure in cavity A will decrease, and the pressure drop across the piston will rise, moving the piston to the right so as to compress the spring; the volume of cavity A will then be decreased. Since the pressure drop across the piston is equal to the pressure drop at nozzle 1, the flow rate through nozzle 1 also increases when the drop across the piston increases. The flows through nozzles 1 and 2 are equalized when the change in the volume A ceases, i.e., when the piston comes to occupy its new equilibrium position. Then the pressure in chamber A will stabilize in accordance with the change of pressure in chamber B, and the resultant of the spring force and the effort from the regulating control and, consequently, the sum of all forces acting on the piston, will return to zero.

Thus, to each definite position of valve 3 there always corresponds a definite position of the servopiston. This type of servomotor is called a static servomotor.

A drawback to this servomotor lies in the fact that the valve control element is affected by the force due to the working-fluid jet leaving nozzle 2. There is almost no harmful effect due to dry-friction forces, however.

Hydraulic servomotors in servosystems. Figure 254 shows a structural diagram for a servomotor in which the servopiston duplicates the motion of the control slide valve owing to proportional hydraulic feedback. In this servomotor, the gear ratio between slide valve and servopiston is unity.

The servomotor has a servopiston 1 with hollow rod 2, within which there is a control slide valve 3, a servopiston spring 4, and a control slide-valve spring 5.

The servopiston is connected to a throttle valve by cylinder 6. The position of the control slide valve is set by the sensing element. Cavity A communicates with the discharge line. In servomotor cavity B, the working liquid is introduced under pressure through nozzle 7; it sets up a force that can move the servopiston to the left. The servopiston moves to the right under the force due to spring 4.

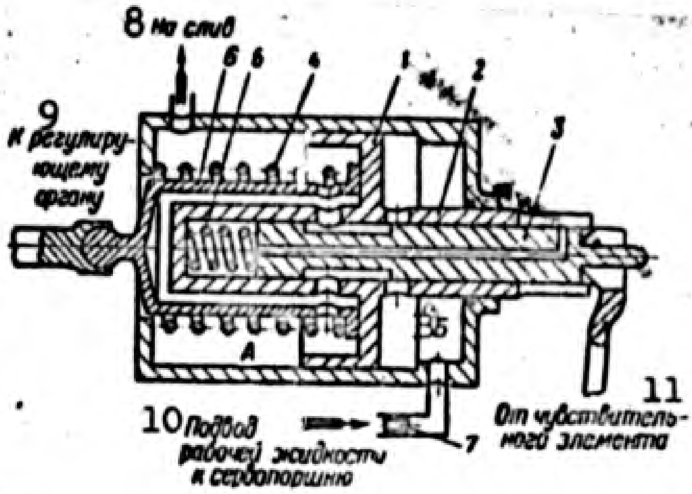

Fig. 254. Hydraulic servomotor with servopiston following position of control slide valve. 1) Servopiston; 2) hollow rod; 3) control slide rod; 4) servopiston spring; 5) control-slide-valve spring; 6) cylinder; 7) nozzle; A and B) cavities; 8) to discharge; 9) to final control element; 10) working-fluid inlet to servopiston; 11) from sensing element.

With the engine inoperative, in the absence of working-fluid pressure, the servopiston rests against a stop at the extreme right-hand position, owing to the spring force. In this case, there is no connection between cavity B and cavity A and the discharge line.

When the pressure of the working fluid increases in cavity B to a specific value, the servopiston moves to the left and is set to a position such that the radial holes in rod 2 of the servopiston coincide with a ring-shaped groove in the control slide valve, which in turn coincides with radial holes in cylinder 6 and in rod 2 on the other side of the piston; as a result, cavity B is connected to cavity A. Then the forces due to the working-fluid pressure and the spring, which act on the servopiston, will equal each other.

When the control slide valve moves to the left owing to the force produced by the sensing element, cavities B and A are disconnected, which increases the pressure of the working fluid in cavity B, and the servopiston moves so as to follow the slide valve, moving through roughly the same distance as the slide valve does.

If the force transmitted from the sensing element to the slide valve diminishes, the force due to spring 5 will move the slide valve to the right, increasing the discharge of working fluid from cavity B, while the flow of liquid into this cavity is limited by nozzle 7. As a result of this, the force due to spring 4 will cause the servopiston to follow to the right by about the same distance as the slide valve moves.

Thus, this entire system represents a servosystem, in which the servopiston follows the slide valve, and moves through a distance equal to about the distance through which the slide valve is displaced. The slight discrepancy between the positions of the slide valve and servopiston is caused by the effect of the servopiston spring force, which depends on the amount of deformation. Thus, in order to keep the servopiston further to the left, it is necessary to have a higher pressure in cavity B than in cavity A, which is done by decreasing the area of the slots used to introduce liquid from cavity B into cavity A.

The required servopiston rate of displacement is set by selecting nozzle 7 to have the necessary throughput.

Chapter 20

CENTRIFUGAL SPEED REGULATORS FOR ROTORS OF GAS-TURBINE ENGINES

1. DIRECT-ACTING SPEED REGULATOR

In a centrifugal direct-acting speed regulator, the sensing element 1 (Fig. 255), driven by the engine rotor, is connected directly to the final control element, which may be a hydraulically balanced slide valve 2.

The slide valve 2 has two rings; the left-hand one is a sealing ring, which the right-hand ring controls the bypassing of fuel through valve 6 from the high-pressure line ahead of the injectors to the inlet of pump 3 through duct 7. Duct 8 is used for discharge.

The regulator is adjusted for the desired engine-rotor speed with the aid of lever 4 of the adjusting mechanism.

In steady operation, where the given rotor speed is constant, the available fuel flow rate through the injectors equals the flow rate required for operation of the engine at the prescribed rotor speed. Since the fuel-pump throughput is greater than the fuel flow rates required in all engine operating regimes, in a steady regime, the slide valve control ring does not completely cover the fuel bypass port for duct 6, and the excess of fuel over the required quantity is bypassed through the resulting slit from the chamber following the pump to the pump intake chamber.

Motion of the slide valve to the right away from the initial position leads to a reduction in the rate at which fuel is supplied to the

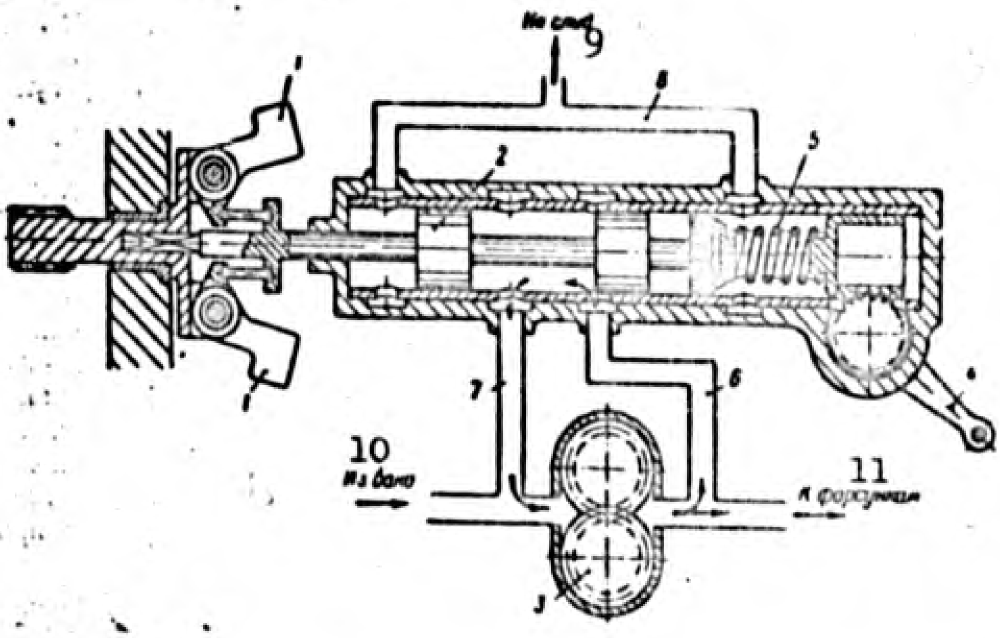

Fig. 255. Basic arrangement of automatic control system using direct-acting centrifugal speed regulator. 1) Sensing element (centrifugal weights); 2) slide valve; 3) pump; 4) control lever; 5) spring; 6, 7, and 8) valves; 9) to discharge; 10) from tank; 11) to injectors.

engine, while when the slide goes to the left, this rate will increase.

Let us consider operation of the regulator in two typical cases: when it is readjusted for a new speed, and when flight speed or altitude change.

When the regulator is reset for a higher speed, it is necessary to use the adjustment mechanism to increase the compression of the sensing-element spring, which displaces the slide valve to the left; this reduces the amount of fuel bypassed to the pump intake side, and increases the amount of fuel supplied to the engine. As a result, the rotor speed rises, and the increasing centrifugal force due to the weights 1 causes the slide valve to return to the right, increasing the rate at which fuel is bypassed to the control ring of the slide valve. Thus the increase in speed is retarded, and at a certain speed higher than the initial value, the slide valve will take up a new equilibrium position.

When the flight speed or altitude change, the regulator must maintain the rotor speed constant. Thus, for example, where the flight altitude increases while the position of the final control organ remains unchanged, the rotor speed will increase in comparison with the prescribed value. Here the regulator comes into play, since there is an increase in the centrifugal force due to the weights, and the final control element (slide valve) moves to the right, reducing the rate at which fuel is supplied to the engine, thus halting the increase in rotor speed.

It is easy to see that as flight speed or altitude vary, the rotor speed will not be held absolutely constant: as the flight altitude increases, it will rise somewhat, while it will decrease as the flight speed increases. Actually, if the flight altitude is increased, in order to maintain a prescribed constant speed, it is necessary to increase the rate at which fuel is bypassed to the slide-valve control ring, i.e., at a greater flight altitude, the slide valve should be located further to the right than at the initial altitude. But since the spring force will increase in this case, it is only possible to maintain the slide valve in this new position if the centrifugal force of the weights increases, i.e., the shaft of the centrifugal regulator must turn at higher speed and, consequently, the engine rotor must turn faster.

To each flight speed and altitude there should correspond a specific position of the final control element and, consequently, a specific rotor speed. In other words, after the conclusion of a transient caused by a change in flight speed or altitude, the regulator should exhibit a static error (residual nonuniformity of control), which will depend on the flight speed or altitude. Such regulators are called static regulators.

A direct-acting speed regulator of the type under discussion will have limited application owing to the large static errors that appear as flight speed and altitude increase. Such regulators are usually employed for low-power gas-turbine engines, for example, in turbostarters, to limit the maximum rotor speed. In this case, the regulator is made to operate in one regime, and no mechanism is provided for adjustment from the cockpit.

2. INDIRECT-ACTION SPEED REGULATOR WITH FLOATING SERVOMOTOR

In an indirect-action speed regulator, the final control element 1 (Fig. 256) is set to the required position with the aid of a servomotor, which consists of the servopiston 2 and control element (slide valve) 4, which is connected with the sensing element 5.

In the example under consideration, in which the fuel-supply system uses a gear-type fuel pump 7, the role of the final control element is played by the fuel-bypass needle valve for ducts 13 and 14, through which fuel is bypassed from the injector lines to the pump input. In a fuel-supply system using a piston-type fuel pump, the servopiston 2 is connected to the pump inclined disk.

Hydraulic equilibrium of the final control element is achieved with the aid of piston 9, which is attached to the end of the needle valve, and the drain channel 12, which transfers fuel seeping through the gaps to the pump input.

With the engine inoperative, the needle valve is held by spring 3 in the very bottom position, so that its shaped section completely covers the fuel bypass port. Spring 8 forces slide valve 4 of the sensing element to move to the very top position; then servomotor chamber A is connected to the working-liquid supply channel, and cavity B to the discharge line.

In the steady (equilibrium) regime, established by the preliminary tension of spring 8 with the aid of control lever 6, the slide valve is located at the neutral position, and its rings cover the ducts

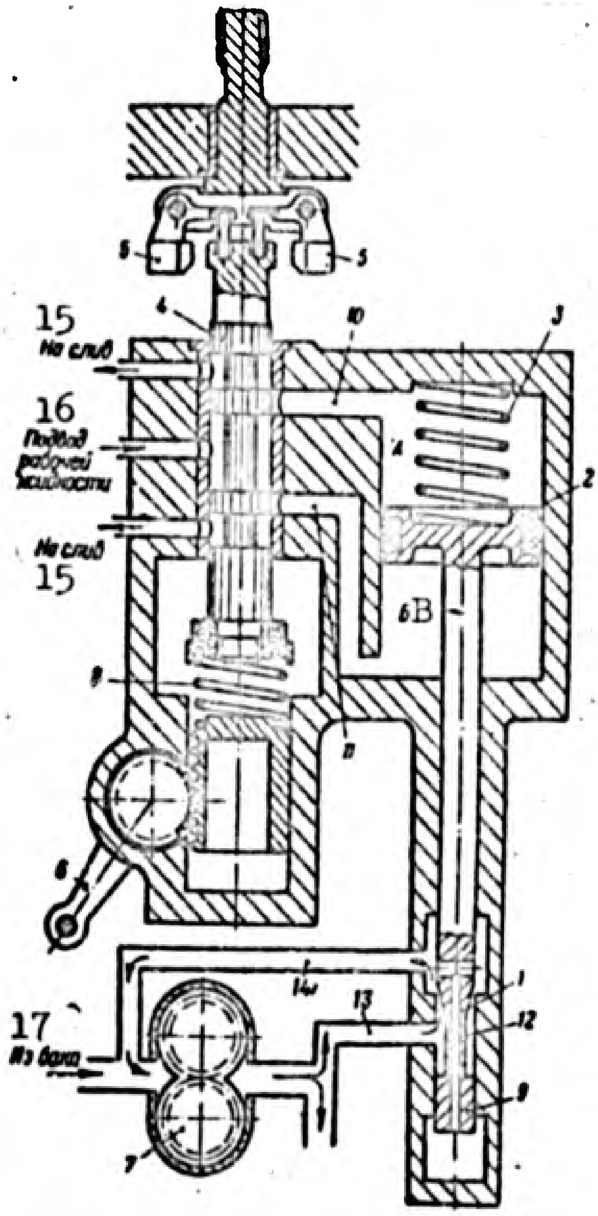

Fig. 256. Basic arrangement of control system using indirect-action speed regulator with floating hydraulic servomotor. 1) Final control element (fuel-bypass needle valve); 2) servopiston; 3 and 8) springs; 4) slide valve; 5) sensing element (centrifugal weights); 6) control lever; 7) pump; 9) piston; 10, 11, 12, 13, and 14) valves; A and B) servomotor chambers; 15) to discharge; 16) working-fluid inlet; 17) from tank.

through which fluid is supplied to the servomotor; the final control element is set by the servomotor into a position such that the available fuel feed rate equals the feed rate required for operation at the prescribed rotor speed. The difference between the pump throughput at the prescribed speed and the required fuel flow rate is handled by bypassing the excess fuel through the bypass needle valve to the pump input.

Let us look at operation of the regulator for a case in which, for example, the flight altitude increases. As the flight altitude increases, the speed of the engine rotor will increase, provided the position of the final control element does not change. The regulator should change the speed to the prescribed value; to do this, its final control element should move to a new position corresponding to a decrease in the available rate at which fuel can be fed to the engine to a value that equals the feed rate required to keep the rotor speed constant at the higher flight altitude.

The regulator comes into play as follows: as the speed increases, the weights fly apart under the effect of the centrifugal forces; this compresses the spring, and the slide valve drops down. Thus, a departure of rotor speed from the prescribed value is converted into displacement of the slide valve, which opens channel 11 so as to admit fluid under the piston and channel 10 to remove it from the chamber above the piston.

The rest of the control process may take place in various manners, depending on the servomotor response time.

With a slowly responding servomotor, the piston will slowly move upward, increasing the amount of fuel bypassed by the needle valve; as a result, the amount by which the speed departs from the prescribed value will be reduced. Here the centrifugal force due to the weights will also begin to diminish, and then the force of the spring will begin to return the slide valve toward the neutral position. Since the piston speed is proportional to the slide-valve displacement, then as the slide valve nears the neutral position, the piston velocity will be reduced to zero. The control process is concluded when the slide valve stops in the neutral position. But this becomes possible when the rate at which fuel is supplied to the engine corresponds exactly to the rate required to maintain the prescribed rotor speed at the given flight altitude. The speed at the beginning and end of the regulation process remains constant. Thus, the static error of the regulator (the residual control nonuniformity) equals zero. Since in the neutral position, the slide-valve rings close the fluid-intake channel, the servomotor piston, together with the needle valve, will be held in the required new position.

With a slow-responding servomotor, a transient process will take place without speed fluctuation, but the transient will last a very long time.

Owing to the length of the transient, slow-responding servomotors are not used with speed regulators.

With a fast-responding servomotor, transients are oscillatory in nature. As an example, let us consider the case of instantaneous readjustment from speed n_0 to a speed n_1 for which the final control element should occupy position m_1 (Fig. 257). Here the reset mechanism is transferred from position h_0 to h_1, which raises the slide valve to position y_1 (elevation of the slide valve corresponds to a minus sign, and thus y_1 is measured downward). The slide valve closes the passage through which fluid is supplied to chamber A (Fig. 256), and the discharge passage through which fluid leaves chamber B of the servomotor. Thus the servopiston drops downward, reducing the amount of fuel transferred through the bypass valve. Owing to the increase in the rate at which fuel is fed to the engine, the rotor speed will rise; this causes the centrifugal force due to the weights to increase; the spring force is overcome, and the slide valve is moved toward the neutral position. Where response time is short, the servopiston rapidly reaches the required position m_1 (point a^{II} in Fig. 257), such that the available fuel flow rate must equal the flow rate required for operation at speed n_1, while owing to the inertia of the rotor, the speed has not yet

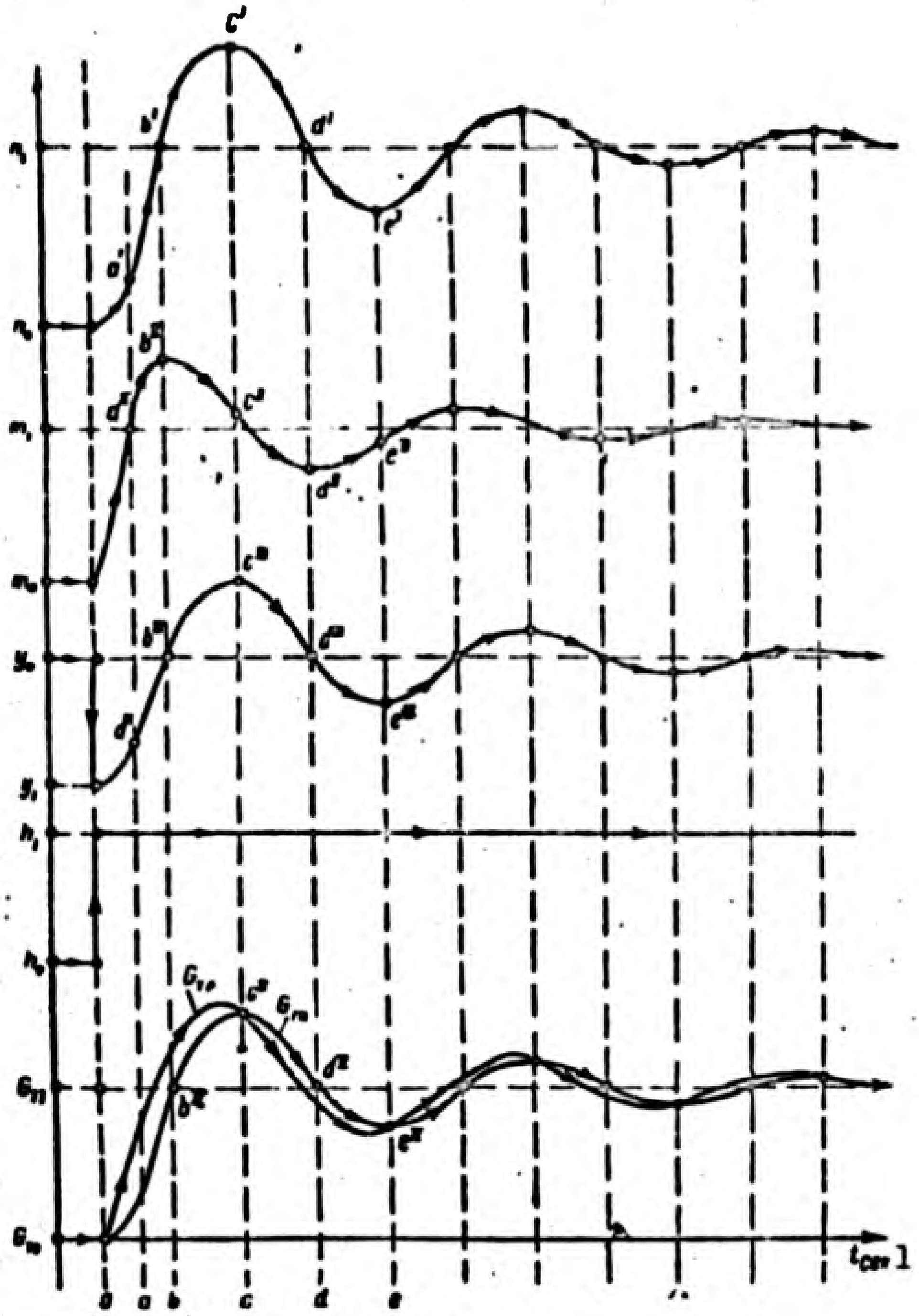

Fig. 257. Motion of control system using a speed regulator with floating servomotor during a transient, where the servomotor is of the fast-response type. 1) t, sec.

reached the prescribed value n_1 (point a^I). Consequently, the slide valve does not reach its neutral position y_0 (point a^{III}), and the control process therefore continues. Since the working fluid continues to be supplied to chamber A, the servopiston continues to move downward, which still further reduces the amount of fuel bypassed and increases the rate at which fuel is fed to the engine. Finally, after a certain period of time, the speed reaches the prescribed value n_1

(point b^{I}), and as a consequence, the slide valve occupies the neutral position (point b^{III}), which halts the delivery and removal of fluid from the servomotor piston, thus halting the servopiston (point b^{II}). With the servopiston in this position, however, the available rate at which fuel can be supplied to the engine is higher than the rate required to maintain the speed n_1 that has been reached (point b^{IV}). As a result, the rotor speed will continue to increase above the prescribed value n_1, and the slide valve will move downward away from the neutral position, now opening the duct so that fluid can be supplied to cavity B (see Fig. 256), and the fluid drained out of cavity A. The servopiston changes its direction of motion, the rate at which fuel is bypassed by the needle valve increases. The rotor speed will still increase, however, while the slide valve will move downward away from the neutral position until the available fuel flow rate equals the required rate for some speed greater than the prescribed speed (points c^{IV} and c^{I}). The curve for the required fuel flow rate G_{tp}, given in Fig. 257, shows the fuel flow rate needed for steady operation of the engine at the rotor speed reached, and in general features, duplicates the law governing the variation of the speed during the transient. The curve for the available fuel flow rate G_{tr} shows the actual rate at which fuel is supplied to the engine during the control process at any given time. Consequently, the difference between G_{tr} and G_{tp} characterizes the amount by which the fuel feed rate is too large or too small during the control process at any given time; this difference causes the engine rotor to accelerate or decelerate.

At time c, the excess fuel flow rate will equal zero. Thus the increase in speed will cease (point c^{I}), and the slide valve will be located in the position furthest away from the neutral position (point c^{III}). Consequently, the control process will continue, i.e., the servopiston will continue to move, increasing the amount of fuel by-passed by the needle valve, and moving through the position m_1 that it should occupy in order to provide the required fuel rate of flow G_{t1} for the rotor speed n_1.

The speed begins to drop, since the available rate of fuel supply to the engine becomes than that required for operation at the rotor speed actually reached. At time d, the speed becomes equal to the prescribed value n_1 (point d^I), and thus the slide valve will be in the neutral position (point d^{III}), which stops the motion of the servopiston (point d^{II}). With this servopiston position and speed n_1, however, the available fuel feed rate is less than that required for operation at speed n_1; thus the speed continues to drop, the slide valve changes the direction of motion of the working fluid in the servomotor, so that the servopiston also changes its direction of motion, i.e., it begins to move toward position m_1, so that the reduction in speed is retarded. At time e, the available flow rate is equal to the required flow rate (point e^{IV}), which halts the reduction in speed (point e^I), and stops the slide valve in a position other than the neutral position (e^{III}).

This is the way in which the control system moves over one cycle of oscillation. At time e, the position of the control system is similar to its position at the initial time. In subsequent oscillation periods, the system moves similarly, except that the amplitude is damped. At the end of the transient process, the speed will have the prescribed value, i.e., n_1, the slide valve will be in the initial neutral position y_1, while the servopiston will occupy the new required position m_1.

The duration of the transient may be slight, but the unacceptably large speed fluctuation makes the regulator unsuitable in this form for controlling TRD [turbojet-engine] rotor speeds.

In order to reduce the speed fluctuations while retaining the fast-response characteristic of the servomotor in a regulator of this type, we employ stabilizing devices that introduce proportional or elastic feedback between servopiston and slide valve.

3. INDIRECT-ACTING SPEED REGULATOR WITH PROPORTIONAL FEEDBACK

Figure 258 shows such a regulator. It differs from the regulator just discussed in the presence of proportional-feedback elements between servopiston and slide valve. The proportional-feedback elements are: the servopiston rod 15, two-arm proportional-feedback lever 16 with pivot 18, the moving sleeve 17 with ports for admission of working fluid to the servopiston, and spring 19, which sets the clearances between the proportional-feedback elements.

If we examine the kinematic arrangement of the feedback elements, we see that the position of the moving sleeve depends on the position of the servopiston. When the servopiston moves downward, the movable sleeve moves upward, and vice versa. The sleeve moves less than the piston in accordance with the ratio of the feedback-lever arms $i = d/c$.

With the engine inoperative, spring 3 holds servopiston 2 and fuel-bypass needle valve 1 in the bottom positions, where the needle valve completely covers the fuel-bypass port between the injector line and the pump input. Owing to spring 19, moving sleeve 17 is in the very top position. Its upward travel is less, however, than that of slide-valve 4 under the action of spring 8. Thus chamber A of the servomotor is connected by duct 10 to the supply line, and chamber B to the discharge line.

In the steady (equilibrium) regime, owing to the given preliminary tension of the spring 8, the bypass valve is set by the servomotor to a position such that the available fuel flow rate to the engine equals

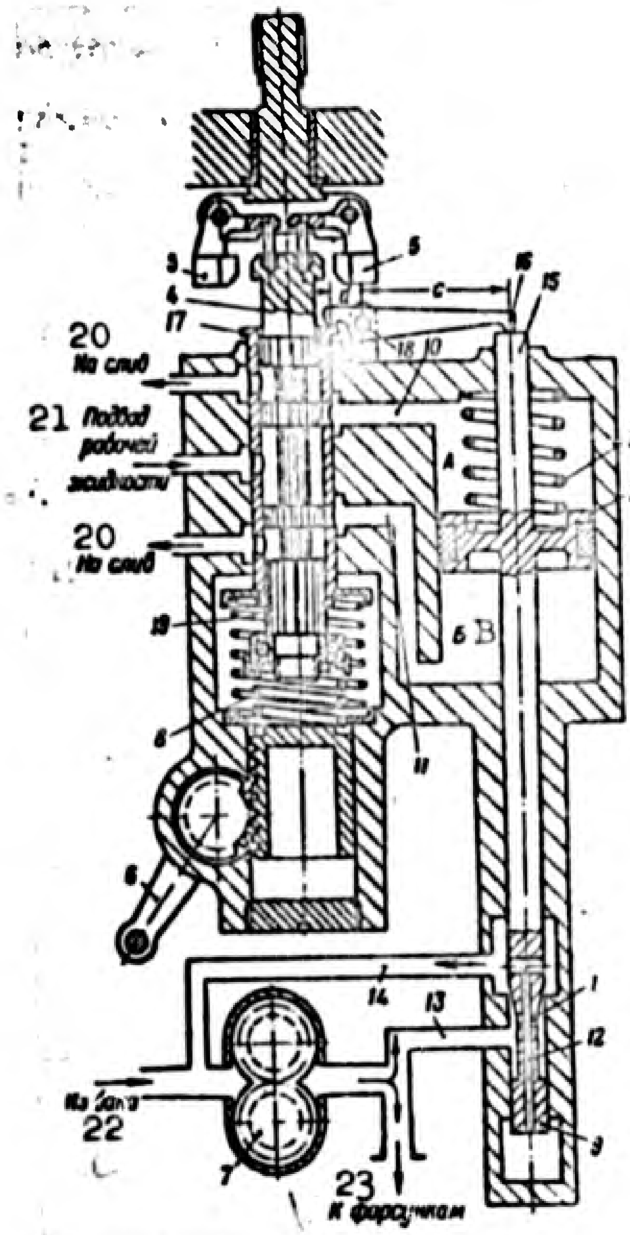

Fig. 258. Basic diagram of indirect speed-control system using servomotor with proportional feedback. 1) Fuel-bypass needle valve; 2) servopiston; 3, 8, and 19) springs; 4) slide valve; 5) centrifugal weights; 6) control lever; 7) pump; 9) piston; 10, 11, 12, 13, and 14) ducts; 15) servopiston rod; 16) feedback lever; 17) moving sleeve; 18) axis of rotation; A and B) servomotor chambers; 20) to discharge line; 21) working-fluid inlet; 22) from tank; 23) to injectors.

that required for operation at the prescribed rotor speed. The excess amount of fuel as supplied by the pump over and above the required flow rate for the given speed is transferred by the bypass valve to the pump intake side. In accordance with this position of the bypass valve and the servopiston, which is connected to it, the moving sleeve is also set with the aid of rod 15 and lever 16. In this regime, the slide valve will be in a position such that its ring covers the port in the sleeve. This acts as a hydraulic lock, which prevents the servopiston from moving while the engine is operating in the steady regime.

Proportional-feedback elements are introduced between the servopiston and control slide valve in order to eliminate excessive speed variations during the control process where a fast-response servomotor is used, and in order to prevent undesirable oscillatory transient processes.

In order to understand the way in which the regulator operates, let us consider the motion of the control-system elements during a transient (Fig. 259). Let the rotor speed at the initial time $\underline{t}$, equal to zero, drop from n_0 to n_1. In accordance with its function, the regulator should restore the prescribed value n_0 by resetting the final control element from position m_0 to the new position m_1.

The regulator operates in the following manner: a decrease in speed from n_0 to n_1 leads to a drop in the centrifugal force of the sensing-element weights, and, provided the initial spring tension remains unchanged, to motion of the slide valve owing to the spring force. The valve moves upward from the initial position y_0 to position y_1. The slide valve opens the port through which working fluid is supplied to cavity A and the port for draining fluid from cavity B (see Fig. 258); as a result, the servopiston rapidly moves downward, reducing the amount of fuel transferred by the bypass valve. The motion of the servopiston is then retarded, however, owing to the fact that it permits the sleeve to move after the slide valve, which leads to a reduction in the area of the slots through which the fluid passes, and a reduction in the speed at which the servopiston moves.

A reduction in the amount of fuel transferred at the sleeve will increase the rotor speed, increase the centrifugal force due to the weights, and change the direction of motion of the slide valve, which begins to move toward the neutral position, in the direction counter to the sleeve motion.

With a fast-responding servomotor, at time a (see Fig. 259), the servopiston reaches the required position m_1 (point a^{II}); the control process does not conclude with this, however, since the speed has not reached the prescribed value n_0 (point a^{I}), and the slide valve is not in the neutral position (point a^{IV}), so that it continues to connect cavity A of the servomotor with the working-fluid supply line, and cavity B to the discharge line. At this time, the height of the slot through which the fluid passes corresponds to the difference in the ordinate values of points a^{IV} and a^{III}. The servopiston thus moves through position m_1.

Owing to the contrary motion of slide valve and sleeve, at time f the port in the sleeve is covered by the slide-valve rings (point f^{III}), so that the working-fluid supply and discharge lines to the servomotor are cut off; as a result, the servopiston comes to a stop (point f^{II}). As a consequence, the sleeve, which is connected to the servopiston, also comes to a stop (point f^{III}), but the slide valve continues to move to the neutral position, since the rotor speed continues to approach the prescribed value; at this time, the slide valve changes the direction of motion of the fluid in the servomotor: chamber A is connected to the discharge line and chamber B to the working-fluid supply line. Thus the servopiston and attached sleeve first stop and then change the direction of motion, moving toward the neutral position. The motion of the sleeve is now opposite in direction to that of the slide valve. At time e, the sleeve with its ports moves under the slide-valve rings, thus terminating the control process, since the increase in rotor speed also ends at the same time.

Looking at the graphs for the transient in the case of a regulator with proportional feedback, and comparing them with the similar graphs

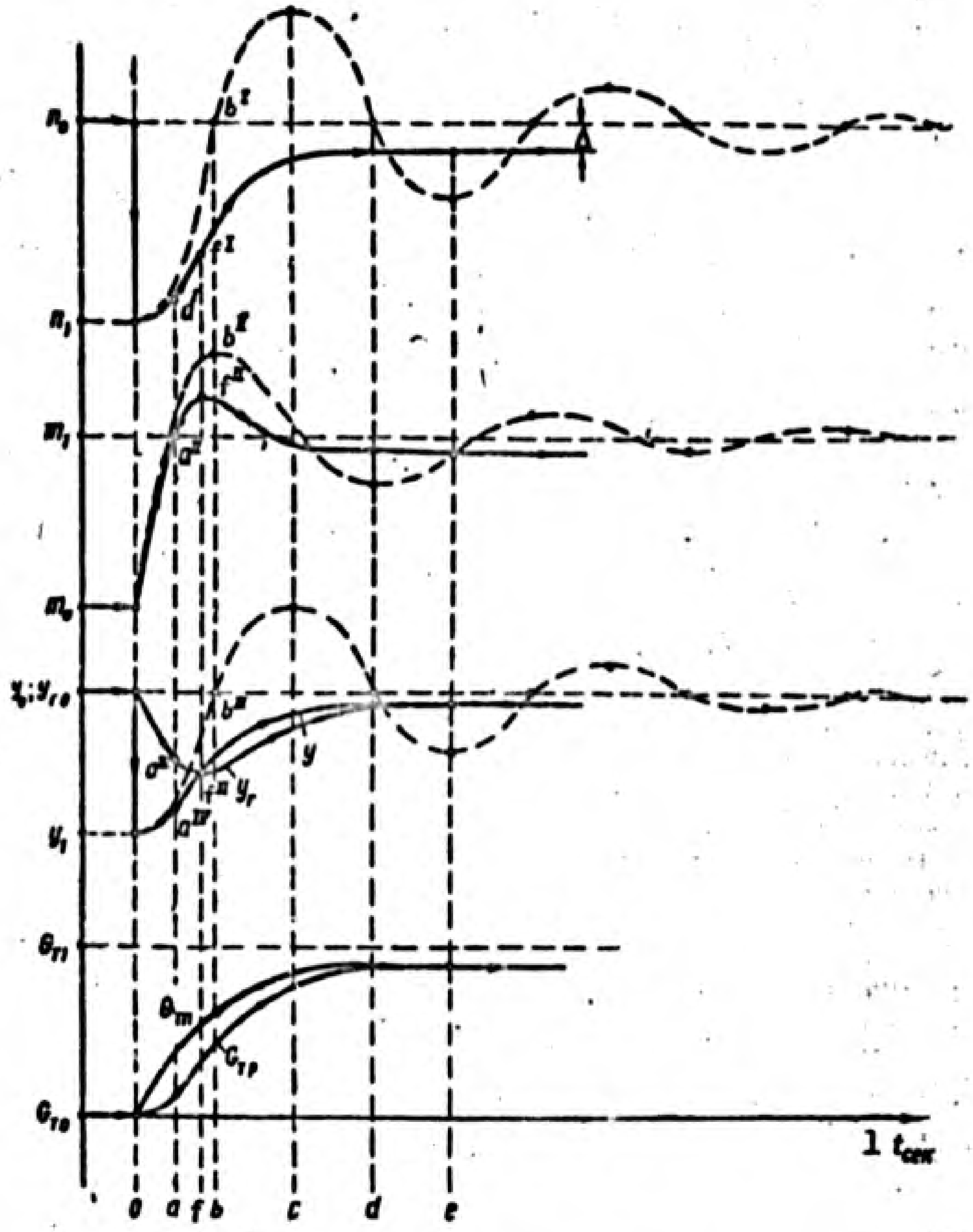

Fig. 259. Motion of control system using servomotor with proportional feedback during transient process. 1) t, sec.

for a regulator without feedback, shown in Fig. 259 by the dashed lines, we see that a speed fluctuation during the control process is prevented by the fact that when the servopiston moves, the proportional-feedback lever and spring cause the moving sleeve to follow the slide valve so as to anticipate cutoff, which occurs not at time b (point b^{III}), as in a regulator without a stabilizing device, but somewhat earlier – at time f (point f^{III}). The servopiston thus remains in a position near to its required position m_1, owing to which the speed smoothly approaches the required value without fluctuation (hunting).

A substantial drawback to this regulator lies in the fact that at the conclusion of a control process initiated by a change in altitude or flight speed, the speed does not return to the prescribed value: when the regulator adjustment is unchanged, and the altitude decreases or flight speed increases, the rotor-speed value will be lower, and vice versa.

Actually, in order to maintain a prescribed speed at lower flight altitude, the servopiston must move downward, increasing the rate at which fuel is supplied to the engine, and must remain in this position. At the same time, there is a change in the position of the sleeve, which leaves the initial neutral position, moving upward (see Fig. 258). Since the control process ends at the instant at which the slide-valve rings close the sleeve port, in the new neutral position, both the slide valve and sleeve will be higher than in the initial situation. As a consequence, the sensing-element spring will be less compressed, and sensing-element equilibrium can occur only at lower rotor speed.

Thus, the introduction of proportional feedback reduces or completely eliminates (as in the case considered above) the fluctuation in speed that occurs during the regulation process (owing to anticipated cutoff of the supply of working fluid to the servopiston); at the same time, however, the indirect-acting speed regulator becomes a static (proportional) device.

The regulator static error depends on the arm ratio of the feedback lever $i = d/c$ (see Fig. 258). The error is greater the larger this ratio. As the quantity $\underline{i}$ diminishes, the tendency of the system to oscillation increases. In the limit, where $i = 0$, we have a regulator without feedback.

4. INDIRECT-ACTION SPEED REGULATOR WITH PROPORTIONAL-PLUS-INTEGRAL SLIDE-VALVE FEEDBACK

In order to eliminate the static error (residual control nonuniformity), it is necessary at the end of a control process initiated by a change in load (altitude or flight speed), for the moving sleeve to return to the same position as at the beginning of the control process.

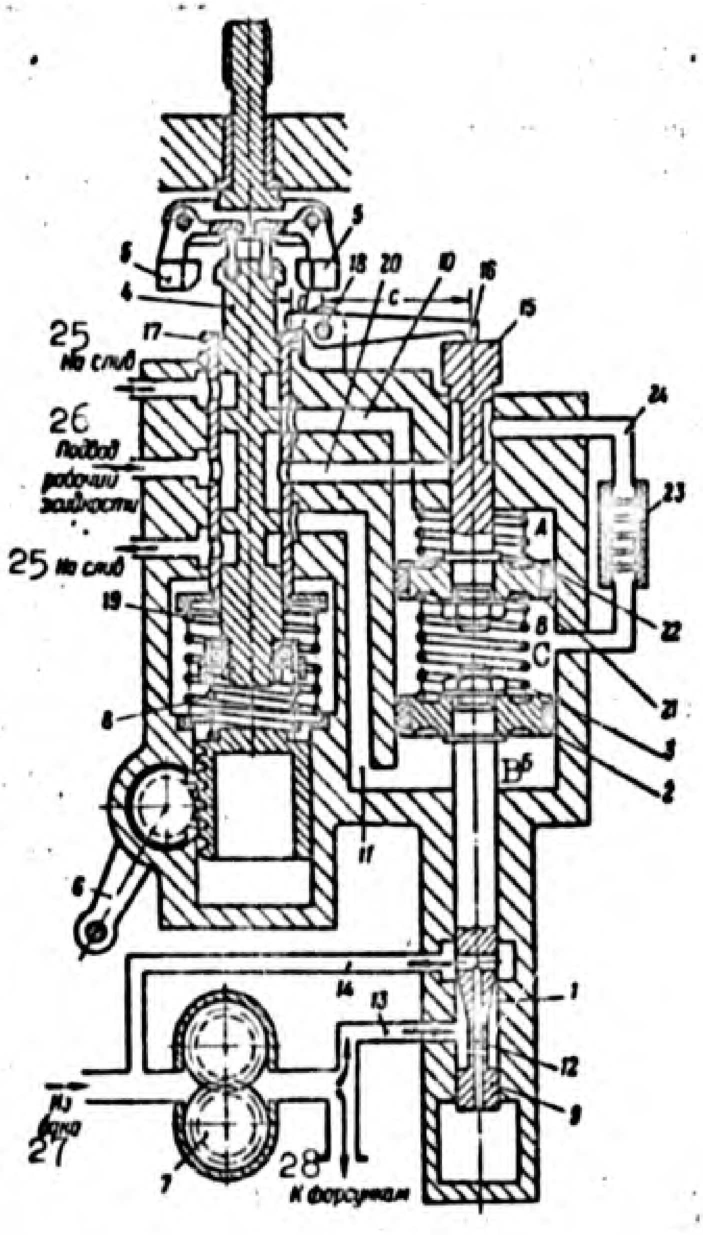

Fig. 260. Basic diagram of control system using speed regulator with proportional-plus-integral feedback slide valve. 1) Fuel-bypass needle valve; 2) servopiston; 3, 8, and 19) springs; 4) slide valve; 5) centrifugal weights; 6) control lever; 7) pump; 9) piston; 10, 11, 12, 13, 14, 20, and 24) ducts; 15) rod of proportional-plus-integral piston; 16) feedback lever; 17) moving sleeve; 18) axis of rotation; 21) proportional-plus-integral piston; 22) spring for proportional-plus-integral piston; 23) throttling element; A, B, and C) chambers; 25) to discharge line; 26) working-fluid supply line; 27) from tank; 28) to injectors.

For this to be the case, the feedback between servopiston and slide valve should not be proportional, but elastic (proportional-plus-integral), i.e., here the feedback lever 16 (Fig. 260) is connected not to the servopiston rod, as in a regulator with proportional feedback, but with rod 15 of the proportional-plus-integral piston 21. At the same time, rod 15 of the proportional-plus-integral piston acts as a slide valve, controlling the rate at which the working fluid is fed to cavity B between the servopiston and the proportional-plus-integral piston or the rate at which it is drained out of this cavity.

When the proportional-plus-integral slide valve moves downward, the working fluid moves into the chamber C between the pistons through duct 20, the ring groove of the slide valve, duct 24, the throttling element 23; when the proportional-plus-integral slide valve is in the top position, its ring groove joins this chamber to the discharge line.

With the engine inoperative, the control slide valve 4 is held in the very top position by spring 8 of the sensing element, against the stop for the centrifugal weights. In this case, the slide-valve rings are so located with respect to the sleeve ports that cavity A above the proportional-plus-integral piston is connected to the working-fluid intake duct, while cavity B under the servopiston is connected to the discharge line. The proportional-plus-integral piston and slide valve are held by the spring 22 of the proportional-plus-integral piston in the very bottom position against the stop of the proportional-plus-integral slide valve; here the cavity C between the pistons is connected through the groove in the proportional-plus-integral slide valve with the working-fluid intake duct. The moving sleeve is shifted by its spring to the very top position.

The sleeve moves less than the proportional-plus-integral piston, in accordance with the arm ratio of the feedback lever $i = d/c$.

The spring of the final control element itself holds the servo-

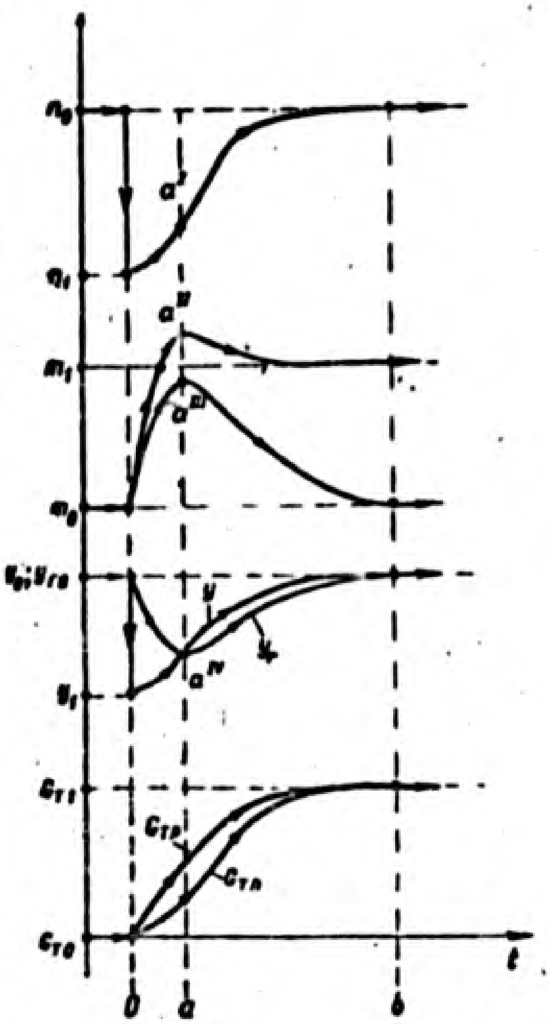

Fig. 261. Motion of control system with proportional-plus-integral feedback during transient.

piston in the very bottom position, so that the needle valve connected with it completely covers the vent through which fuel is transferred from the injector line to the pump input.

The regulator elements also occupy this position when the engine is first being started.

When the engine leaves a steady operating regime, set by the initial tension of the sensing-element spring, the proportional-plus-integral piston is acted on by fuel-pressure forces so that it sets the elements of the proportional-plus-integral feedback device to the neutral position. In this position, the proportional-plus-integral slide valve lower ring disconnects the interpiston cavity C and the working-fluid supply line, while the upper ring isolates this cavity from the discharge line. The servopiston of the needle bypass valve holds it in a position such that the available rate of flow of the fuel to the engine equals the flow rate needed for operation at the given rotor speed. The rings of the control slide valve close the port in the sleeve, fixing the position of the servopiston and proportional-plus-integral piston.

As an example, let us consider the operation of a regulator where at the initial time, the speed of the rotor drops from n_0 to n_1. Then the centrifugal weights will move together, and the springs will lift the control slide valve upward to position y_1 (Fig. 261), so that the slide valve will open the inlet through which working fluid reaches chamber A (see Fig. 260), and chamber B will be connected to the discharge line. At the first instant of time, both pistons, together with the volume of the interpiston chamber C will move downward at the same speed, while spring 19 forces the moving sleeve upward after the control slide valve to early cutoff. Thus, at the initial instant of time, the regulator acts as a regulator with proportional feedback.

At the next instant of time, "elasticity" begins to appear in the feedback: the downward displacement of the proportional-plus-integral piston opens the duct through which working fluid is supplied to the interpiston cavity; the volume of this cavity is increased, and as a result the servopiston moves downward somewhat more rapidly than the proportional-plus-integral piston. The increase in the rate at which fuel is supplied to the engine increases the rotor speed and moves the slide valve downward, against the direction of motion of the sleeve, owing to which the area for passage of fluid in the sleeve ports is decreased, the motion of the piston is slowed down, and both pistons stop in the cutoff position (see point a^{IV} in Fig. 261): the servopiston in position a^{II} and the proportional-plus-integral piston in position a^{III}. The speed at this time has still not yet reached the prescribed value n_1 (point a^{I}), and it continues to increase. Then the slide valve, continuing to move downward, connects chamber A (Fig. 260) with the discharge line, and chamber B with the working-fluid supply line. Here both pistons change their directions of motion; the proportional-plus-integral piston will now move toward the neutral position more rapidly than the servopiston, however, since working fluid continues to arrive in the interpiston chamber C, while chamber A is connected to the discharge line. The supply of working fluid to the interpiston chamber is cut off at the instant the proportional-plus-integral piston with the proportional-plus-integral slide valve takes up its initial neutral position. Then the moving sleeve will also occupy its initial position. As a consequence, the control slide valve will also occupy its initial neutral position as soon as the rotor speed returns to the initial value. In accordance with decreasing altitude or increasing flight speed, the servopiston will take up a lower position. Thus, exactly the same neutral control-slide-valve position (and exactly the same prescribed rotor speed) can correspond to different positions of the servopiston, depending on the altitude and flight speed. As a consequence, the static regulation error becomes equal to zero, exactly as with floating regulators.

The required rate of filling or emptying of the interpiston chamber during the control process is provided by the installation of a throttle element 23 (see Fig. 260), or some other resistance along the path of the motion of the working fluid. In the limiting case, in which the diameter of the throttle-element vents equals zero (infinitely large hydraulic resistance), the regulator under discussion will become a speed regulator with proportional feedback.

5. CORRECTED-SPEED REGULATOR FOR TRD ROTOR

In addition to regulators that keep the physical rotor speed constant, turbojet engines also employ regulators that keep the corrected speed constant. The corrected speed is given by the following expression: $n_{pr} = n/\sqrt{T^*_N}$, where n is the physical rotor speed and T^*_N is the stagnation temperature of the air at the compressor intake.

This formula shows that if the temperature T^*_N varies, it is necessary to change the physical rotor speed in order to keep the corrected speed constant; this variation should be governed by the law $n = \text{const}\sqrt{T^*_N}$. The simplest way of satisfying the required law governing the variation in physical rotor speed is to introduce an influence on the regulator adjustment mechanism that is associated with a gauge measuring the air temperature at the compressor intake.

Figure 262 shows one possible basic arrangement for an adjustment mechanism using a lever mechanism. The tension on spring 1 and, consequently, the prescribed speed depends on the position of control lever 2 and the amount of deformation of the temperature element that measures the temperature T^*_N.

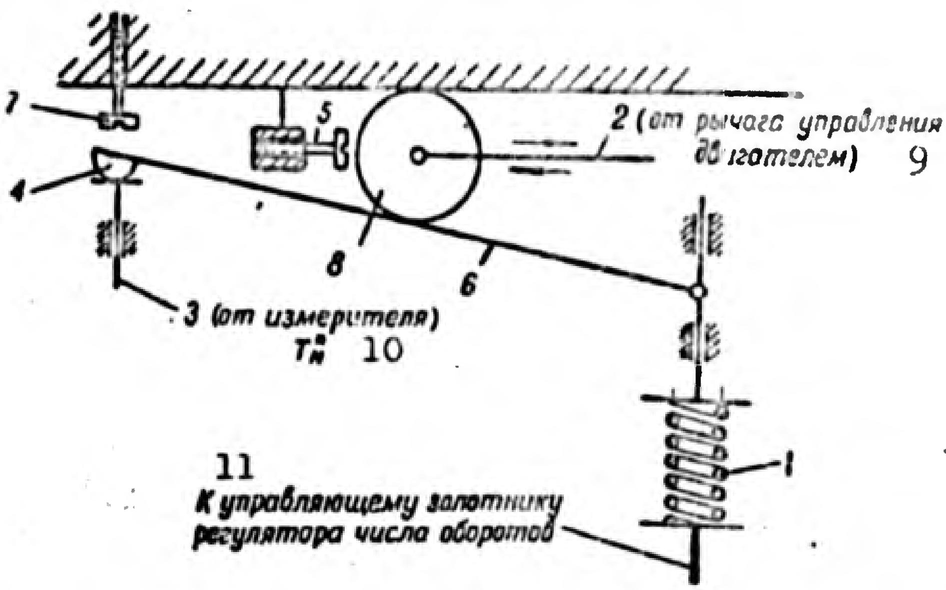

Fig. 262. Possible basic arrangement of adjustment mechanism for corrected-speed regulator. 1) Spring; 2) control-lever rod; 3) thermometer-element rod; 4) cam; 5 and 7) stop screws; 6) lever; 8) roller; 9) from engine control lever; 10) from gauge measuring T^*_N; 11) to control slide valve of speed regulator.

Moving the control lever to the stop screw 5 adjusts the device for maximum corrected rotor speed. As the temperature increases, the tension on spring 1 increases, thereby increasing the physical rotor speed. At a certain temperature, lever 6, which turns about roller 8, will reach the stop screw 7, and the spring tension will cease to depend on the temperature T^*_N; in this case, the physical speed will reach its maximum value.

Chapter 21

CONSTANT FUEL FEED-RATE REGULATORS

1. OPERATING STABILITY OF ENGINE USING GEAR-DRIVEN FUEL PUMP WITH CONSTANT RATE OF FUEL SUPPLY

It has been shown in Chapter 19 that a wide-range centrifugal-type speed regulator provides the required quality of automatic control only in an engine operating-regime zone ranging from cruising to maximum speed, inclusive. In remaining operating regimes, the speed is controlled manually, which is possible if an engine with a gear-driven fuel pump has positive self-regulation in a given position of the engine control lever. A TRD with a gear-driven fuel pump not having a regulator has a region of unstable operation (as we have mentioned earlier) at a rotor speed close to the idle-running speed, owing to the fact that the fuel feed rate depends on the rotor speed. If, however, a special regulator is used to keep the fuel feed rate independent of the rotor speed for a given position of the control lever, an engine with an engine-driven fuel pump will operate stably in all regimes, including the idle-running regime.

We shall analyze the operating stability of an engine using an engine-driven fuel pump with constant fuel feed rate that does not depend on rotor speed.

As we can see from Fig. 263, when the rotor speed departs from the prescribed value n_1 with a constant fuel feed rate, the available and the required fuel feed rates will no longer be equal: the available fuel feed rate will remain constant while the required fuel feed rate

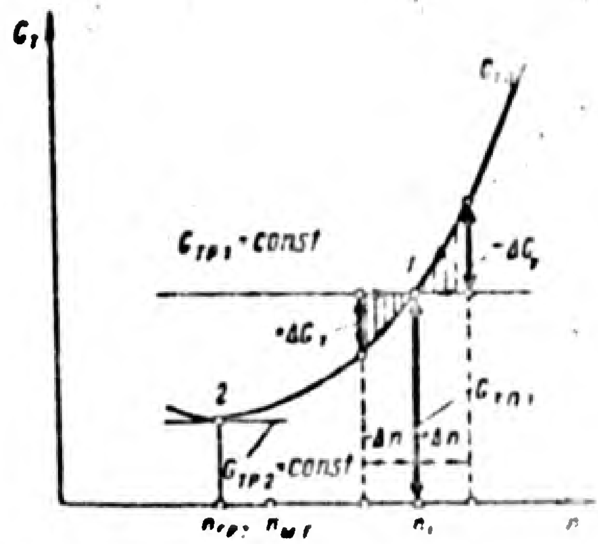

Fig. 263. The problem of operating stability of a TRD [turbojet engine] using fuel pump with constant fuel feed rate.

will change.

If the rotor speed drops by a small amount Δn, the fuel feed rate required to operate the engine at a rotor speed equaling $n_1 - \Delta n$ becomes less than the available fuel feed rate; thus the excess fuel (see $+\Delta G_t$ in the figure) supplied to the engine goes to increase the rotor speed to the prescribed value n_1. Under such conditions, the turbine torque exceeds the compressor torque, i.e., the rotor speed will return to the prescribed value n_1 owing to the positive excess torque at the engine rotor.

If, however, the rotor speed increases by a small amount Δn, the fuel feed rate required to operate the engine at speed $n_1 + \Delta n$ will become larger than the available feed rate. As a result of the inadequacy of the fuel feed rate, therefore (see $-\Delta G_t$ in the figure), the rotor speed will drop to the prescribed value n_1 owing to the negative excess torque at the engine rotor.

Thus, at a rotor speed n_1 and constant fuel feed rate G_{tr1}, required for the given regime, the engine will operate stably, i.e., it possesses positive self-regulation.

By a similar argument, we can see that the engine will also operate stably with other constant fuel feed rates corresponding to operating regimes lying on the ascending branch of the curve for the required fuel feed rate.

The boundary separating the regions of positive and negative engine self-regulation corresponds to the minimum required fuel feed rate (point 2). To the left of point 2, an engine with an engine-driven fuel pump and constant fuel feed rate will possess negative self-regulation. The rotor speed n_{grt} corresponding to point 2, is called the <u>limiting</u> speed for constant fuel feed rate. Since $n_{grt} < n_{mg}$, an engine with an engine-driven fuel pump and constant fuel feed rate will operate stably over the entire range of working speeds.

In order to maintain the rate at which fuel is fed to the engine constant with a specific accuracy for a given position of the throttle valve, two types of regulator are used: regulators maintaining a constant pressure drop at the throttle valve or regulators maintaining a constant fuel pressure ahead of the throttle valve.

With such regulators, the sensing element reacts to the pressure drop at the throttle valve or to a change in fuel pressure ahead of the valve, respectively.

The sensing element may simultaneously act as a final control element as, for example, in a reducing valve, or can control the position of a final control element with the aid of a servomotor. In the first case we have a <u>direct-acting</u> regulator, and in the second case an <u>indirect</u> regulator. A throttle valve is used to reset the regulator to another rotor speed. An increase in the area of the valve flow area with constant pressure drop across the valve (or constant pressure ahead of the valve) will lead to an increase in the fuel flow rate and the engine-rotor speed.

2. THROTTLE VALVES

The throttle valve is used to control the operation of an engine from the cockpit so as to change the engine operating regime. In the majority of fuel-supply systems, the throttle valve is also used as an engine shutoff valve.

The basic element of a throttle valve is the metering element, which is positioned in the valve housing with the aid of the control lever. Here the metering element (a needle valve or rotary slide valve) is used to vary the fuel-flow area in the throttle valve in accordance with a specific law. This is done by introducing a shaped section at the metering element or by using specially shaped ports in the valve housing. The law governing the variation in flow area in the throttle valve is so chosen as to provide the required relationship between the valve throughput and the position of the valve control lever that will ensure that the required relationship will be obtained between the engine-rotor speed and the position of the valve lever. Where the system contains an automatic centrifugal speed regulator, appropriate shaping of the metering element may be used to provide a required transition from manual to automatic speed control. In this case, the throttle-valve control lever is connected not only to the metering element, but also to a coupling that goes to the mechanism that adjusts the centrifugal regulator for the desired engine-rotor speed.

Throttle valves are classified into the following types:

1) longitudinal adjustment of metering element;

2) rotary adjustment of metering element about its own axis;

3) longitudinal and rotary adjustment of metering element.

The fundamental elements of the first type of throttle valve are: the adjustable needle 1 (Fig. 264) with a specially shaped end, a lever 2 connected to shaft 7, which is attached to gear 3, which engages a rack machined into the needle.

Where a centrifugal speed regulator is used, the shaft usually bears an integral gear 8, which engages a rack cut into coupling 4 of the regulator adjusting mechanism. When the valve lever is moved, it produces a simultaneous longitudinal displacement of the throttle-valve needle and the adjusting-mechanism coupling.

When the throttle-valve lever is set against the engine shut-down stop, the body of the needle covers all fuel-entry ports, and as a result the supply of fuel from pump to injectors is halted.

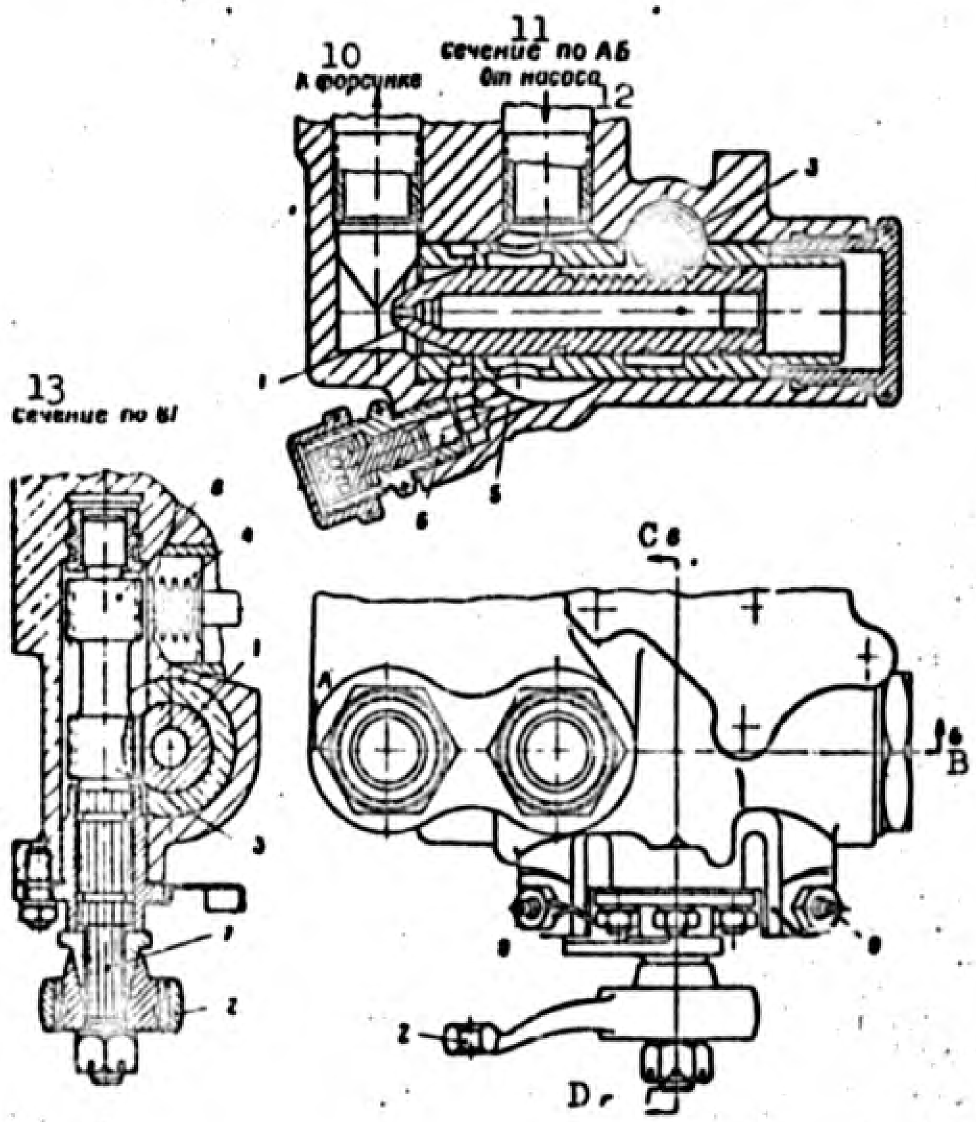

Fig. 264. Throttle valve with longitudinal adjustment of metering needle. 1) Needle; 2) lever; 3 and 8) gears; 4) coupling; 5) idling duct; 6) idling screw; 7) shaft; 9) stop screw; 10) to injector; 11) section through AB; 12) from pump; 13) section through CD.

When the throttle-valve lever is set to the "idle running" posi-

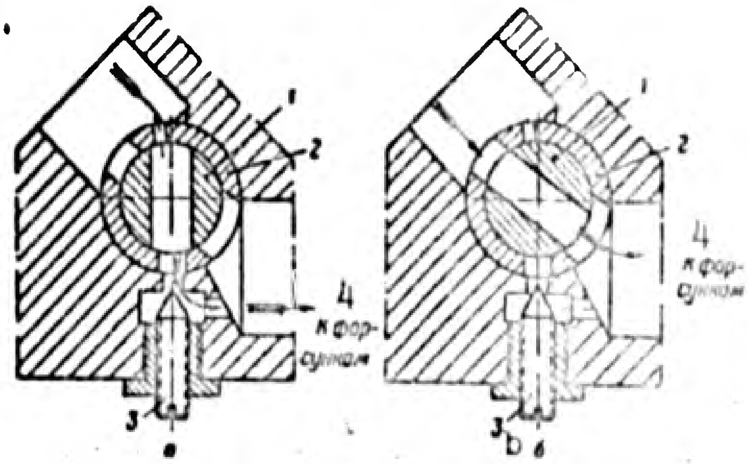

Fig. 265. Throttle valve with rotary slide valve. a) "Idle-running" position; b) "full gas" position; 1) slide valve; 2) bushing in slide-valve housing; 3) idling screw; 4) to injectors.

tion, the needle opens a path for the fuel through the idle-running valve 5 alone. This fuel passage contains a regulating element -- the idle-running screw 6, which is used to adjust the flow area and, consequently, the fuel flow rate and rotor speed when the engine is running idle. A reduction in flow area leads to a reduction in engine-rotor speed, and the reverse relationship also holds.

When the valve control is turned further, the shaped portion of the needle opens the way for fuel to enter through the main valve port.

The off and full-open positions of the valve lever are adjusted with the aid of stop screws 9 in the valve housing.

A throttle valve with a rotary metering element (Fig. 265) has a slide valve 1 with specially shaped ducts or tapers, and ports in sleeve 2 of the slide-valve housing, which are arranged in a suitable fashion.

In the engine-shutoff position, the slide valve completely covers the fuel-entry ports. In the "idle running" position of the valve, fuel is supplied only through the idling duct, whose flow area is adjusted by the idling screw 3. As the valve is opened further, the flow area increases in accordance with the prescribed law.

Figure 266 shows a throttle valve which can turn about its own

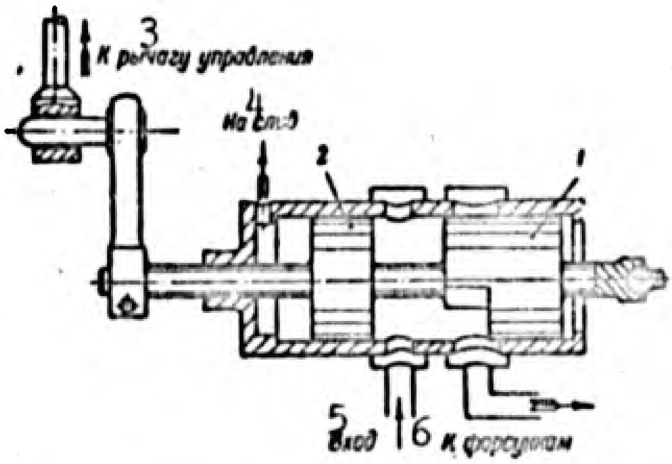

Fig. 266. Throttle valve with longitudinal and rotary adjustment of metering element. 1 and 2) Rings; 3) to control lever; 4) to discharge line; 5) inlet; 6) to injectors.

axis and also move axially with respect to a stationary sleeve which has a rectangular port. By turning the valve about its own axis, the pilot can change the valve flow area when the engine operating regime changes. To do this, there is a specially shaped bevel on control ring 1. Axial displacement of the valve is used to adjust the fuel feed rate when external conditions change (see Chapter 22 below, section "Fundamental Types of Indirect Speed-Control Systems and Their Properties").

In the engine-shutoff position, the valve strikes a stop at the extreme left-hand position. Ring 2 ensures that the valve is in hydraulic balance.

3. REGULATORS HOLDING FUEL PRESSURE DROP ACROSS THROTTLE VALVE CONSTANT

Direct-acting regulator. Figure 267 shows the basic arrangement of a regulator connected into a fuel-supply system using a gear pump.

The regulator has a differential valve 1, spring 2, and spring screw 3. The differential valve acts as a sensing element; it responds to the pressure drop across the valve (i.e., to the difference in the pressures ahead of and behind the valve) and as a final control element that transfers fuel from the line ahead of the throttle valve to the pump intake. The value of pressure drop maintained by the valve is set by the initial tension on spring 2 with the aid of screw 3. The valve is thus acted on by the force F_{pr} due to the spring and the force due to the difference in fuel pressures before and after the valve.

With the engine inoperative, the valve is pressed against a stop by the spring, and it closes the vent through which fuel is transferred from the line ahead of the throttle valve to the pump intake.

As the rotor speeds up while the engine is being started, the fuel pressure drop across the valve rises to the value set by the spring tension. If the valve control lever is in the "idle running" position, the prescribed pressure drop is reached at a certain rotor speed that is less than the idle-running speed. Until this speed is reached, all fuel delivered by the oscillating element of the pump will be sent to the engine through the throttle valve.

As the rotor speed increases further, the pump throughput and fuel pressure at the valve increase, and as a result the valve opens the fuel bypass vent between the pump outlet side and pump inlet side by the amount needed for the force due to fuel pressure at the pump and the spring force to be equal in each steady regime.

As the rotor speed increases, the amount of fuel transferred increases, which increases the displacement of the valve, the spring-compression force and, consequently, the fuel pressure. Thus the pressure drop across the valve and the rate at which fuel is supplied to the engine will not remain absolutely constant, but will increase somewhat.

Thus, the direct-acting regulator under discussion possesses a residual control error, i.e., it is a static device whose error rises as the amount of fuel bypassed increases. This is undesirable, natu-

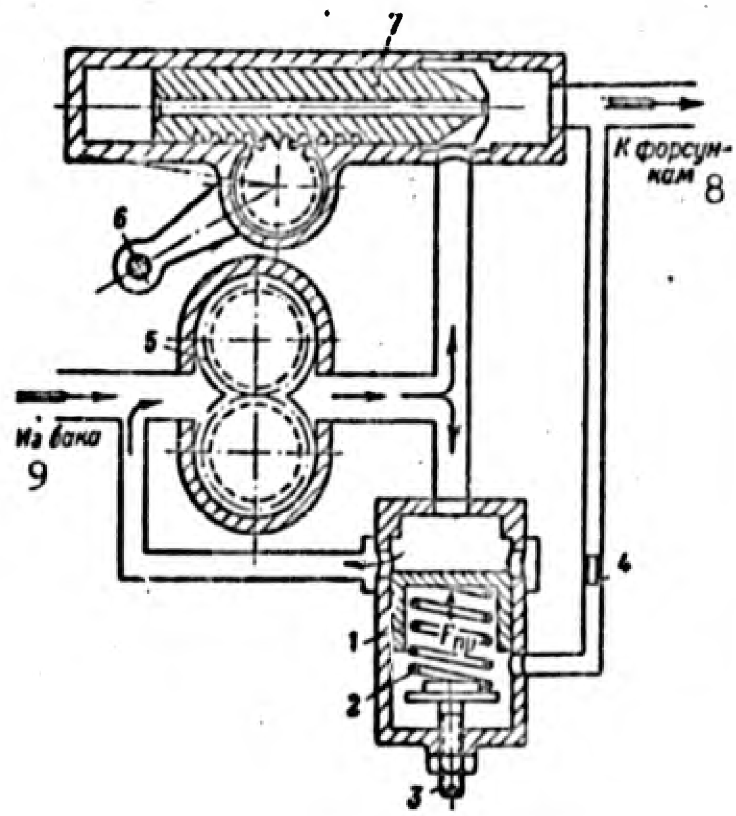

Fig. 267. Basic arrangement of direct-acting constant pressure-drop regulator. 1) Differential valve; 2) spring; 3) spring screw; 4) nozzle; 5) pump; 6) throttle-valve lever; 7) throttle-valve needle; 8) injectors; 9) from tank.

rally, as it affects engine operating stability.

The regulation error may be reduced by decreasing spring stiffness and increasing the fuel bypass area, for example, by increasing the diameter of the valve. This will increase the tendency of the valve to go into oscillation, however. Thus where large amounts of fuel must be bypassed, a regulator using a two-stage valve is used to maintain the fuel pressure drop across the valve constant; it takes the form of a combination of a valve 1 (Fig. 268) which holds the fuel pressure drop constant across the throttle valve, and a bypass valve 8.

The bypass valve is acted on by the difference in fuel pressures before and after the constant-drop valve 1 (i.e., the difference in the fuel pressures after the pump and before the throttle valve), and the force due to spring 9. The bypass valve transfers excess fuel out of the high-pressure system and within this chamber maintains a pressure that exceeds the pressure ahead of the throttle valve by a con-

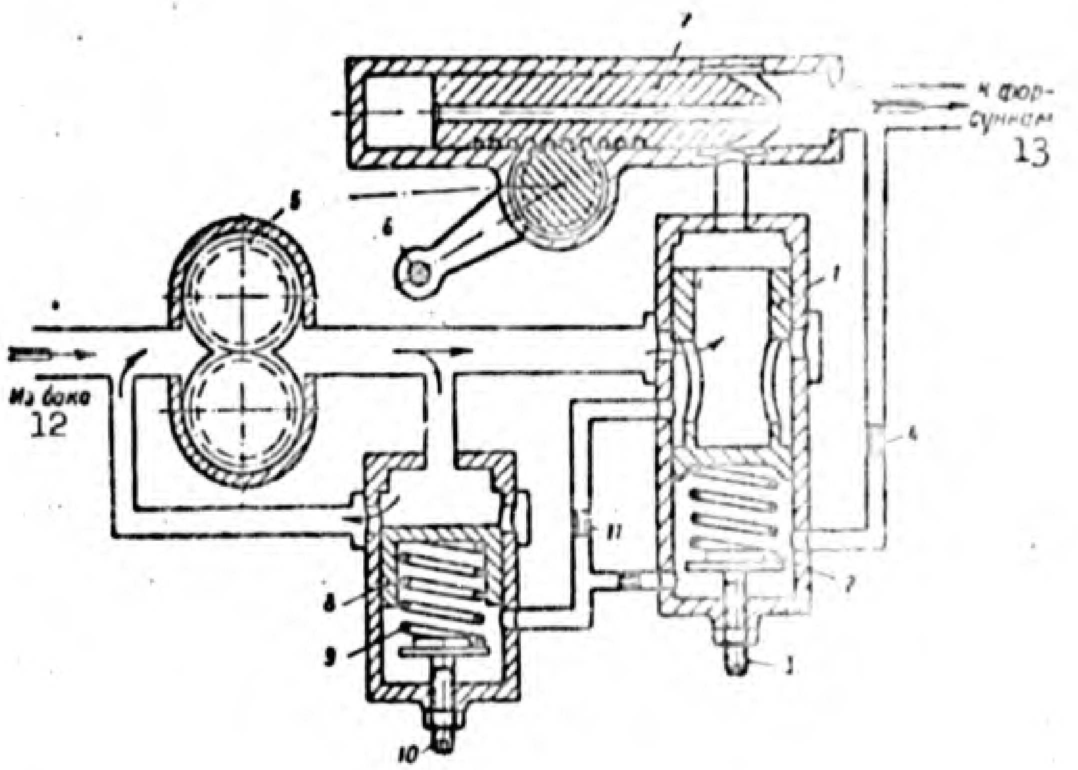

Fig. 268. Basic arrangement of direct-acting regulator with bypass valve used to hold pressure drop constant. 1) Constant-pressure-drop valve; 2) spring; 3) screw; 4 and 11) nozzles; 5) pump; 6) throttle-valve control lever; 7) valve needle; 8) bypass valve; 9) spring; 10) screw; 12) from tank; 13) to injectors.

stant amount adjusted by screw 10 of spring 9 on the bypass valve. The fuel pressure ahead of the throttle valve is in turn higher than the pressure after the valve (ahead of the injectors) by a constant amount adjusted with the aid of screw 3 and spring 2 of constant-drop valve 1.

Valve 1 operates by throttling the pressure of the fuel passing through the valve. If the pressure drop across the throttle valve rises in comparison with the pressure set by the tension of spring 2, the valve, acted on by this drop, moves so as to reduce the fuel flow area. The fuel that does not pass through the valve is transferred from the line by valve 8 to the pump intake side.

Thus, the constant-pressure-drop valve corrects the operation of the bypass valve, reducing the control error for the pressure drop across the valve; it cannot eliminate this error completely owing to the fact that this element itself possesses a definite static error.

The pressure-drop control error is also reduced by the use of in-

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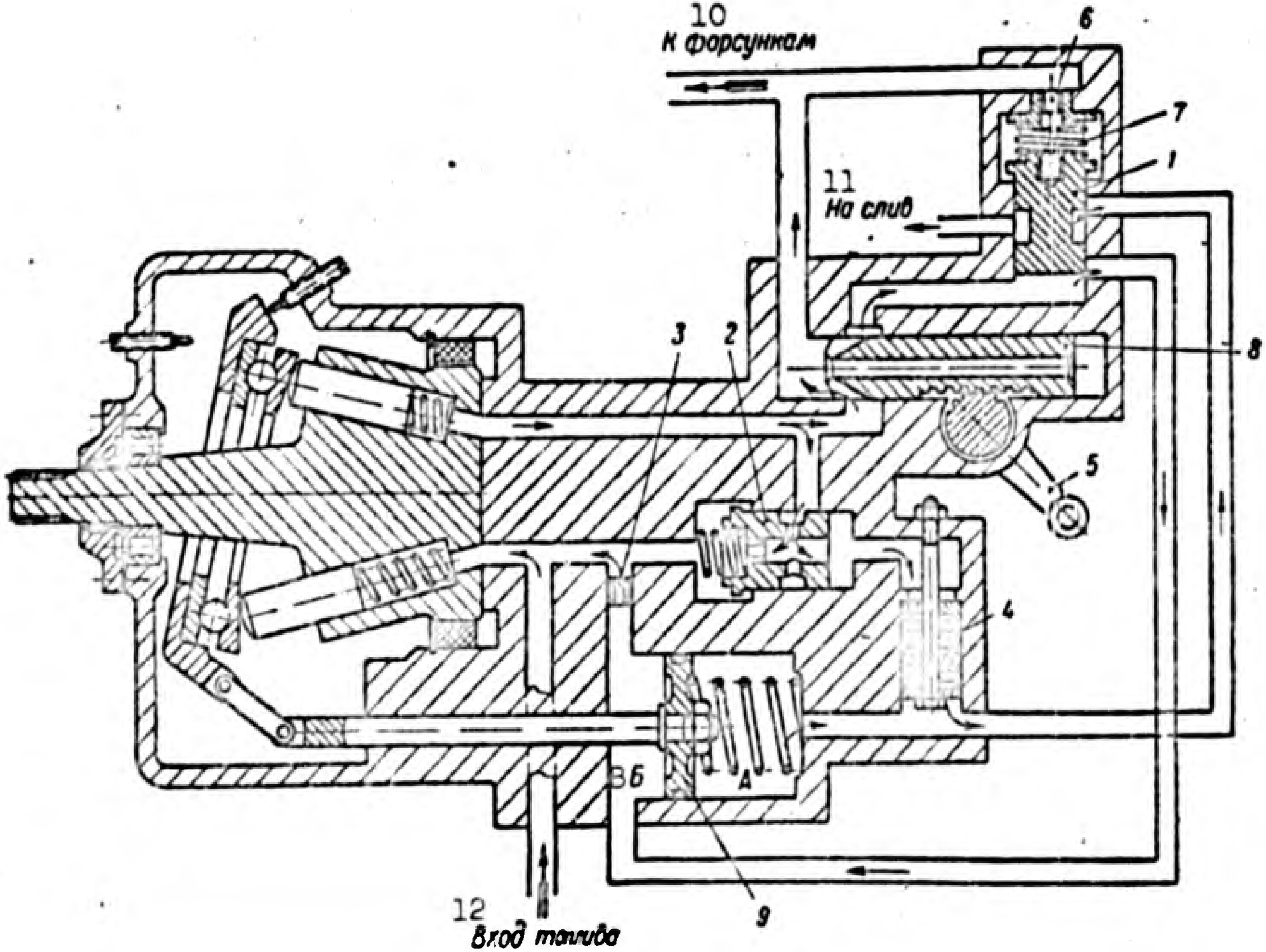

Fig. 269. Basic arrangement of regulator holding fuel pressure drop across throttle valve constant. 1) Differential valve; 2) constant-pressure valve for working fluid; 3 and 6) nozzles; 4) throttle element; 5) throttle-valve control lever; 7) spring; 8) throttle-valve needle; 9) servopiston; A and B) chambers; 10) to injector; 11) to discharge line; 12) fuel intake.

1/2 FRAMES

direct regulators.

Indirect regulator. Such a regulator is shown in Fig. 269. The regulator has a differential valve 1, which responds to a change in the fuel pressure drop across the throttle valve, and which exerts control through the supply of working fluid to chamber B and the discharge of this fluid from chamber A in the servopiston cylinder; there is also a valve 2 to maintain the working-fluid pressure constant ahead of the servomotor, a nozzle 3 to control the discharge of working fluid from chamber B of the servopiston cylinder, a nozzle regulating the flow of fluid into chamber A and, finally, a throttle-valve needle 8.

Let us see how the regulator operates to maintain the fuel pressure drop across the throttle valve constant.

1. Operation of Regulator when Throttle Valve Moves to New Position in Order to Change Engine-Rotor Speed

With the engine inoperative, there is no fuel pressure in the line, and therefore the servopiston is held by a spring in the position for which the angle of inclination of the pump disk is greatest.

As the engine is started, when the throttle valve is in a position corresponding to idle running, the servopiston holds the disk at the maximum angle of inclination until there is a pressure drop across the valve that has the value established by the tension of the differential-valve spring. This pressure drop is reached at a rotor speed below the idling rotor speed.

As the speed increases further to a value corresponding to the idle-running regime, the differential valve rises and moves the servopiston to the right (see Fig. 269) decreasing the angle of inclination of the plate. Thus the prescribed constant drop across the valve is maintained.

In order to increase the rotor speed, it is sufficient to turn the throttle-valve control lever 5 so as to increase the valve flow area. Initially, opening of the valve reduces the pressure drop across the valve and, as a consequence, the pressure under the differential valve is reduced; as a result, the spring forces this valve downward, and its lower ring cuts off the passage for liquid to chamber B, while the upper ring blocks the discharge line from chamber A. At the same time, the working fluid continues to flow out of chamber B through nozzle 3 and into chamber A through valve 2. Servopiston 9 thus moves to the left until the pressure drop across the valve is restored to the prescribed value.

With a high-speed servomotor, the displacement of the servopiston and restoration of the prescribed pressure drop across the throttle valve occurs within a fairly short interval of time. The fuel pressure ahead of the injectors and the rate at which fuel is supplied to the engine will then increase. Thus the rotor speed will also rise.

The increased rotor speed will in turn lead to an increase in pump throughput and subsequent increase in the fuel pressure drop across the throttle valve to a value exceeding that prescribed; this causes the differential valve to open, and the servopiston moves to the right, producing a corresponding reduction in the angle of inclination of the disk so as to restore the prescribed pressure drop.

2. Regulator Operation under Variations in Altitude and Flight Speed with Position of Throttle Valve Unchanged

If the throttle-valve control lever is set to a definite position while the aircraft altitude increases or flight speed decreases, the upsetting of the relationship between the available and required fuel flow rates causes the rotor to speed up, leading to an increased pressure drop across the throttle valve and, consequently, across the differential valve; this causes the latter valve to rise, and the required reduction in fuel flow rate and restoration of prescribed pressure drop across the valve is obtained with the aid of the servopiston. Thus, for all flight speeds and altitudes, as long as the setting of the throttle-valve lever is unchanged, the rate at which fuel is supplied to the engine will remain nearly constant. As a result, while the control of the throttle valve remains in the same position, the rotor speed will increase with increasing altitude or decreasing flight speed. A given rotor speed can be maintained by the pilot only by moving the throttle-valve control so as to reduce the valve flow area.

A positive feature of a pressure-drop regulator is the fact that the rate at which fuel is fed to the engine does not depend on variations in combustion-chamber pressure or variations in the hydraulic resistances of injectors or lines following the throttle valve (produced, for example, by wear, injector clogging, etc.). A change in these factors during service will have no effect on the fuel flow rate. Thus, for example, "erosion" of flow ports in injectors will reduce the hydraulic pressure in the line and increase the rate at which fuel flows to the engine. However, since the fuel pressure ahead of the injectors will drop, the pressure-drop valve will come into play and produce a corresponding drop in the fuel pressure ahead of the throttle valve, thus restoring the prescribed pressure drop. As a consequence, when the throttle valve is in the same position, the previous rate at which fuel is supplied to the engine will always be maintained together with the previous rotor speed.

4. REGULATOR HOLDING FUEL PRESSURE CONSTANT AHEAD OF THROTTLE VALVE

The basic arrangement of such a regulator is shown in Fig. 270.

The basic regulator elements are: the throttle-valve needle 1, servopiston 2, sensing element 3 (diaphragm with rod), which responds to a change in the pressure of the fuel ahead of the throttle valve, a valve 4 with spring 6, lever 5, attached to elastic partition 7 and holding valve 4. The lever receives the force exerted by spring 6, the force due to the fuel pressure in the discharge line from chamber A of the servopiston cylinder, and the force due to the fuel pressure ahead of the throttle valve (behind the pump).

By means of nozzles 9 and 10, as well as valve 4, the fuel pressure in chamber A of the servopiston cylinder may be varied within the required limits. Nozzle 9 is smaller in diameter than nozzle 10. If the pump oscillating element is not turning, there will be no fuel pressure beyond the pump. Valve 4 is forced by its own spring to close nozzle 10, while the servopiston spring holds the servopiston in the far left-hand position, i.e., in the position corresponding to maximum inclination of the fuel-pump disk. When the engine is started, as the speed increases while the throttle-valve position remains unchanged, the pressure ahead of the valve will increase to a value determined by the tension on the spring of valve 4. A further increase in speed will lead first to an increase in the pressure ahead of the valve and opening of nozzle 10 by valve 4; this will cause the servopiston to move to the right, reducing the inclination of the pump disk and restoring the prescribed fuel pressure ahead of the valve.

1. Regulator operation with displacement of throttle valve to new position in order to change engine-rotor speed.

In order to increase the rotor speed, it is sufficient to set the throttle valve to a new position. Then as a result of the control process, the regulator will set the position of the fuel-pump disk so as to increase the rate at which fuel is supplied to the required new value.

In broad terms, the regulator will operate in the following se-

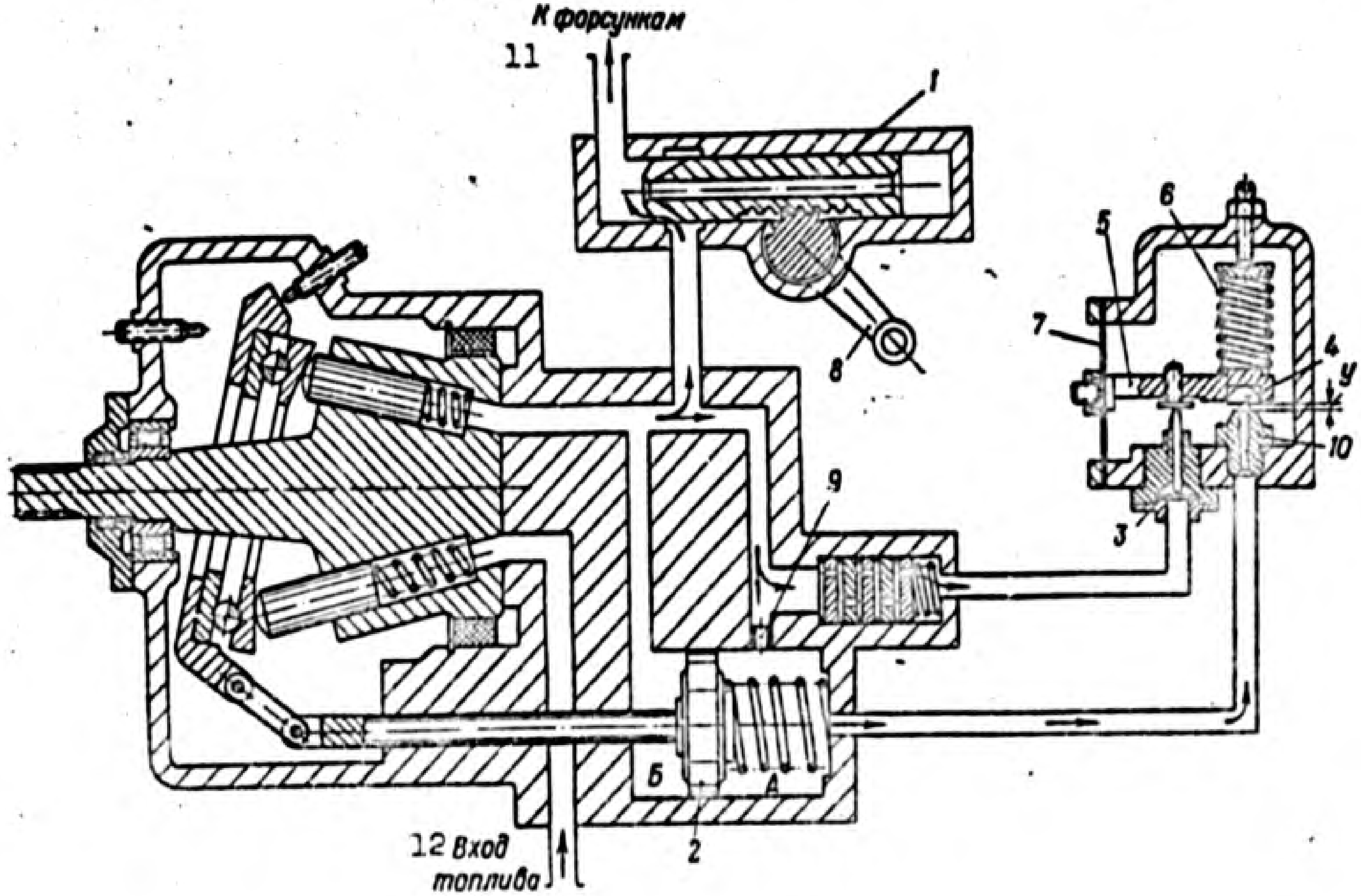

Fig. 270. Basic arrangement of regulator used to hold fuel pressure constant ahead of throttle valve. 1) Throttle-valve needle; 2) servopiston; 3) sensing element; 4) valve; 5) lever; 6) spring; 7) elastic partition; 8) throttle-valve control lever; 9 and 10) nozzles; A and B) chambers; 11) to injectors; 12) fuel intake.

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quence: a reduction in fuel pressure ahead of the throttle valve when it is opened will be detected by sensing element 3, and as a result the force due to the fuel pressure, transmitted to lever 5 of the valve, will be reduced. The force produced by spring 6 will cause the valve to reduce the flow area of nozzle 10 and reduce the rate at which fuel flows from the servomotor chamber A. The increase in pressure in this chamber will move the servopiston to the left, causing an increase in the fuel feed rate; as a result, the fuel pressure ahead of the throttle valve will increase. At the instant the initial pressure value is restored, in approximation, the servopiston will stop since the force due to the fuel pressure will set valve 4 to approximately its initial position. If during the control process the pressure ahead of the valve exceeds the prescribed value, the increasing force due to the fuel pressure will raise valve 4 and increase the rate at which fluid flows out of servomotor chamber A, resetting the servopiston to the position in which the pressure ahead of the valve equals roughly the prescribed value. Restoration of the initial fuel pressure with a large valve flow area provides the necessary fuel flow rate to operate the engine at increased rotor speed.

Where the throttle valve is closed down in order to reduce rotor speed, the regulator operates in the opposite sequence.

2. Operation of regulator with variation in altitude and flight speed, throttle-valve position unchanged.

When the flight altitude increases with the position of the throttle valve unchanged, the rotor speed and, consequently, the fuel pressure ahead of the throttle valve will increase at first. As a result, valve 4 is lifted, the servopiston reduces the inclination of the disk, and the fuel pressure is restored to roughly its initial value. The regulator thus provides a constant fuel flow rate for all flight speeds and various altitudes, while the required fuel feed rate varies. Thus with increasing altitude or decreasing flight speed, the rotor speed will increase.

In the absence of special automatic devices for holding the rotor speed constant, this may be done manually by closing down the throttle valve.

Regulators holding the fuel pressure ahead of the throttle valve constant have the following basic drawbacks:

1) large pressure drop across the valve when the engine operates at low rotor speeds, causing excessive loading of fuel-pump elements;

2) the rate at which fuel is supplied to the engine and, consequently, the engine-rotor speed depend on the back pressure in the combustion chambers and on injector characteristics, which may vary during operation, for example, owing to wear on injectors, injector clogging, etc. Actually, "erosion" of tangential injector ducts will lead to a drop in the fuel pressure ahead of the injectors and in all other fuel-line cross sections. A drop in the fuel pressure ahead of the throttle valve will bring the regulator into operation; it reestablishes the fuel pressure ahead of the valve by shifting the pump to an increased fuel feed rate for the same position of the throttle lever. Finally, large pressure drops across the throttle valve require small flow areas in order to provide the required fuel flow rates and, consequently, very accurate machining of the throttle-valve needle shape.

These drawbacks, as we shall show below, are not present in regulators that hold the fuel pressure drop across the throttle valve constant.

5. BLOCKING OF AUTOMATIC SPEED REGULATOR WHERE CONSTANT FUEL-FEED RATE REGULATOR IS OPERATING

As we mentioned above, in Chapter 19, the section "Fundamental

Properties of Centrifugal Sensing Elements," a centrifugal regulator automatically controls the rotor speed over a range extending from the speed at which automatic operation of the regulator begins, n_{nar}, to the maximum speed n_{max}, and over the range from n_{mg} to n_{nar}, the speed is controlled manually with the aid of a regulator that holds the fuel pressure drop across the throttle valve constant.

In existing turbojet engines, n_{nar} amounts to 0.7 to 0.8 of the value of n_{max}.

The regulator holding the fuel pressure drop constant across the throttle valve and the automatic speed governor are prevented from operating simultaneously by blocking one of the regulators when the other is operating.

It is usual in present-day regulators to block the automatic speed regulator by preliminary compression of the sensing-element spring [11].

The blocking mechanism (Fig. 271) consists of an inside cylinder 1 and adjusting screw 3 (the screw adjusts the point at which the speed regulator begins automatic operation). Cylinder 1 is located in an outside cylinder 2, which has a rack connected with a gear 5 on the lever 10 of throttle-valve needle 6. The spring 4 of the regulator sensing-element adjustment mechanism rests in the inside cylinder 1. Owing to preliminary tension on the spring, for all speeds from idle running to the beginning of automatic operation of the regulator, the control slide valve of the centrifugal regulator is in a position somewhat away from the neutral position (Fig. 271a), since the centrifugal force due to the weights is inadequate at such speeds to overcome the force of the spring and to set the slide valve to the neutral position.

As a result, the working fluid is supplied to chamber A under piston 11 and is taken out of chamber B of the inclined-disk servopiston. The force due to fuel pressure and to the spring holds piston 11

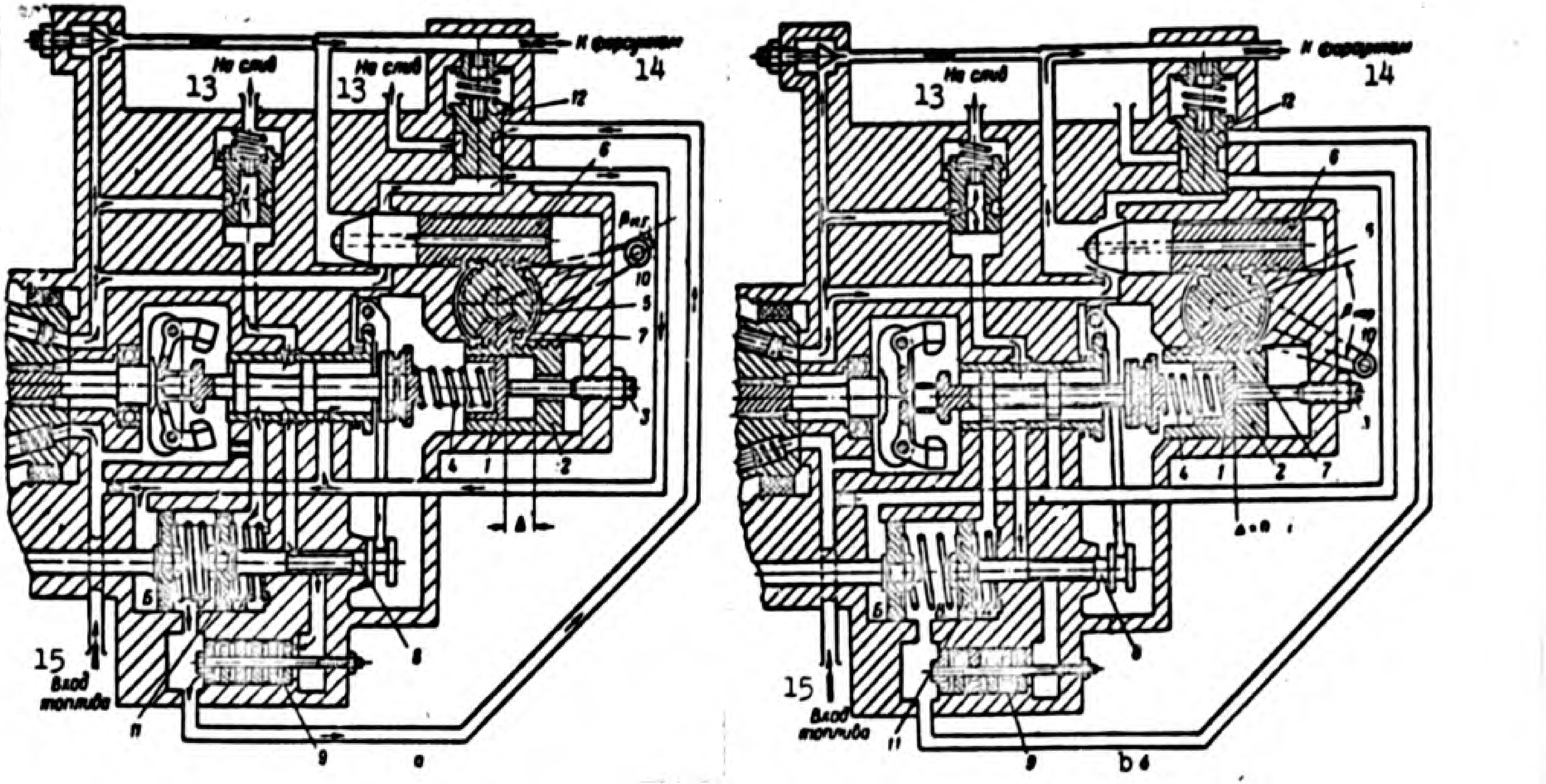

Fig. 271. Basic arrangement of blocking mechanism used with speed regulator where constant-pressure-drop valve is employed. a) Position of regulator elements before automatic operation begins; b) position of regulator elements at beginning of automatic operation; 1) Inside cylinder; 2) outside cylinder; 3) adjusting screw; 4) spring; 5 and 7) gear teeth; 6) throttle-valve needle; 8) slide valve; 9) throttle unit; 10) throttle-valve lever; 11) piston; 12) pressure-drop valve; A, B, and C) chambers; 13) to discharge line; 14) to injectors; 15) fuel intake.

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against the stop in the extreme left-hand position, so that the working fluid enters through the annular groove in slide valve 8 and throttle element 9 into the interpiston chamber C. As a result, the servopiston tends to force the disk into the position of maximum angle of inclination; this is opposed, however, by the constant-pressure-drop valve, which releases working fluid from the interpiston chamber and opens the passage for taking working fluid into chamber B under the servopiston, thus holding the pump disk in the position that provides the prescribed pressure drop across the valve.

When the throttle-valve lever is moved in the manual-control zone, the pressure drop across the valve in steady regimes remains constant owing to the operation of pressure-drop valve 12. Here the force due to preliminary tension of the centrifugal-regulator spring does not change owing to the presence of a gap Δ between the inside and outside cylinders. The gap between the cylinders is completely taken up when the throttle-valve lever is in the position corresponding to the beginning of automatic regulator operation (Fig. 271b). In this position, an amount of fuel reaches the engine through the throttle valve such that the rotor speed reaches the value n_{nar} and the centrifugal force produced by the weights sets the control slide valve to the neutral position, having overcome the spring force; piston 11 sets the feedback elements to the neutral position as well. Further opening of the throttle valve leads to readjustment of the automatic regulator to a higher rotor speed (i.e., produces increased tension in the regulator-adjusting spring).

Thus, owing to the preliminary tension of the sensing-element spring, the constant-pressure-drop valve controls the position of the inclined-disk servopiston over the speed range from n_{mg} to n_{nar}, and the centrifugal speed regulator controls the position of the fuel-pump servopiston over the speed range from n_{nar} to n_{max}.

Here the valve maintaining the fuel pressure drop constant goes out of operation and has no effect on the servopiston position. The valve is disconnected owing to the fact that when the throttle-valve lever moves through the position corresponding to an angle of rotation β_{nar}, there is a sharp increase in the valve flow area owing to the notches provided and the increased taper at the end of the shaped section of the throttle-valve needle. As a result, the pressure drop across the valve is lower than that for which the valve spring is set, and its spring holds the valve against the stop over the entire range of centrifugal-regulator automatic operation. Despite the decrease in pressure drop, the increased flow-section areas in the valve provides the increase in fuel flow rate through the throttle valve needed for operation at higher rotor speed in the zone of automatic centrifugal-regulator operation.

In adjusting the speed at which automatic regulator operation commences, it is necessary to match the position of the throttle-valve needle to the position of the inside and outside cylinders: when a sharp increase in valve flow area begins, the outside cylinder should affect the position of the inside cylinder. Various designs are used to make this possible in regulators presently used.

Thus, for example, an element for adjusting the speed at which automatic operation begins may be designed as follows. Shaft 1 (Fig. 272), carrying gear 5, is connected by a rack to the external cylinder 2; the shaft carries gear 3, which engages the rack on throttle-valve needle 4. The shafts are connected together by coupling 6. One end of the coupling is connected by splines to the inside shaft 1; the other end is connected through worm gear 7 to the outside shaft 9. Through this connection, rotation of worm gear 7 turns the outside shaft with

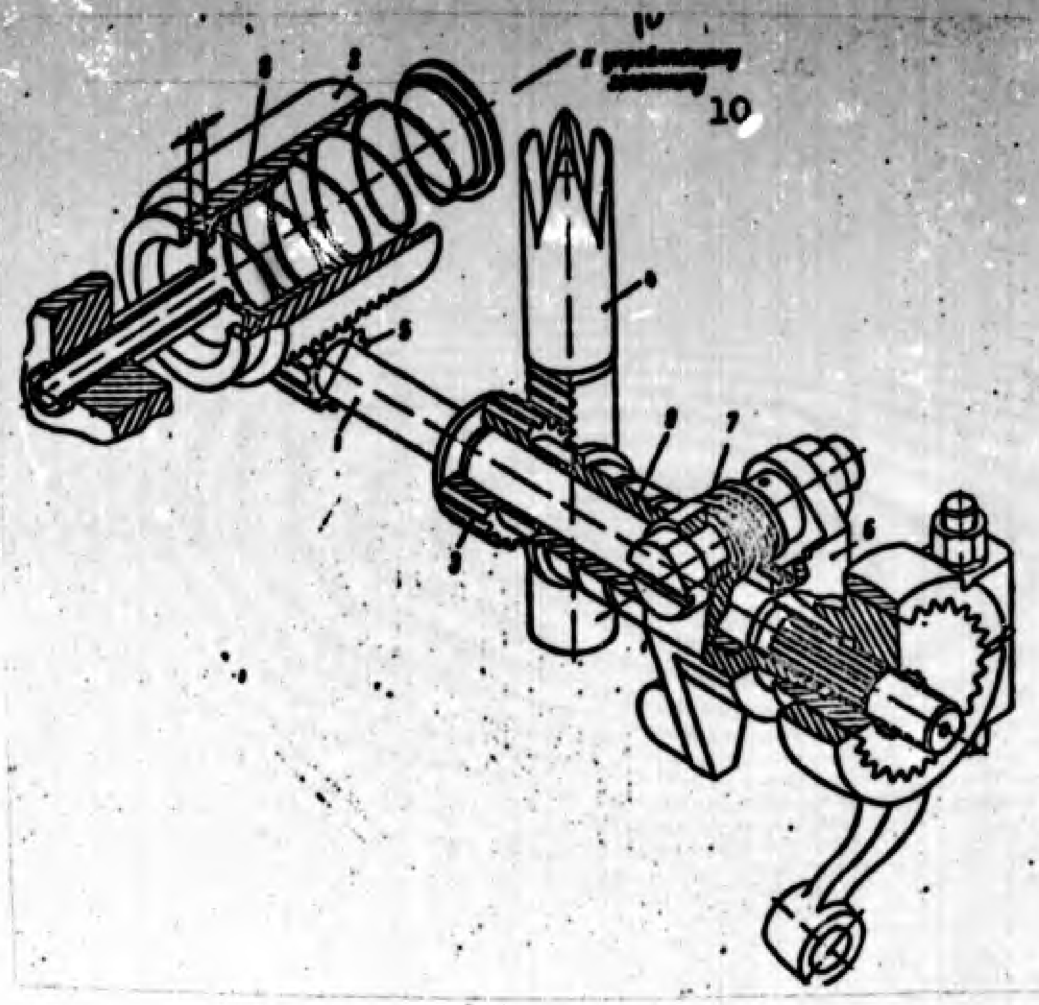

Fig. 272. Unit for adjusting speed-regulator blocking mechanism. 1) Inside shaft; 2) outside cylinder; 3 and 5) gear teeth; 4) throttle-valve needle; 6) coupling; 7) worm gear; 8) inside cylinder; 9) outside shaft; 10) to control slide valve.

respect to the inside shaft, so that the required relationship may be obtained between the positions of the valve needle and the outside cylinder.

Chapter 22

INDIRECT SPEED-CONTROL SYSTEMS

1. FUNDAMENTAL TYPES OF INDIRECT SPEED-CONTROL SYSTEMS AND THEIR PROPERTIES

In indirect speed-control systems, the regulator sensing elements measure the stagnation pressure and stagnation temperature at the engine intake; these quantities determine the required fuel feed rate. In accordance with the variation in these parameters, the regulators correct the fuel supply rate so as to prevent the speed from departing from the prescribed value. Frequently, in order to simplify the control system, only the variation in total pressure is measured; this, however, leads to inaccurate regulation.

The fuel feed rate is corrected by action of the sensing element through a servomotor on the position of the throttle valve, with a constant pressure drop across the valve, or action on the fuel pressure ahead of the valve, which has constant flow area.

Figure 273 shows the basic arrangement of a system using the first method. The system consists of a gear fuel pump 1, throttle valve 2, which is set by servomotor 3, a valve 4 to maintain constant pressure drop across the valve, a bypass valve 5, and detectors 6 and 7 for the total pressure and air temperature at the engine intake.

Signals are transmitted from the detectors with the aid of lever 8 to the control slide valve 9, which determines the position of the servopiston 10 and the attached throttle valve 2. Thus, for example, when the air temperature increases, detector 7 (a hydraulic-type tem-

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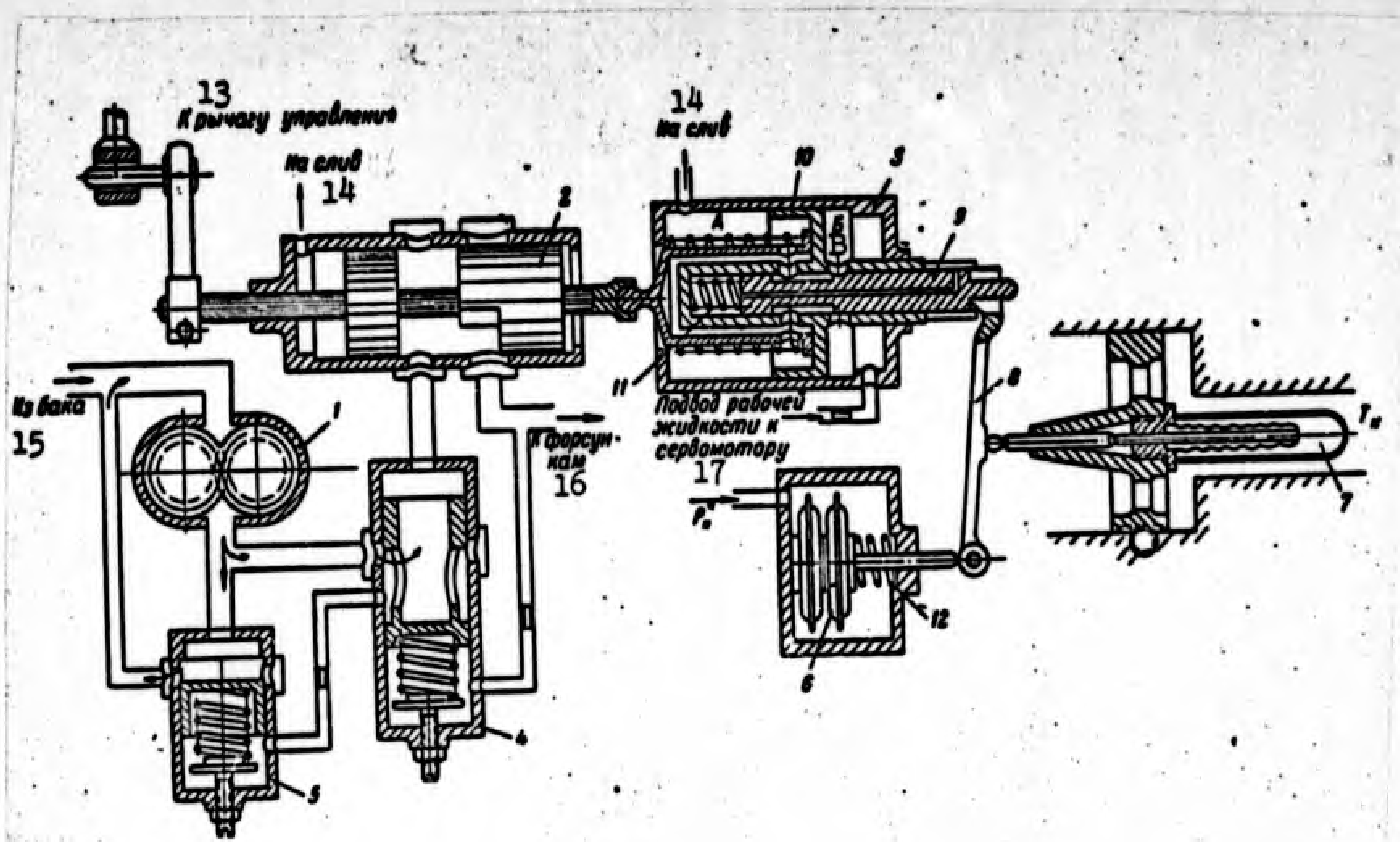

Fig. 273. Basic arrangement of indirect control system correcting fuel feed rate by changing flow area in throttle valve. 1) Fuel pump; 2) throttle valve; 3) servomotor; 4) constant-pressure-drop valve; 5) bypass valve; 6 and 7) total-pressure and temperature detectors; 8) lever; 9) control slide valve; 10) servo-piston; 11 and 12) springs; A and B) chambers; 13) to control lever; 14) to discharge line; 15) from tank; 16) to injector; 17) supply of working fluid to servomotor.

perature detector) moves its rod to the left, lever 8 turns about a pivot on a rod connected to the aneroid capsules of detector 6 (a total-pressure detector of the diaphragm type), and also moves control slide valve 9 to the left, causing the servopiston to move to the left and reducing the flow area of the throttle valve. Thus with a constant pressure drop across the valve, the rate at which fuel is fed to the engine will decrease.

When the temperature decreases, the system operates in the reverse order.

If the air pressure increases, the aneroid capsules of detector 6 are compressed, the aneroid-capsule rod moves to the left owing to the force produced by spring 12; at the same time, spring 11 moves the slide valve to the right, causing lever 8 to turn about the rod of the thermometer of detector 7 as a pivot. The servopiston then moves after a slide valve, thus increasing the flow area in the valve and, consequently, the fuel flow rate as well.

When the pressure decreases, the system operates in reverse order.

Figure 274 shows the basic arrangement of a system in which the fuel feed rate is corrected by changing the fuel pressure ahead of the throttle valve.

The sensing element, which responds to variations in flight speed and altitude is a set of aneroid capsules 7 in a barostat located in a special chamber to which the total pressure at the engine intake is applied.

As the air pressure changes, the effective force at the aneroid capsules changes; this force is transmitted to a lever and, consequently, it affects the position of control valve 4.

There are two possible cases of system operation.

1. The air pressure increases (flight altitude decreases or air-

craft flight speed increases). In this case, the aneroid elements are compressed and, consequently, apply less force to the lever, which turns and constricts the nozzle with the aid of valve 4. As a result, the servopiston moves to the left, thus increasing the rate at which fuel is fed to the engine; as a result, the fuel pressure ahead of the valve increases and, consequently, the force due to the fuel pressure rises; this force is transmitted to lever 5 by the diaphragm 10 and rod 3. This force sets the valve to roughly the initial position (the valve does not return exactly to the initial position owing to the servomotor static error).

2. The air pressure is reduced (flight altitude increases or aircraft flight speed decreases). In this case, the aneroid elements expand and the force with which they press against the lever increases. Then the equilibrium of the lever is upset and it turns around the pivot, causing the fuel to flow through the nozzle in larger quantities. The servopiston moves to the right, causing pump throughput to decrease; then the fuel pressure behind the pump and the force due to the fuel pressure, which acts on lever 5, will also decrease. The decrease in this force will cause the valve to return nearly to its initial position (it will not return to it exactly), causing the servopiston to cease its movement.

In both of the cases considered, the engine-rotor speed is not maintained exactly owing to the fact that the actual barostat characteristic curve differs from that needed to ensure constant rotor speed for different aircraft flight speeds.

The barostat characteristic is the name given to the relationship between the fuel pressure behind the pump (ahead of the throttle valve) p_{nas}, and the air pressure at the intake side, p^*_N (Fig. 275).

During operation, it may become necessary to match the required

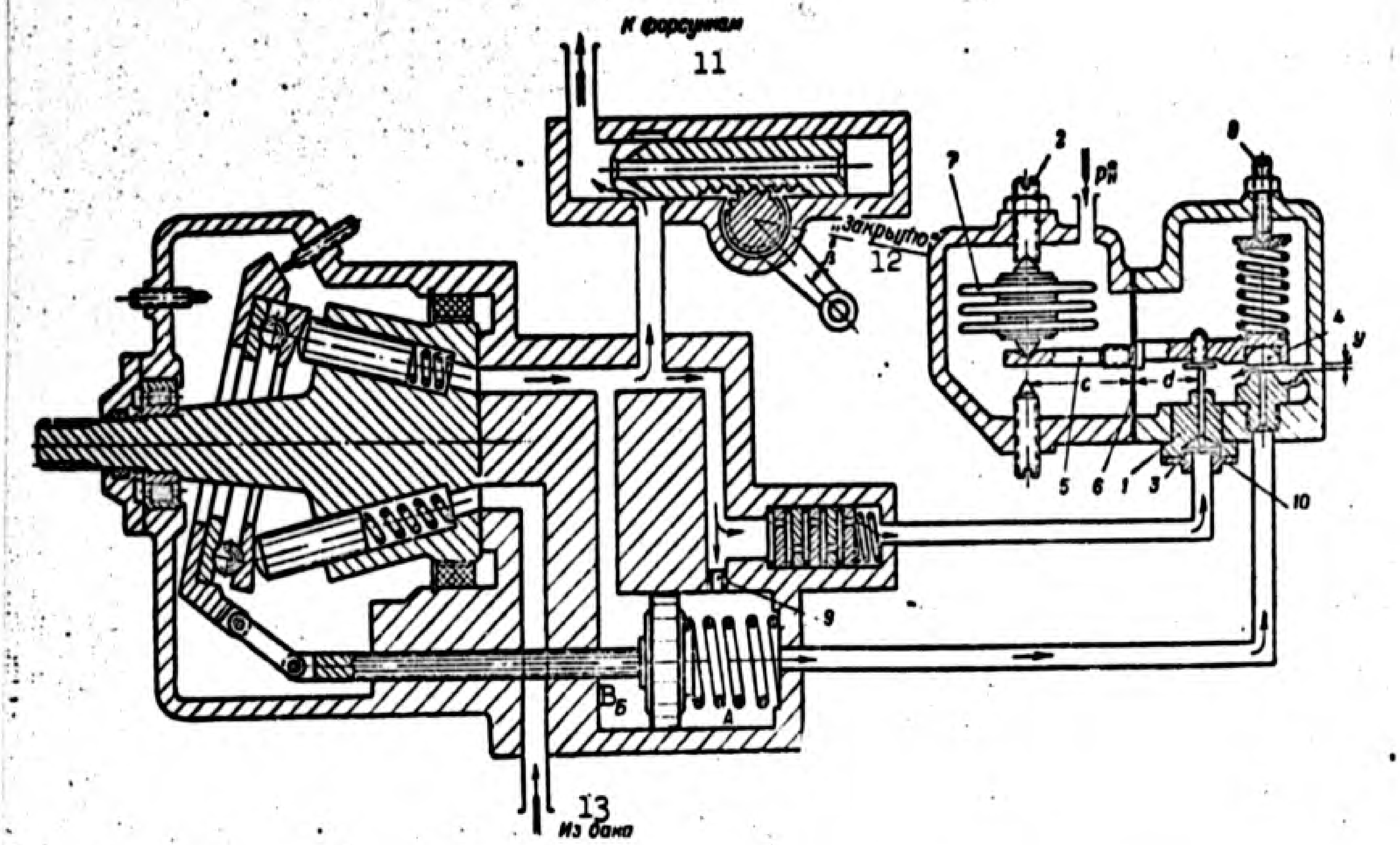

Fig. 274. Indirect control system with correction of fuel feed rate by variation in fuel pressure ahead of throttle valve. 1) Eccentric bushing; 2 and 8) adjusting screws; 3) rod; 4) control rod; 5) lever; 6) elastic partition; 7) set of barostat aneroid capsules; 9) nozzle; 10) diaphragm; A and B) chambers; 11) to injector; 12) "closed"; 13) from tank.

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and actual barostat characteristics so as to bring the law governing the change in speed with altitude close to the desired law. This is done by changing the position of the actual barostat characteristic with respect to the required characteristic. The slope of the actual characteristic may be changed by varying the ratio of arms c and d of lever 5 (see Fig. 274); the design provides for this to be done by moving the rod of diaphragm 3 in the eccentric bushing 1. An increase in the ratio c/d leads to an increase in the slope of the actual characteristic, an increase in the fuel pressure ahead of the throttle valve, and in the rotor speed, both when the engine is operating on the ground and under high-altitude conditions. The actual barostat characteristic may be lowered by increasing the force of the aneroid element and decreasing the spring-tension force. The spring-tension force and the preliminary compression of the aneroid element are changed with the aid of special adjusting screws 2 and 8. The actual and required barostat characteristics intersect at only two points (see Fig. 275). This means that there are only two values of p^*_N and, consequently, two altitudes H_1 and H_2 (Fig. 276) for which the speed will equal the prescribed value.

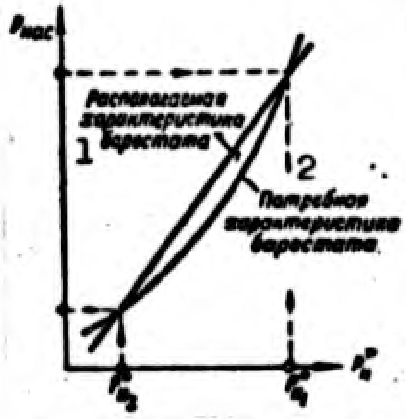

Fig. 275. Barostat characteristics. 1) Actual barostat characteristic; 2) required barostat characteristic.

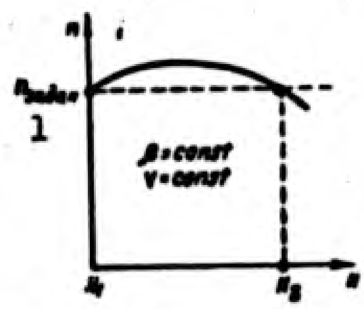

Fig. 276. Variation in speed with flight altitude for linear actual barostat characteristic. 1) n_{zadan}.

At intermediate altitudes, with the position of the throttle valve unchanged, the rotor speed will exceed the prescribed value. In order to hold to the desired speed, the pilot should use the throttle-valve control lever to introduce an appropriate correction.

A rotor maximum-speed governor is used to keep the speed from exceeding the maximum permissible value.

2. ROTOR MAXIMUM-SPEED GOVERNORS IN INDIRECT CONTROL SYSTEMS

A rotor maximum-speed governor is a regulator with a fixed adjustment of the sensing element. Governor sensing elements may be of the centrifugal, pendulum, or hydrocentrifugal types, while they use standard valves or slide valves as control elements.

Governor using a centrifugal-pendulum sensing element and slide-valve control element. This governor differs from the centrifugal speed regulator discussed above only in the fixed adjustment of spring 5 (see Fig. 255), whose tension is so adjusted as to limit the maximum permissible engine-rotor speed. If the rotor exceeds the prescribed maximum speed, the slide valve will move to the left, the rate at which fuel is transferred from the injector line to the pump input will increase, and the rotor speed will drop to the prescribed limiting value.

Governor using hydrocentrifugal sensing element and control valve (Fig. 277). This type of governor uses a hemisphere valve 1, located on lever 3 with axis of rotation 8, and exerting control by opening or closing nozzle 2. Nozzle 2 is connected through a duct to chamber A of the servopiston cylinder. At all speeds below the speed at which the governor comes into play, valve 1 closes nozzle 2 owing to the presence of spring 4 and of the gap between the valve lever and the stop for elastic diaphragm 5, which is pulled upward by spring 6. Fuel is sent into the chamber above the diaphragm, which acts as the sensing element, through port 11; the fuel is under the pressure set up in the chamber of the oscillating element owing to the action of centrifugal forces when the fuel passes through the inclined radial holes 9 into the oscillating element of the fuel pump. The force exerted by fuel pressure

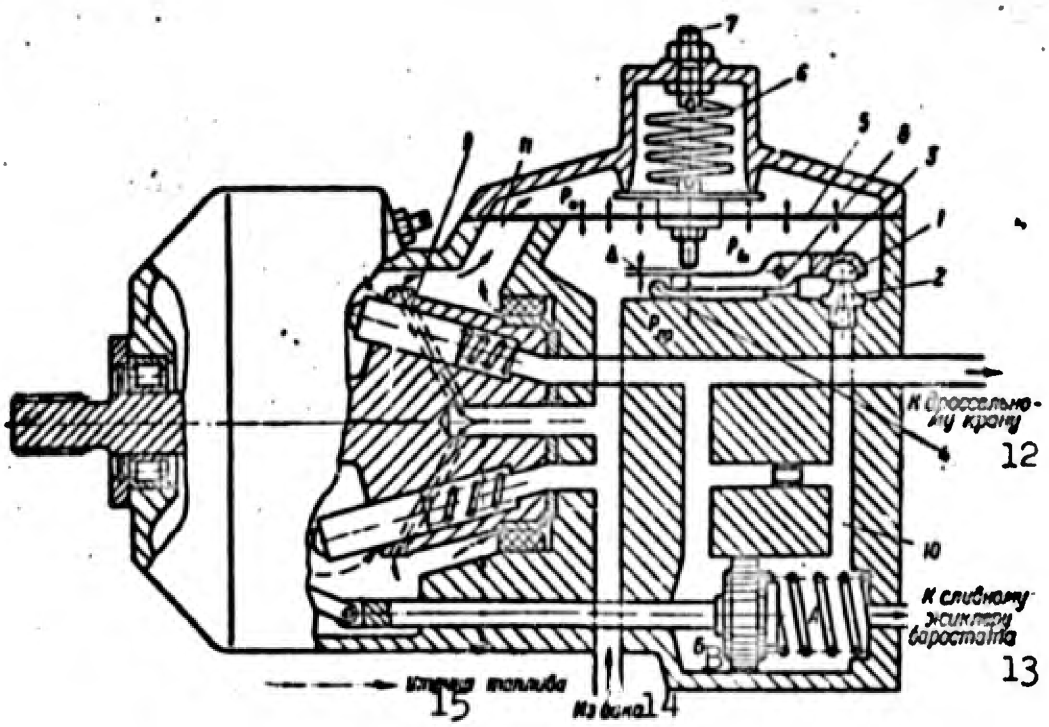

Fig. 277. Basic arrangement of rotor maximum-speed governor using hydrocentrifugal sensing element. 1) Hemisphere valve; 2) nozzle; 3) lever; 4 and 6) springs; 5) elastic diaphragm; 7) adjusting screw; 8) axis of rotation; 9) radial holes; 10) duct; 11) port; A and B) chambers; 12) to throttle valve; 13) to barostat discharge nozzle; 14) from tank; 15) fuel leakage.

is a function of the square of the rotor speed. The fuel pressure at the pump inlet acts from below on the diaphragm.

When the maximum rotor speed is reached, the fuel pressure in the chamber above the diaphragm increases to the point at which the diaphragm bends and its stop presses against the lever, turning it and opening nozzle 2. When this happens, the resultant force exerted by springs 4 and 6 is overcome. Fuel flows out from chamber A of the servomotor through the open nozzle 2; as a result, the servopiston moves to the right, reducing the fuel flow rate and thus counteracting the increase in speed over the maximum value. The maximum speed is determined by the compression of spring 6, which is adjusted by screw 7.

Operation of the governor shown in Fig. 277 is affected by the force due to fuel pressure at the outlet from nozzle 2, applied to the lever of the hemisphere valve, by the leakage of fuel from the high-pressure line into the rotating-unit chamber, which is connected with

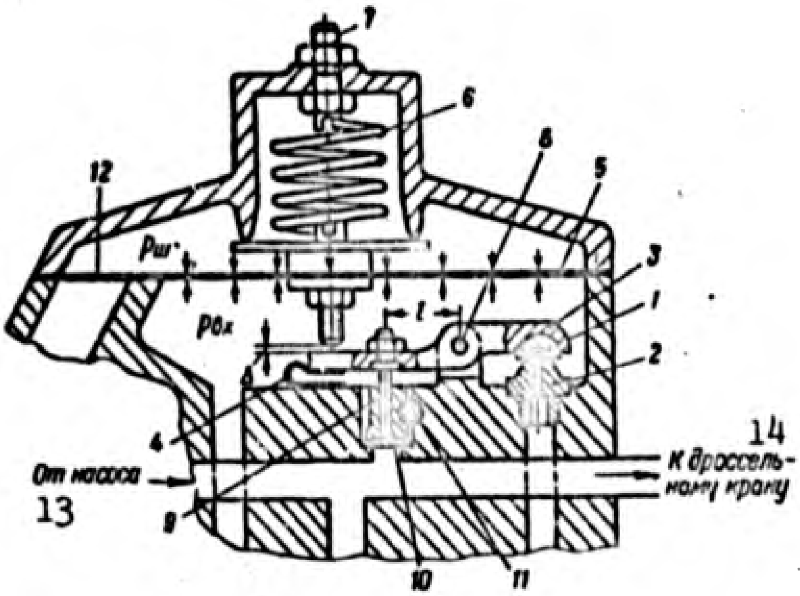

Fig. 278. Basic arrangement of maximum-speed governor using hydrocentrifugal sensing element and compensating device. 1) Hemisphere valve; 2) nozzle; 3) lever; 4 and 6) springs; 5) elastic diaphragm; 7 and 11) adjusting screws; 8) axis of rotation; 9) rod; 10) eccentric sleeve; 12) port; 13) from pump; 14) to throttle valve.

the chamber above the governor diaphragm, and by the specific gravity of the fuel (fuel leakage is shown by the dash-dot arrows).

Let us examine the effect of each factor on the maximum speed separately.

When the governor is in operation, the magnitude of the force due to fuel pressure at the outlet will vary owing to variation in fuel pressure in the line beyond the pump, and will depend on the amount by which the governor valve is lifted.

A decrease in the fuel pressure beyond the pump, for example, with an increase in flight altitude, will lead to a corresponding reduction in the pressure in chamber A of the servomotor cylinder and a reduction in the force due to the fuel pressure at the outlet. As a result, for the governor valve to go into governor operation, the pressure of the fuel against the governor diaphragm must be high; this pressure depends on the rotor speed. As a consequence, as the flight altitude increases, the maximum engine-rotor speed will increase.

In order to reduce this undesirable effect, lever 3 of the governor is loaded by a second fuel-pressure force applied on the other side of the lever pivot 8 (Fig. 278). Rod 9 of this compensating device is located in an eccentric bushing 10, which is used with adjusting screw 11 to change arm l along which the compensating force is applied and thus to adjust the maximum rotor speed when the engine operates under high-altitude conditions. As this arm is shortened, the compensation effect decreases, leading to an increase in the maximum speed as the flight altitude increases.

An increase in n_{max} with increasing altitude will also be aided by a decrease in the fuel pressure above the governor diaphragm; this occurs as a result of decreased leakage of fuel from the high-pressure line, where the fuel pressure drops as the flight altitude increases.

The direction of fuel leakage in the pump oscillating-unit gap is shown in Fig. 277 by the dash-dot arrows. The fuel pressure in the inclined-flight chamber, which acts on the diaphragm, is made up of a pressure that depends on the rotor speed and a pressure caused by flow of fuel through the gaps from the high-pressure line to the supply line. As the flight altitude increases, the second pressure component decreases, and it is thus possible to bring the governor into play only when the first pressure component is of higher value, i.e., when the rotor speed is higher, since the spring force which must be overcome remains constant.

An increase in fuel specific gravity (for example, with a decrease in fuel temperature) leads to a reduction in the rotor maximum speed when the governor adjustment remains unchanged, since the fuel-pressure force that must be exerted on the diaphragm in order to overcome the spring force and bring the governor into play will be reached at a speed that is lower the greater the density of the fluid.

<u>Governor with hydrocentrifugal sensing element and control slide</u>

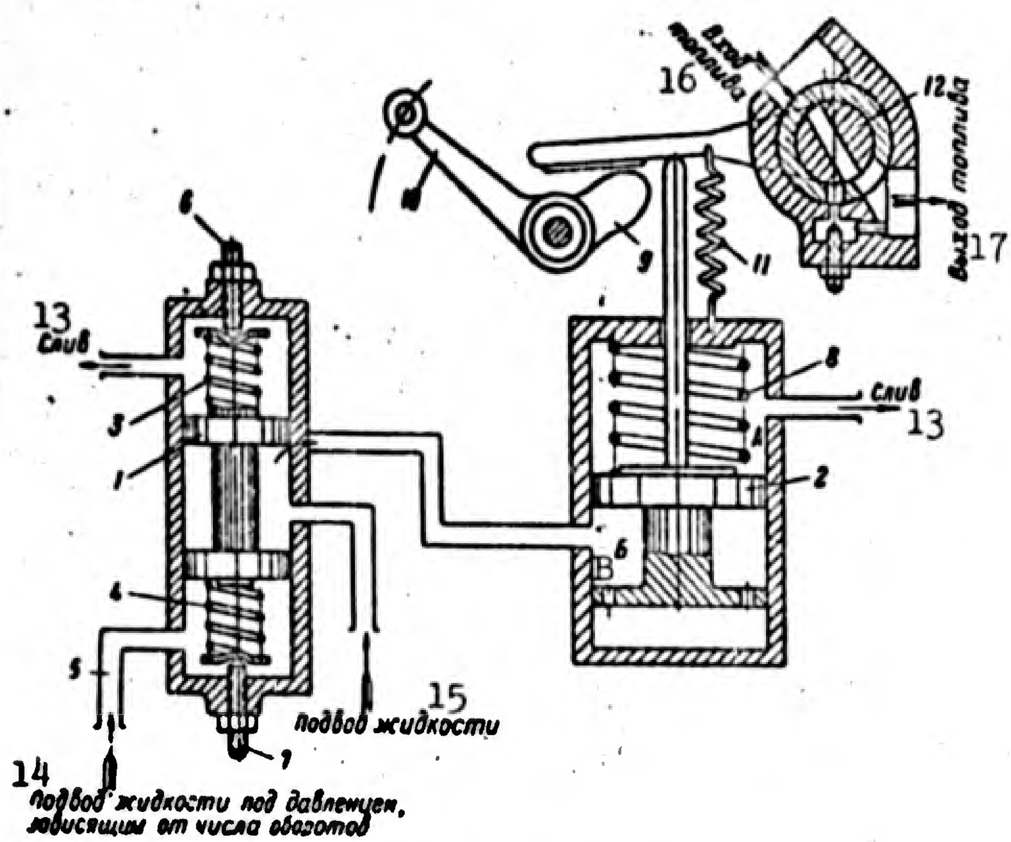

Fig. 279. Maximum-speed governor using hydrocentrifugal sensing element and control slide valve. 1) Slide valve; 2) servopiston; 3, 4, 8, and 11) springs; 5) duct; 6 and 7) adjusting screws; 9) cam; 10) lever; 12) throttle valve; A and B) chambers; 13) discharge; 14) supply of liquid under pressure depending on speed; 15) supply of fluid; 16) fuel intake; 17) fuel outlet.

valve (Fig. 279).

The slide valve 1, which controls the supply of working fluid to servopiston 2, is loaded by the forces due to springs 3 and 4 and the force produced by the pressure of the fuel supplied through duct 5. This force depends on the rotor speed. The initial spring tension is set by the adjusting screws 6 and 7; screw 6 adjusts the maximum speed for which the governor is set.

At all speeds below the maximum speed the slide valve is in a position somewhat below the neutral position, so that it connects chamber B of the servopiston cylinder with the discharge line. The servopiston is held against a stop at the very bottom position by spring 8. Cam 9 controls the position of throttle valve 12; the cam is attached to the shaft of engine control lever 10. The valve lever is held in contact with the cam by the tension of spring 11.

If the maximum speed is exceeded, the slide valve is moved upward away from the neutral position by the increasing force produced by the pressure of the working fluid; the slide valve opens the passage through which working fluid enters chamber B, closes down the throttle valve, and reduces the rate at which fuel is supplied to the engine.

Chapter 23

AUTOMATIZATION OF FUEL SUPPLY DURING ACCELERATION OF TURBOJET-ENGINE ROTORS

Turbojet engines [TRD] are required to have good pickup, i.e., when the control lever is moved the engine must go rapidly from one operating regime to another, especially from the idle-running regime to maximum speed, which is of great importance in a rapid aircraft takeoff run.

Engine acceleration is evaluated in terms of the minimum possible time required for the engine to go from minimum to maximum thrust.

The acceleration time depends on the method used to throttle down the thrust and on the rotor moment of inertia.

One of the most widely employed throttling methods involves variation of the rate at which fuel is fed to the engine with constant exhaust-nozzle throat area. In this case, a variation in fuel feed rate causes a change in the amount by which the turbine torque exceeds the compressor torque, and a change in rotor speed.

Another method of throttling thrust consists in changing the exhaust-nozzle critical area while holding the rotor speed constant. In order to increase engine thrust, it is necessary to reduce the area of the throat section, which increases the turbine back pressure and reduces the speed. The speed regulator maintains the speed constant by increasing the rate at which fuel is fed to the engine, causing the engine thrust to increase. This method does not permit heavy thrust throttling, and is thus used only in combination with the first method.

For TRD, acceleration time under ground conditions is 8-15 sec. With increasing altitude and decreasing flight speed, the acceleration time changes owing to a reduction in the excess torque at the engine rotor and an increase in rotor speed in the idle-running regime. A reduction in the excess torque causes an increase in acceleration time, while an increase in idling speed reduces the acceleration range and, consequently, decreases the acceleration time. The resultant action of all factors may increase the acceleration time or keep it constant in comparison with the acceleration time under ground conditions.

Acceleration of an engine rotor to a new speed can only occur where the available fuel feed rate is greater than the required feed rate for the given speed. When the system is readjusted to the higher speed, an extreme increase in fuel feed rate may occur, exceeding the value permissible for the acceleration process. The resulting excessive change in the gas temperature ahead of the turbine may present a danger to the engine, since it can lead to overheating of the turbine or surge operation of the engine. As the speed rises and the rate at which air is supplied to the combustion chamber increases, the temperature of the gas ahead of the turbine will decrease smoothly to a new equilibrium value. A sharp increase in the rate at which fuel is fed to the engine during rotor acceleration will lead to an over-rich mixture, impairing combustion stability and shutting down the engine.

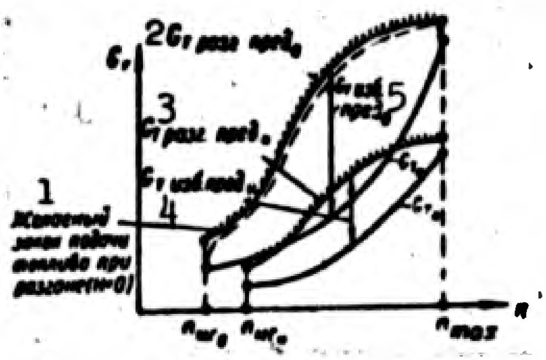

Fig. 280. Fuel feed rate in steady regimes and maximum feed rate during engine-rotor acceleration as functions of rotor speed under ground and high-altitude conditions. 1) Desired variation in fuel feed rate during acceleration (H = 0); 2) $G_{t\ razg.pred_0}$; 3) $G_{t\ razg.\ pred_n}$; 4) $G_{t\ izb.\ pred_n}$; 5) $G_{t\ izb.\ pred_0}$.

In order to keep these things from

happening, fuel feed rates are limited during engine-rotor acceleration to maximum permissible values. The boundary of the maximum permissible fuel feed rates during acceleration at various rotor speeds has been established experimentally and is shown in Fig. 280 for ground conditions (subscript zero) and high-altitude conditions (subscript N) in acceleration. With increasing flight altitude, the excess fuel $G_{t\ izb.pred} = G_{t\ razg.pred} - G_t$, supplied to the engine, will become less in accordance with the decreased air feed rate.

The acceleration time will be least where the fuel feed rate during acceleration equals or is somewhat less than the maximum permissible value (see the dashed line in Fig. 280). During acceleration, the fuel feed rate may be adjusted manually to follow a desired (optimum) law of variation by moving the control lever at a particular speed, which requires a pilot with considerable skill in handling the controls, and which distracts the pilot's attention from other flight functions. Turbojet engines are therefore provided with special automatic acceleration devices, which automatically limit the fuel feed rate during acceleration when the regulator is rapidly reset.

The operation of an automatic acceleration device may be based on various principles; the following types of devices are most commonly used at present, however.

1. A device limiting the fuel pressure ahead of the injectors, which responds to a difference in the air pressures behind and ahead of the compressor, and which is called a pneumatic automatic acceleration control.

2. A time-based device that limits the rate at which the speed regulator can be reset, called an automatic hydraulic acceleration retarder.

3. A rotor-speed based device that limits the rate at which the

fuel flow-rate regulator can be reset, called an automatic tachometer acceleration control.

4. A time-based device limiting the increase in fuel pressure ahead of the injectors.

1. AUTOMATIC PNEUMATIC ACCELERATION CONTROLS

Automatic pneumatic acceleration controls used at the present time restrict the rate at which fuel is fed to the engine during acceleration in accordance with the variation in the air flow rate, since the relationship between the fuel feed rate and air flow rate determines the gas temperature ahead of the turbine and the nature of the mixture.

The rate at which fuel is fed to an engine is determined, all other conditions being equal, by the fuel pressure ahead of the injectors. The air flow rate depends on the air pressure drop at the compressor. Therefore, in order to restrict the rate at which fuel is fed to an engine during acceleration, we need only restrict the air pressure ahead of the injectors in accordance with the variation in air-pressure drop at the compressor.

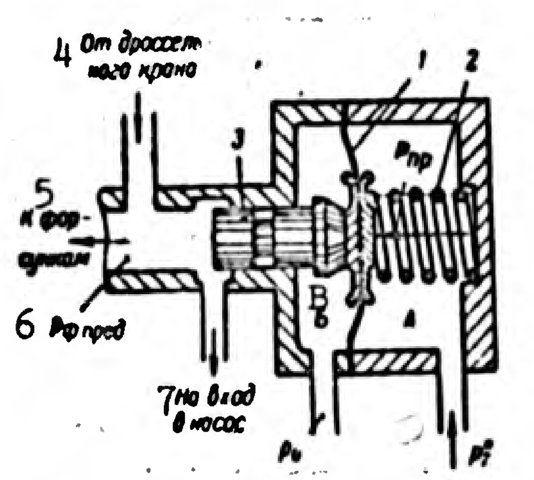

Fig. 281. Direct-acting automatic acceleration control. 1) Elastic diaphragm; 2) spring; 3) slide valve; A and B) chambers; 4) from throttle valve; 5) to injectors; 6) $p_{f\ pred}$; 7) to pump intake.

Figure 281 shows a very simple direct-acting automatic acceleration control used with a gear pump and connected into the fuel-supply system. The automatic device has an elastic diaphragm 1, spring 2, and slide valve 3. The air pressure due to the compressor is applied to chamber A, while chamber B is connected to the atmosphere. In steady operation, the force due to the spring and the force due to

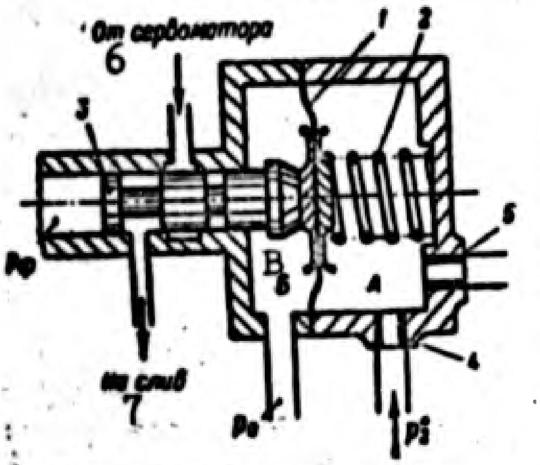

Fig. 282. Indirect automatic acceleration control, regulating air pressure drop across diaphragm. 1) Elastic diaphragm; 2) spring; 3) slide valve; 4 and 5) restricting nozzles; A and B) chambers; 6) from servomotor; 7) to discharge line.

the air-pressure drop will exceed the force produced by the fuel pressure ahead of the injectors. The slide valve will therefore shut off the fuel-discharge duct leading from the chamber ahead of the injectors to the pump intake.

If during engine-rotor acceleration, the pressure ahead of the injectors rises more rapidly than is permissible on the basis of the air pressure drop at the compressor, the slide valve will move and open the passageway through which part of the fuel can be discharged to the fuel-pump intake.

In an automatic indirect acceleration controller connected into a system with a centrifugal regulator (Fig. 282), the slide valve 3 controls the discharge of fuel from the inclined-disk servomotor, thus affecting the rate at which pump throughput increases during engine acceleration.

These very simple automatic acceleration controls do not provide an optimum fuel feed rate during acceleration of an engine rotor. In order to bring the fuel feed rate close to the desired pattern, it is necessary to change the spring tension and air pressure in chamber A; to do this, it is sufficient to install a nozzle 4 at the inlet to chamber A and a nozzle 5 at its outlet (see Fig. 282). Then the air pressure in chamber A will be less than the pressure after the compressor and greater than atmospheric pressure. As the spring force or area of intake nozzle 4 increase, as well as when the area of the outlet (scavenging) nozzle 5 decreases, the available fuel flow rate will increase during acceleration. The actual characteristic of such an automatic acceleration control differs from the desired characteristic, however. For a better match between the actual characteristic and the desired characteristic, the intake-nozzle area is changed in accordance

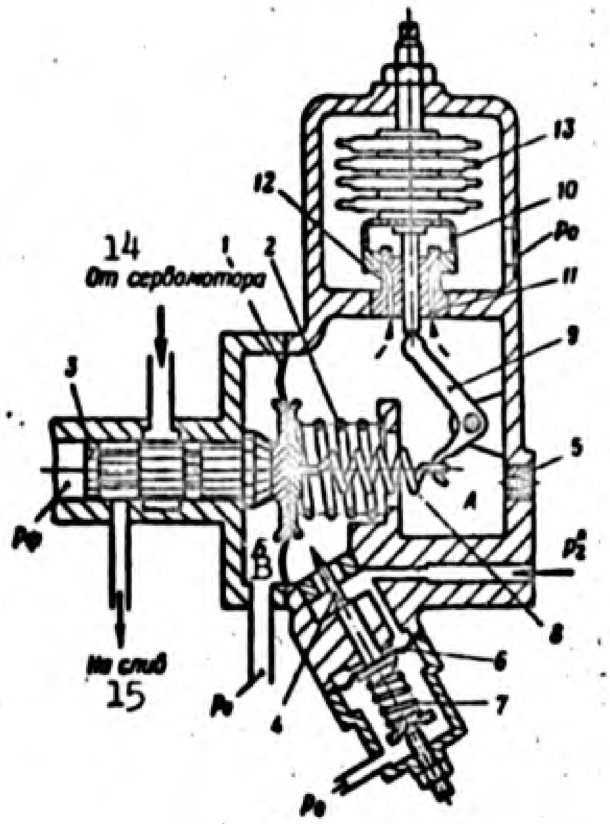

Fig. 283. Automatic acceleration control using variable adjustment based on pressure drop at the compressor, with altitude correction. 1 and 6) Elastic diaphragms; 2, 7, and 8) springs; 3) slide valve; 4) specially shaped needle; 5) nozzle; 9) lever; 10) slide valve; 11 and 12) scavenging nozzles; 13) aneroid unit; A and B) chambers; 14) from servomotor; 15) to discharge line.

with the change in the air pressure drop at the compressor with the aid of a specially shaped needle 4 (Fig. 283), attached to an elastic diaphragm 6 whose position depends on the pressure drop. A suitable choice of the shape of needle 4, as well as of the tension force and stiffness of the spring will provide an optimum rate of fuel feed to the engine during rotor acceleration under ground conditions. With increasing flight altitude, the actual adjustment of the automatic device begins to depart from the required adjustment, since the excess fuel supplied to the engine during acceleration must be reduced (see

Fig. 280). With increasing flight altitude, the limit of maximum permissible fuel feed rates drops downward and its slope becomes less. The automatic acceleration control should be appropriately readjusted.

An aneroid unit 13 (see Fig. 283) is used as the sensing element responding to variations in atmospheric pressure; as this element expands, the resultant force of the springs loading the diaphragm will drop and the scavenging nozzles 11 and 12 will open at specific altitudes.

An important feature of this automatic device is the need for providing exactly the required adjustment of the device for operation at high altitudes, since the adjustment tolerances (automatic-device errors) become commensurate with the required excess amounts of fuel during engine acceleration at high altitudes. Thus engine acceleration at high altitudes is hampered or becomes altogether impossible if the error of the automatic device is equal to or greater than the required excess amount of fuel and is opposite in sign.

Certain systems use devices that retard the mechanical resetting of speed regulators to provide satisfactory engine acceleration at high altitudes. Even here, it is impossible to do without an automatic acceleration control of the type discussed, since where a centrifugal regulator is used, there is a zone of manual speed regulation in which a hydraulic retarder will not operate, and engine-rotor acceleration is governed by an automatic acceleration control.

2. AUTOMATIC HYDRAULIC-RETARDER ACCELERATION CONTROLS

A hydraulic-retarder automatic acceleration control is a hydraulic servomotor working in a time-programed servosystem. It is a component of the mechanism that resets the centrifugal regulator to the desired engine-rotor speed.

The operation of a hydraulic-retarder automatic acceleration control is essentially as follows: when the engine control lever is moved rapidly, a special servomechanism readjusts the speed-governor springs from values corresponding to the speed at which the regulator comes into automatic operation to values corresponding to maximum speed over a specific time interval.

The basic elements of the servomotor are a single-acting servopiston 1 (Fig. 284), which acts through a two-arm lever 2 on a spring 3 of the centrifugal-pendulum sensing element, and a control element — the hemisphere valve 4, mounted in housing 5. Rod 6 is attached to the servopiston; it has holes that connect chamber A of the servopiston cylinder to the discharge line. Working fluid is sent into chamber A under constant pressure through duct 17 and throttle unit 7; it flows out through nozzle 18 in rod 6. The fuel-pressure force exerted on the servopiston is balanced by the forces due to springs 8 and 3, and depends on the amount of fluid discharged from chamber A through nozzle 18. The amount of fluid discharged in turn depends on the position of valve 4.

Spring 9 and moving plate 10 are installed on housing 5 of valve 4; nut 11, which turns in cylinder 12 acts on the moving plate; through a rack cut into the cylinder 12 and gear teeth 13, the cylinder is connected to lever 14 of the throttle valve.

With the engine inoperative, when the throttle-valve lever is in the shutoff position, there is a gap between nut 11 and plate 10; the servopiston and valve 4, together with housing 5 of the valve, will be pressed against stop screw 15 by the forces due to springs 8 and 3; screw 15 adjusts the point at which automatic operation of the centrifugal regulator begins (for a further discussion of this, see Chapter 21, section "Blocking of Automatic Speed Governor in Operating Regimes of Constant Fuel-Feed-Rate Regulator"). This gap is taken up when the valve control lever is in a position corresponding to the angle β_{nar}, as in the case considered in this section.

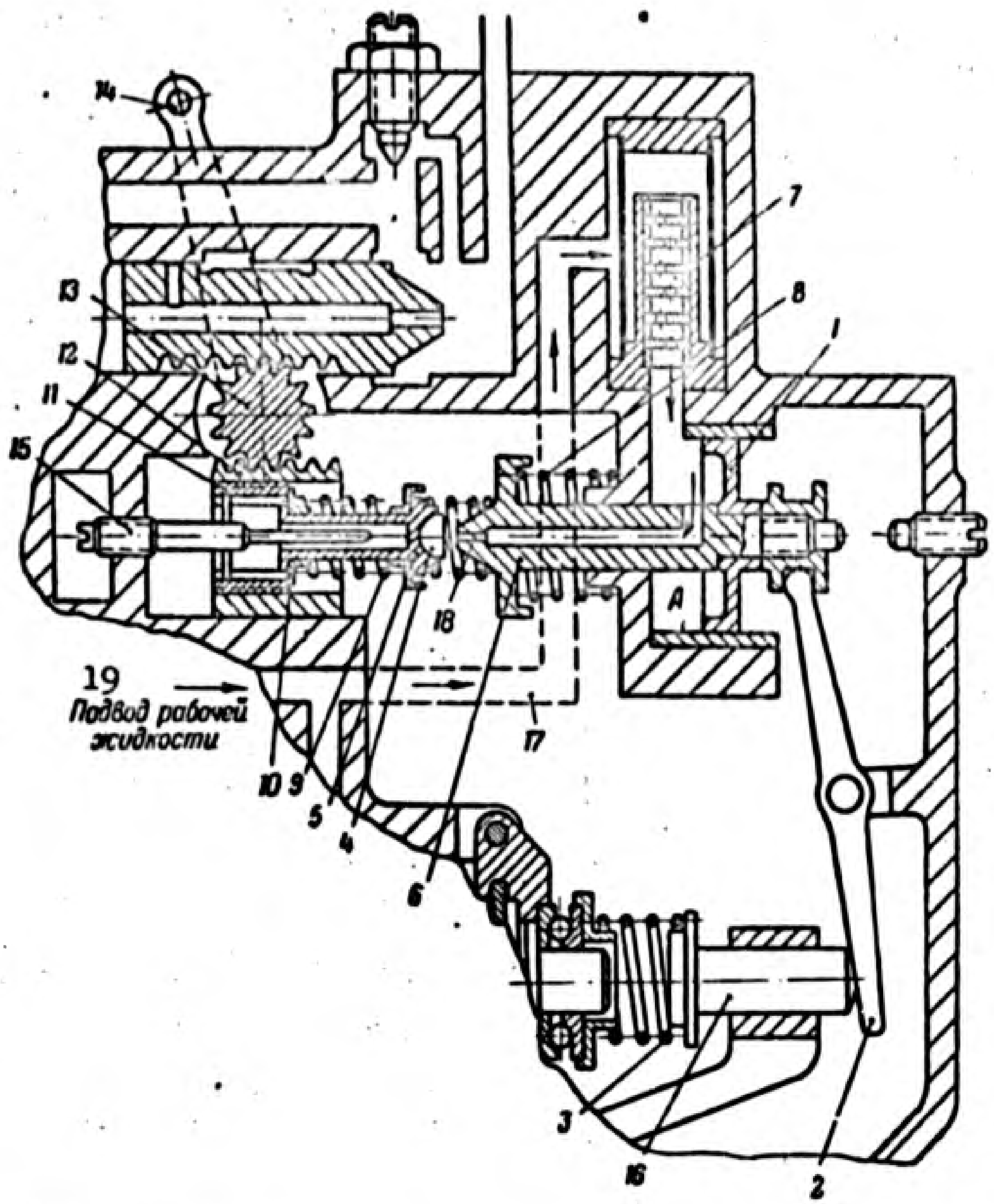

Fig. 284. Basic arrangement of hydraulic retarder. 1) Servopiston; 2) two-arm lever; 3, 8, and 9) springs; 4) hemisphere valve; 5) housing; 6) rod; 7) throttle element; 10) moving plate; 11) nut; 12) cylinder; 13) gear; 14) throttle-valve lever; 15) adjusting screw; 16) rod; 17) duct; 18) nozzle; A) chamber; 19) working-fluid intake.

When the control lever is rapidly moved toward increasing speed from n_{nar} to $n > n_{nar}$, plate 10 will move rapidly, compressing spring 9, owing to which valve 4 will rapidly cover nozzle 18. The outflow of fluid from chamber A through nozzle 18 ceases, while this chamber continues to receive fluid arriving from the system through throttle element 7. The servopiston moves slowly to the right at a rate depending on the rate at which fuel flows through the throttle unit and on the servopiston area. Through lever 2 and rod 16, the piston resets spring 3 of the regulator.

As the servopiston moves to the right, spring 9 straightens out and causes valve 4 to follow the servopiston. When the servopiston has moved to the right by about the amount by which spring 9 was initially compressed, it ceases to move, since nozzle 18 has opened, which causes the pressure in chamber A to drop to the value at which the fuel-pressure force balances the resultant force produced by springs 8 and 3.

In order to make the process of engine-rotor acceleration automatic over a speed-variation range of n_{mg} to n_{nar}, an additional automatic pneumatic acceleration control is inserted into an automatic control system using a centrifugal regulator.

3. AUTOMATIC TACHOMETER ACCELERATION CONTROLS

An automatic tachometer acceleration control operates on the principle that the required excess fuel feed rates during engine-rotor acceleration are provided in accordance with the speed that has actually been reached during acceleration.

The main element of the automatic device is a hydraulic servomotor, which consists of a slide valve 1 (Fig. 285), servopiston 2, and mechanical proportional feedback elements: gear 3, shaft 4, and cam 5.

The slide valve 1 acts as the sensing and control element. It receives the forces due to springs 6 and 7, as well as the force due to the pressure of the working fluid, which is applied to the chamber containing spring 6, and which depends on the engine-rotor speed.

The servopiston is loaded by the force due to spring 8 and the force due to the pressure of the liquid supplied to the servopiston-cylinder chamber through nozzle 9, which determines the rate at which chamber A fills and, consequently, the rate at which the servopiston

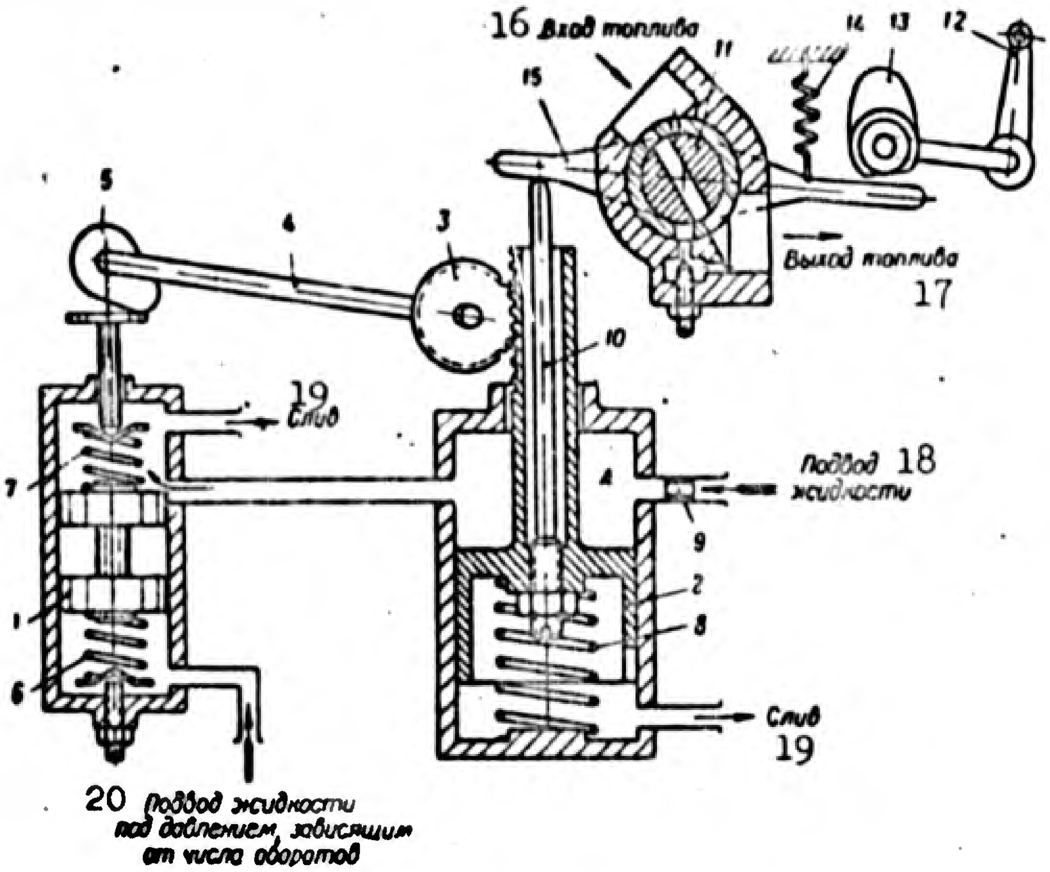

Fig. 285. Basic arrangement of automatic tachometer acceleration control. 1) Slide valve; 2) servopiston; 3) gear; 4) shaft; 5 and 13) cams; 6, 7, 8, and 14) springs; 9) nozzle; 10) rod; 11) throttle valve; 12) control lever; 15) valve control lever; A) cavity; 16) fuel intake; 17) fuel outlet; 18) fluid intake; 19) discharge line; 20) supply of fluid under pressure depending on speed.

moves. Fluid flows out of chamber A at a rate that depends on the position of slide valve 1.

Rod 10, which is connected to the servopiston, is a movable stop, which limits the amount by which throttle valve 11 is opened.

The throttle valve is controlled with the aid of control lever 12, cam 13, and spring 14. When the control lever is turned toward increasing engine speed, cam 13 engages valve lever 15, so that the valve is opened by the force of spring 14. Cam 13 provides the required relationship between the valve position and the required fuel flow rates in steady engine operating regimes.

When the engine is running idle, the valve lever 15 rests against the stop of cam 13, while there is a gap between the servopiston rod 10 and valve lever 15. Spring 14 causes lever 15 to turn through the gap until it reaches the stop on rod 10; this provides a specific initial excess rate of fuel feed to the engine, leading to an increase in rotor speed. The increase in speed increases the pressure of the working fluid in the cylinder chamber of slide valve 1, and moves the slide valve so as to reduce the rate at which fluid flows out of chamber A of the servopiston cylinder. The servopiston moves downward, permitting the throttle valve to open, thus increasing the supply of fuel to the engine.

At the same time, the feedback elements cause the servopiston to move slide valve 1 downward, so that the servopiston ceases to move. Then the forces produced by springs 6 and 7 increase in accordance with the increasing force due to fuel pressure, which acts on the slide valve.

The amount by which the servopiston moves downward depends on the gear ratio in the feedback loop. In accordance with the prescribed program, to each speed actually reached during the acceleration process, there should correspond a definite position of the servopiston. This correspondence is provided by the shape of the feedback cam 5.

After the engine-rotor acceleration process has concluded, lever 15 rests against the stop on cam 13; rod 10 then has no effect on the position of the valve.

The advantage of an automatic tachometer acceleration control lies in the fact that it permits fairly simple governing of the required fuel feed rate during acceleration in accordance with an optimum program.

4. TIME-PROGRAMED DEVICE RESTRICTING THE FUEL-PRESSURE INCREASE AHEAD OF INJECTORS

A time-base device restricting the fuel pressure ahead of the injectors is a time-programed adjusting device.

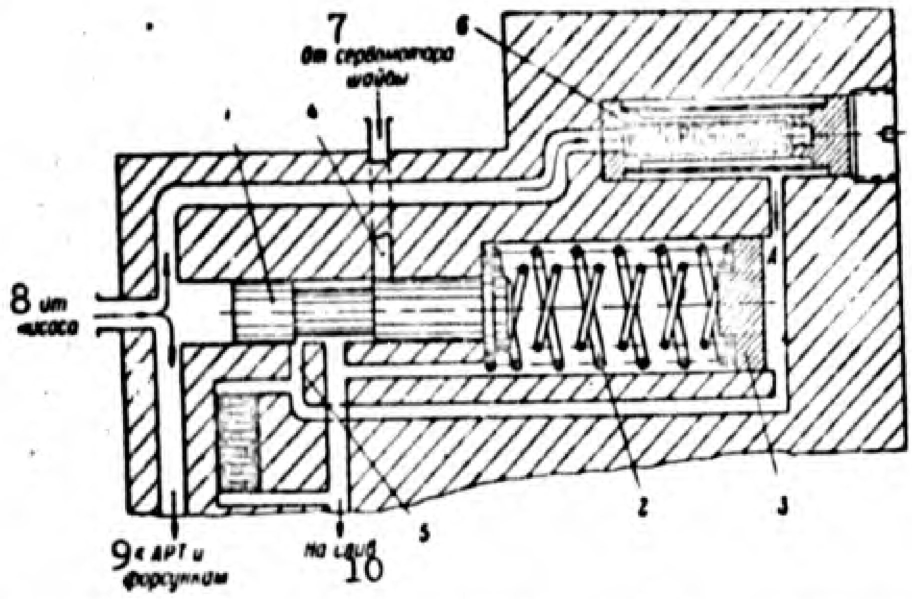

Fig. 286. Device restricting increase in fuel pressure ahead of injectors. 1) Slide valve; 2) spring; 3) piston; 4 and 5) ducts; 6) throttle unit; A) chamber; 7) from inclined-disk servomotor; 8) from pump; 9) to ART and injectors; 10) to discharge line.

Slide valve 1 (Fig. 286) is at the same time a sensing and control element for the servomotor. On the left, the fuel pressure ahead of the automatic fuel distributor (ART) acts on the slide valve, while on the right it receives the force due to spring 3, which is subject to variable adjustment through the motion of piston 3.

In steady regimes, as well as when the control lever is moved slightly or smoothly, so that the fuel pressure ahead of the ART remains below the limiting value, the slide valve closes duct 4, and does not interfere with the operation of the fuel flow-rate regulator. Under sharp and sizeable movements of the control lever, where the fuel pressure ahead of the ART tends to exceed the limiting value, the slide valve moves somewhat to the right, thus reducing the speed with which the inclined-disk servopiston moves, and reducing the rate of increase of the fuel pressure.

The restricting-device readjustment piston 3 moves to the left and increases the tension on the spring since chamber A fills owing to the increase in fuel pressure ahead of the ART and the opening of duct 5. The rate at which this chamber fills is the decisive factor determining the time performance characteristic of the restricting device.

For given design parameters, the choice of the restricting-device time characteristic (its adjustment) is made by changing the hydraulic resistance of throttle element 6.

This device (as with the automatic hydraulic-retarder acceleration control) requires correction for external conditions, which involves design complications.

Chapter 24

ADDITIONAL AUTOMATIC DEVICES IN FUEL-SUPPLY SYSTEM FOR MAIN COMBUSTION CHAMBERS

The additional automatic devices used in the fuel-supply system for the main combustion chambers include: devices limiting the minimum fuel pressure ahead of the injectors, maximum permissible fuel-pressure limiters and maximum pump throughput limiters, devices limiting the maximum air pressure after the compressor, automatic devices (valves) for releasing fuel when a jet projectile is fired [27].

1. AUTOMATIC DEVICES LIMITING THE MINIMUM FUEL PRESSURE AHEAD OF INJECTORS

As the flight altitude increases and aircraft speed decreases, the required rate of fuel supply to the engine for constant rotor speed is reduced by decreasing the fuel pressure ahead of the injectors.

When running idle at high altitudes, as well as when the control lever is pulled back sharply, the fuel pressure ahead of the injectors may drop below the minimum permissible value based on conditions for good fuel atomization, impairing combustion stability and leading to engine shutdown. The minimum permissible fuel pressures ahead of the injectors that will provide combustion stability without flame interruption and extinction are determined experimentally for each actual engine type, and will vary slightly as a function of altitude and flight speed.

In order to keep the fuel pressure from dropping below the minimum permissible value, automatic limiting devices are used for the minimum fuel pressure ahead of the injectors; they are <u>direct-acting</u> or <u>indirect</u>

minimum-pressure valves.

A direct-acting minimum-pressure fuel valve opens a parallel passage for fuel that runs directly from a chamber ahead of the throttle valve to a chamber after the valve when the fuel pressure following the valve (ahead of the injectors) drops below the minimum permissible value.

Figure 287 shows the basic arrangement of a valve connected into a system with a barostat. Valve 2 moves within the throttle-valve housing in guide sleeve 8, and is loaded by the force produced by the difference in the fuel pressures before and after the valve. The valve

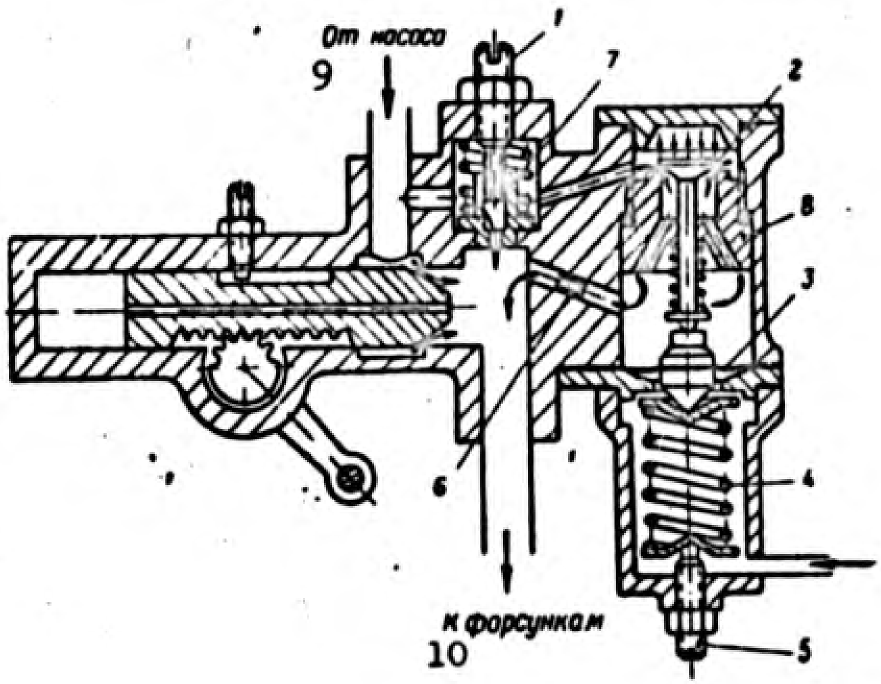

Fig. 287. Direct-acting minimum-fuel-pressure valve (ahead of injectors). 1 and 5) Adjusting screws; 2) valve; 3) diaphragm; 4) spring; 6) nozzle; 7) sleeve; 8) guide sleeve; 9) from pump; 10) to injectors.

sensing element is the diaphragm 3, which receives the force produced by the difference in fuel pressures before the injectors and at the discharge line, as well as the force produced by spring 4, whose tension determines the minimum permissible fuel pressure ahead of the injectors; the spring tension is regulated by screw 5.

On the ground, with the throttle-valve lever set against the idle-running stop, the fuel pressure ahead of the injectors is greater than

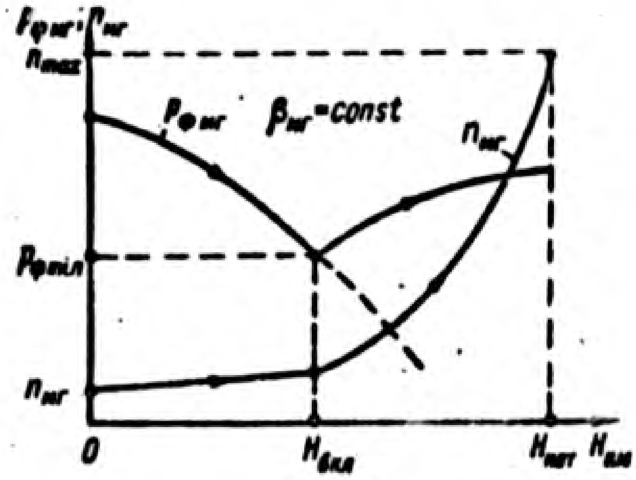

Fig. 288. Variation in minimum fuel pressure ahead of injectors and idle-running rotor speed with flight altitude in the presence of a direct-acting minimum-pressure valve.

the minimum permissible value. The force due to this pressure then bends diaphragm 3 downward, and valve 2 closes the parallel passage for fuel to the injectors. Fuel can go to the injectors only through the idling duct, whose area in sleeve 7 is adjusted by screw 1, and through the small slot in the valve. As the flight altitude increases with the control lever against the idling stop, the fuel pressure ahead of the injectors will drop owing to the operation of the barostat regulator, which maintains the rotor speed n_{mg} (Fig. 288) nearly constant up to the altitude at which valve H_{vkl} comes into play. Thus the total force due to fuel pressure on valve 2 (see Fig. 287) and diaphragm 3 of the valve will be reduced, and at a certain flight altitude H_{vkl}, the valve will be forced by spring 4 to open the additional passageway for fuel through holes drilled in guide sleeve 8, preventing a reduction in fuel pressure ahead of the injectors.

After the valve has come into play, the fuel pressure ahead of the injectors will not remain constant as the altitude increases, but will rise somewhat (see Fig. 288), owing to the fact that the direct-acting valve admits more fuel to the injectors than is required. The speed will then increase to a maximum value n_{max} at an altitude H_{pot}

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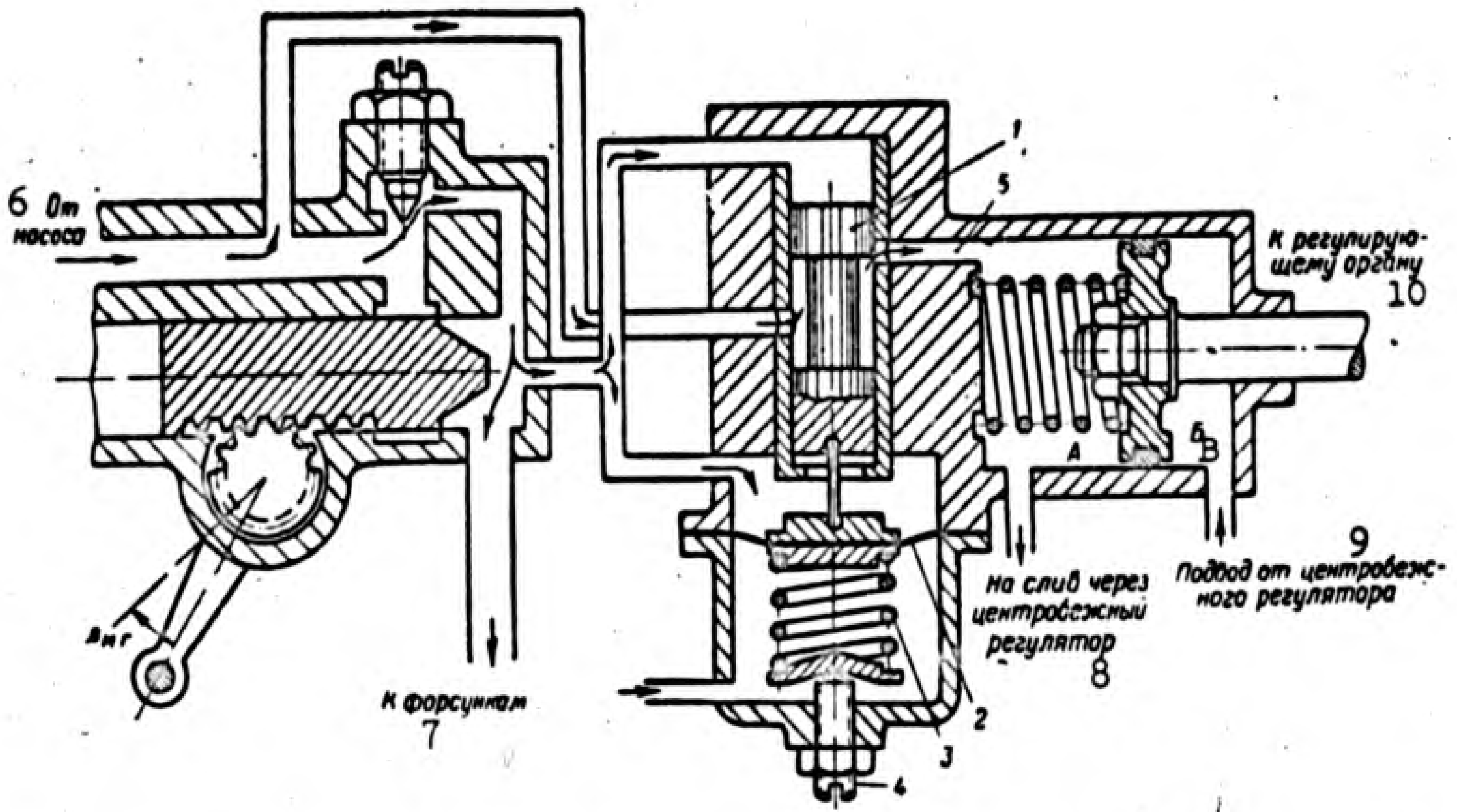

Fig. 289. Indirect minimum-fuel-pressure valve. 1) Slide valve; 2) diaphragm; 3) spring; 4) adjusting screw; 5) duct; A and B) chambers; 6) from pump; 7) to injector; 8) to discharge line through centrifugal regulator; 9) intake from centrifugal regulator; 10) to final control element.

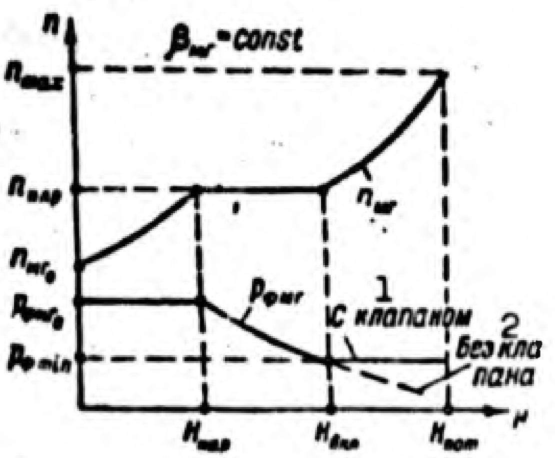

Fig. 290. Variation in speed with flight altitude in the presence of a constant fuel-pressure-drop valve on throttle valve, a centrifugal speed regulator, and a minimum fuel-pressure valve ahead of the injectors. 1) With valve; 2) without valve.

called the engine ceiling.

The regulating elements of the valve are (see Fig. 287) screw 5, spring 4, and nozzle 6. In order to increase the maximum fuel pressure ahead of the nozzles at high altitudes (i.e., in order to reduce the altitude at which the valve is actuated), it is necessary to turn screw 5 of spring 4. If, in addition, it is necessary to provide a smaller variation in fuel pressure with altitude, the nozzle diameter should be reduced as this will reduce the supply of fuel to the injectors.

An increase in the maximum fuel pressure ahead of the injectors as the flight altitude increases will lead to a reduction in engine ceiling. For this reason, indirect minimum fuel-pressure valves are more commonly used. They act on the position of the final control element with the aid of a servomotor. Figure 289 shows a valve connected into an automatic control system with a centrifugal speed regulator and a constant-fuel-feed-rate regulator.

The sensing element of this valve is diaphragm 2; the pressure of the fuel ahead of the injectors is applied to one side of this diaphragm, and the pressure of the fuel at the pump intake to the other. If the fuel pressure ahead of the injectors is higher than the minimum permitted value, the force due to the difference in fuel pressures on the two sides of the membrane will be greater than the force produced by spring 3, and the valve will rest against the stop in the lowest position. Here the ring of slide valve 1 will cut off duct 5 through which working fluid is supplied to the servomotor, which is controlled completely by the constant-fuel-feed-rate regulator or the centrifugal speed regulator.

With the control lever at the idle-running stop, as the flight altitude increases, the regulator holding the fuel feed rate constant comes into operation; this maintains the fuel pressure ahead of the injectors constant. As a result, the rotor speed during idling rises to a value corresponding to the point at which automatic operation of the centrifugal regulator begins (Fig. 290). With a further increase in flight altitude, the rotor speed remains constant and equal to n_{nar} owing to the decreased inclination of the pump disk produced by the centrifugal regulator. The fuel pressure ahead of the injectors is reduced, and may reach an impermissibly low value. When the pressure drops below the minimum permissible value, as set by the tension of spring 3 (see Fig. 289) with the aid of screw 4, the minimum fuel-pressure valve comes into operation, since diaphragm 2 is bent upward by the force produced by spring 3. Slide valve 1 opens the channel through which working fluid enters chamber A of the servomotor; as a result, the supply of fuel to the injectors is increased, and the fuel pressure ahead of the injectors is restored (to the minimum permissible value). With increasing flight altitude, the constant fuel pressure ahead of the injectors causes the rotor speed to rise to the maximum permissible value at the engine ceiling. Thus, after the valve has come into operation, its effect on the position of the final control element counteracts that due to the centrifugal regulator; when the minimum-pressure valve operates, it has a greater effect on the servo-piston position than does the centrifugal regulator, which tends to maintain the speed n_{nar} constant, i.e., the speed for which the sensing-element spring in the regulator has been adjusted.

2. DEVICES LIMITING THE MAXIMUM FUEL PRESSURE AND MAXIMUM THROUGHPUT OF PUMPS IN FUEL-SUPPLY SYSTEMS. DEVICES LIMITING MAXIMUM AIR PRESSURE AFTER THE COMPRESSOR. AUTOMATIC DEVICES FOR RELEASING FUEL DURING FIRING [27]

When the throttle valve closes a line rapidly, and also during high-speed flight at low altitudes and low ambient temperatures, where the required fuel feed rate rises sharply, it is possible for the fuel pressure to rise above the maximum value permitted if damage to the pump and fuel-line elements is to be prevented.

In order to limit the maximum pressure of the fuel in fuel-supply systems, special automatic limiting devices are used. In most cases, such a limiting device is a direct-acting reducing valve that detects either the absolute fuel pressure ahead of the throttle valve or the pressure drop across this valve, and which transfers fuel from the pump outlet to its intake.

Very frequently, the maximum rate at which fuel is fed to the engine is restricted by using a stop to arrest the motion of the piston-pump inclined disk.

In systems providing for limitation of the maximum air pressure after the compressor, the limitation is carried out by automatically restricting the rate at which fuel is fed to the engine.

An automatic device (valve) for releasing fuel during firing reduces the supply of fuel to the engine when an aircraft fires jet projectiles, or cannon located near the compressor intake device are fired, since penetration of powder-produced gases within the TRD may cause a flameout. Gases entering the engine at high temperature will reduce the relative rotor speed and air flow rate. While the rate at which fuel is supplied to the engine remains nearly unchanged, the temperature of the gases ahead of the turbine will rise, leading to unstable compressor operation (surging). In addition, penetration of powder-produced gases to the combustion chamber will impair the process of fuel combustion. These factors can also cause engine flameout.

The fuel-release valve control interlocks with the fire control. During firing, the release valve readjusts the speed regulator to a lower speed, thus reducing the rate at which fuel is fed to the engine [27].

Chapter 25

AUTOMATIC CONTROL SYSTEMS FOR AFTERBURNERS
[17, 18, 19, and 27]

1. GENERAL INFORMATION

As we know, the most common method for augmenting the thrust of a TRD [turbojet engine] is combustion of additional fuel in a special afterburner behind the engine turbine. The condition requiring the operation of the engine turbocompressor to remain constant when the afterburner is turned on dictates an increase in the nozzle throat area; this increase must be the larger the more the gases in the afterburner are heated and, consequently, the greater the degree of thrust augmentation. If after the afterburner has been cut in, the increased nozzle throat area remains constant, the afterburner is called a fixed-geometry afterburner. There is only one regulating factor for such an afterburner, the rate G_{tf} at which fuel is fed to it. If with the afterburner on, it is possible to change the nozzle throat area in steps or smoothly, the afterburner is called a variable afterburner; in addition to G_{tf}, there is one more regulating factor with such a burner – the nozzle throat area F_{sf}.

In a fixed-geometry afterburner, i.e., a burner having one regulating factor (G_{tf}), there is just one corresponding control parameter. As the control parameter, we may use the gas temperature ahead of the turbine, T^*_3. Since when the afterburner is on, the engine is required to deliver maximum thrust, we should take as the control program the condition T^*_{3max} = const. There will be the following relationship between the controlling factor G_{tf} and the controlling parameter T^*_3: as G_{tf} increases, there will be an increase in the temperature T^*_f of the afterburner gases, so that if the condition F_{sf} = const is satisfied, there will be an increase in the gas pressure following the turbine and a drop in the gas pressure drop at the turbine; this will cause the engine rotor speed to decrease. Where there is a rotor-speed regulator operating on the deviation-control principle, this speed reduction will be eliminated by an increase in the rate at which fuel is fed to the main combustion chambers and by an increase in the gas temperature ahead of the turbine. When the fuel feed rate G_{tf} drops, the entire process is reversed. Thus we can see that an increase in G_{tf} leads to an increase in the temperature T^*_3. An increase in the temperature T^*_3 may be eliminated by reducing the rate at which fuel is fed to the afterburner.

As the controlled variable, we may use the gas temperature T^*_f in the afterburner. Here the relationship between T^*_f and G_{tf} is self-evident: an increase in G_{tf} leads to an increase in T^*_f. It is also clear that an increase in T^*_f may be eliminated by reducing G_{tf}. With a control program $T^*_{f\,max}$ = const, the temperature T^*_3 will in general not remain constant, which is a drawback to this choice of controlled variable.

We may also use the gas temperature T^*_4 at the turbine outlet as the controlled variable. It is lower than T^*_3 and T^*_f, and may be measured with great accuracy. Where the control program T^*_{4max} = const is selected, however, we are still left with the drawback mentioned, i.e., there is some change in the temperature T^*_3 when the air temperature at the compressor intake varies.

The difficulties involved in measuring the temperatures T^*_3, T^*_f, and T^*_4 make it necessary to seek other controlled variables that can be measured with reliable fast-responding devices. Such a parameter may be the ratio of the air pressure p^*_2 after the compressor to the pressure of the gases after the turbine, p^*_4. Investigations have shown that this ratio p^*_2/p^*_4 remains nearly constant when the turbocompressor is running at top speed, no matter how the external engine operating conditions change. This makes it possible to use the simple control program p^*/p^*_4 = const. The following relationship holds between p^*_2/p^*_4 and G_{tf}: as G_{tf} increases, the ratio p^*_2/p^*_4 decreases owing to the increase in p^*_4 caused by the increase in temperature T^*_f. It is clear that in order to prevent the ratio p^*_2/p^*_4 from dropping below a prescribed value, it is necessary to reduce the rate at which fuel is supplied to the afterburner.

Where a variable-geometry afterburner is used, one of the controlled variables may be T^*_3, T^*_4, or p^*_2/p^*_4. The other controlled variable may be the temperature T^*_f. Implementing one of the programs T^*_{3max} = const, T^*_{4max} = const, or $(p^*_2/p^*_4)_{max}$ = const by varying, for example, the rate at which fuel is supplied to the afterburner, we can hold T^*_f constant by changing the nozzle throat area or vary it in order to change the degree of thrust augmentation and thus the engine thrust with the afterburner on.

Fuel may be fed to the afterburner by fixed centrifugal injectors. Afterburner fuel pumps will usually be of the piston or gear types, driven by the engine rotor. A special cycle valve 1 is usually installed between the injector assembly and the pump (see Fig. 293). With given injector characteristics, this valve makes it possible to obtain the desired relationship between the rate at which fuel flows through the entire injector assembly and the fuel pressure p_{nf} after the pump (Fig. 291). The fuel pressure p_{nf} is higher than the fuel pressure p_f ahead of the injectors; the difference $p_{nf} - p_f$ depends on

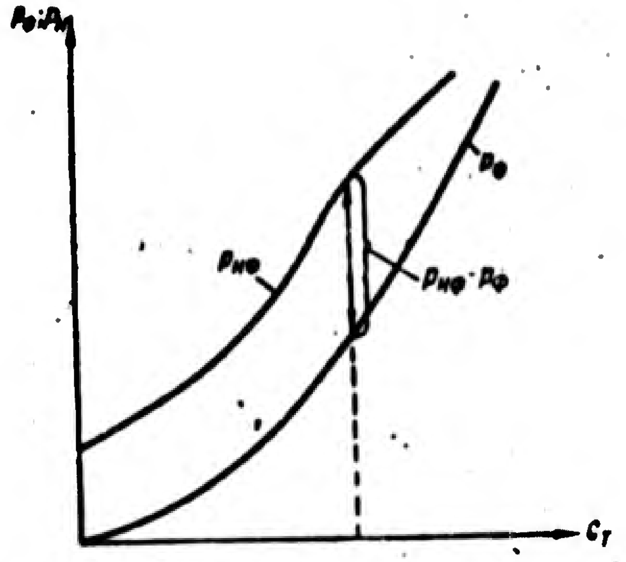

Fig. 291. Fuel pressure after pump and ahead of afterburner manifold nozzles as a function of fuel flow rate.

the shape of the valve and the characteristic of spring 2 (see Fig. 293). To permit the rate at which fuel flows into the afterburner to be regulated, the fuel pump is provided with a control element: an inclined disk (for a piston pump) or bypass valve (for a gear pump). The afterburner fuel pump usually has a valve 3 that is used to turn the afterburner on and off. The motion of valve 3 is time-programed to ensure that the delivery of fuel when the afterburner is turned on follows the law required by the conditions that ensure reliable afterburner starting. In order to avoid a severe disturbing effect on the turbocompressor when the afterburner is turned on and off, the fuel feed rate is matched appropriately to the nozzle-flap aperture when the afterburner is turned on or off.

<u>Open</u>- or <u>closed</u>-loop control systems may be used for afterburners.

2. OPEN-LOOP AUTOMATIC CONTROL SYSTEMS FOR AFTERBURNER FUEL DELIVERY

In the general case, in order to provide a desired afterburner control program (for example, T^*_{3max} = const), the rate at which fuel is fed to the afterburner with F_{sf} = const is a function of external parameters – the pressure and temperature of the air at the compressor intake. It turns out that the required rate of fuel supply to the afterburner is directly proportional to the stagnation pressure p^*_1 at the compressor intake, while it is a more complicated function of the air temperature T^*_N. We shall only note that as the temperature T^*_N rises, the required rate of fuel supply to the afterburner will decrease, as shown in Fig. 292. Thus open-loop systems in general should correct the rate at which fuel is fed to the afterburner in accordance with the pressure p^*_1 and the temperature T^*_N. The majority of systems in use, however, only use correction based on p^*_1 for some average value of the T^*_N vs. p^*_1 relationship. This simplifies the control system, but leads to errors in maintaining the desired control program, when the temperature T^*_N begins to depart from the theoretical value for a given p^*_1. It is clear that when the temperature T^*_N goes up and the air flow rate drops, such a system will supply more fuel to the afterburner than is needed, resulting ultimately in an increase in the temperature T^*_3. In order to avoid turbine failure, this possible temperature rise must be taken into account when the design temperature T^*_3 is found with the afterburner in operation. Conversely, when the temperature T^*_N is lowered, there will be some drop in the temperature T^*_3 and the full potential of the TRD afterburner system will not be used.

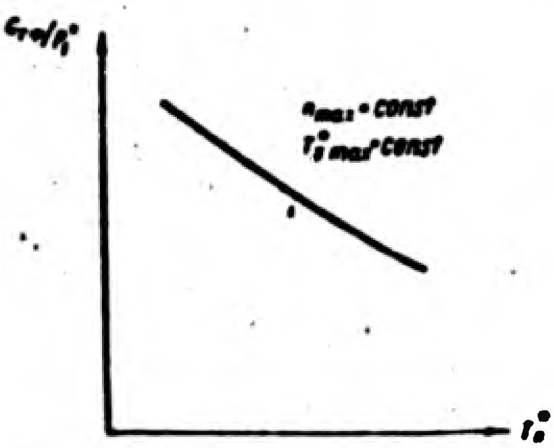

Fig. 292. Required rate of supply of fuel to afterburner as a function of air parameters at compressor intake.

The open-loop afterburner fuel-supply control system shown in Fig. 293 is a servosystem whose input variable is the air pressure p^*_1 at the compressor intake and whose output variable is the fuel pressure p_{nf} at the afterburner-pump outlet. In principle, this system is sim-

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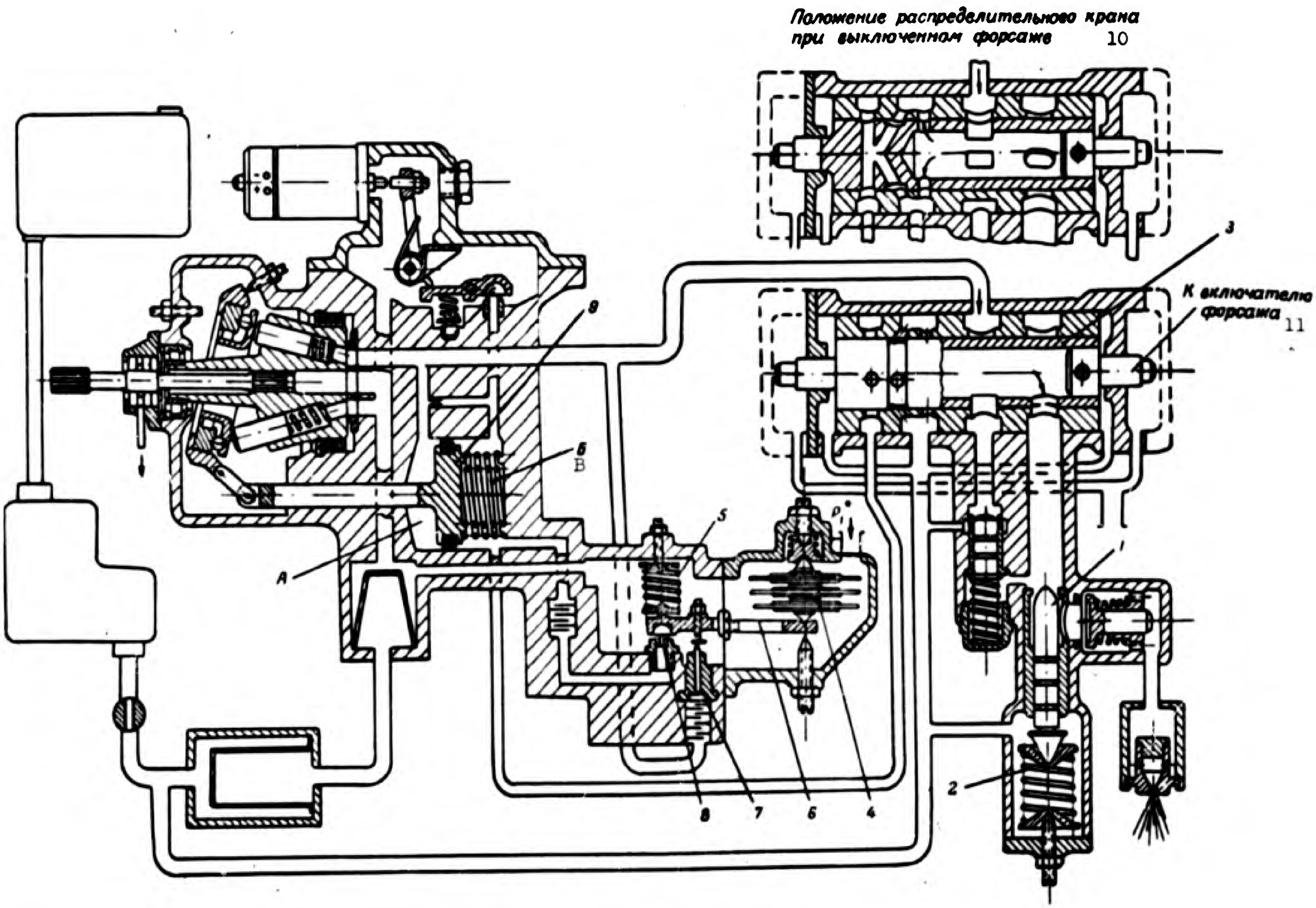

Fig. 293. Design of open-loop control system for afterburner fuel supply. 1) Cycle valve; 2 and 5) springs; 3) valve; 4) aneroid unit; 6) lever; 7) diaphragm; 8) nozzle; 9) servopiston; A and B) chambers; 10) position of distributor valve with afterburner off; 11) to afterburner switch.

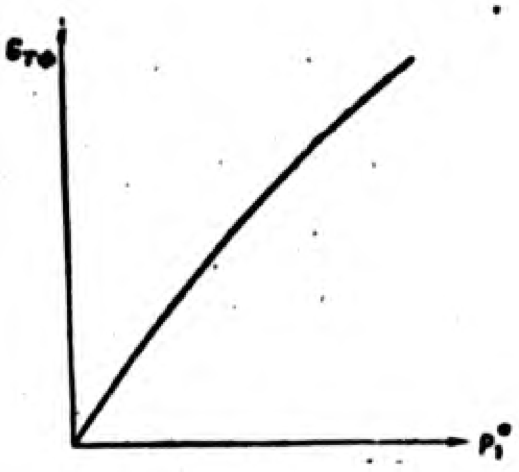

Fig. 294. Required characteristic for open-loop afterburner control system.

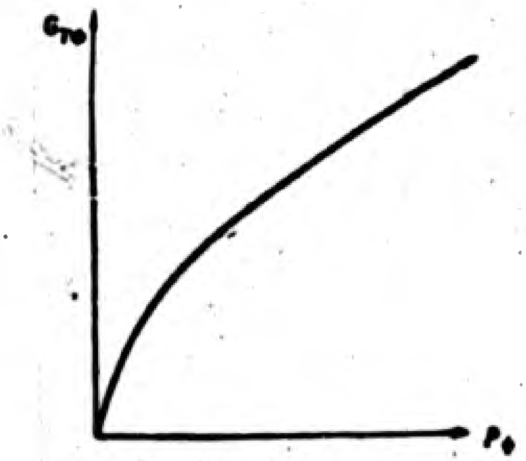

Fig. 295. Characteristic curve for afterburner injector unit.

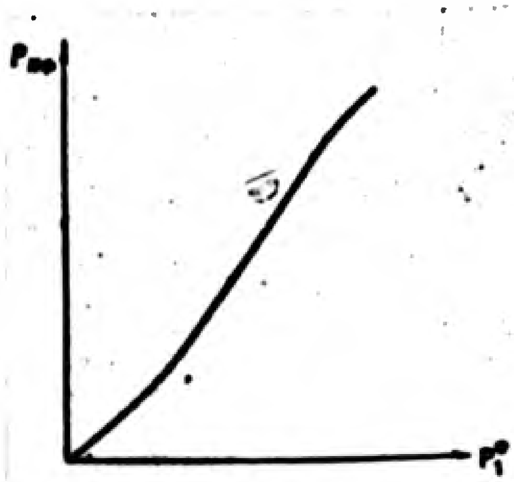

Fig. 296. Required characteristic of afterburner fuel feed-rate regulator.

ilar to the indirect speed-control system examined before (see Chapter 22, section "Fundamental Types of Indirect Speed-Control Systems and Their Properties"). In system design, the basic relationships are the fuel flow rate required by the condition that the desired afterburner control program be maintained as a function of the air pressure p^*_1 (Fig. 294), and the characteristic curve for the injector assembly (Fig. 295). These two relationships make it possible to find the required regulator characteristic connecting the fuel pressure at the pump outlet, p_{nf}, and the air pressure p^*_1 at the compressor intake (Fig. 296). By a suitable choice of shape for valve 1 (see Fig. 293) and of its spring 2 it is possible to make the required regulator characteristic linear, i.e., the relationship between p_{nf} and p^*_1 will be one of direct proportion. This greatly simplifies the implementation of the required characteristic by appropriate choice of the following regulator parameters: dimensions of the aneroid element 4, tension of spring 5, arm ratio of lever 6, dimensions of diaphragm 7, etc.

The regulator operates in the following manner. When, for example, the pressure p^*_1 increases, the aneroid element is compressed, the lever turns, and somewhat reduces the flow area of nozzle 8. The flow of fuel through the nozzle is reduced, which increases the pressure in chamber B so that servopiston 9 moves to the left, increasing the angle of inclination of the disk. Pump throughput then increases along with the fuel pressure p_{nf} and the rate at which fuel is fed to the afterburner. The piston stops moving when the increasing pressure p_{nf} causes lever 6 to turn and open the nozzle just enough to compensate for the effect of the increase in p^*_1. When the pressure p^*_1 decreases, the regulator will operate in exactly the opposite manner.

When the afterburner is on, the regulator acts in the following fashion: valve 3 is opened with the aid of a special electric motor. There is no connection between this valve and the engine control lever, and it can occupy only two extreme positions. In this case, with the nozzle flaps in a fixed position, with the afterburner on, engine thrust can only be controlled by changing the rotor speed, i.e., by changing the rate at which fuel is supplied to the main combustion chamber, since the rate at which fuel is supplied to the afterburner will remain constant and independent of the rotor speed. This method of thrust control with the afterburner on is uneconomical, since a TRD with afterburner makes better use of the fuel delivered to the main chambers, where the air pressure is high, than of fuel introduced into the afterburner. In throttling down engine thrust, it is thus desirable to reduce the rate at which fuel is supplied to the afterburner, rather than the main chamber feed rate. This is impossible, however, with the control method described.

An open-loop afterburner fuel-supply control system may sometimes take the form of a servosystem in which the input variable is p^*_1 and the output variable the quantity $\underline{m}$, which characterizes the position

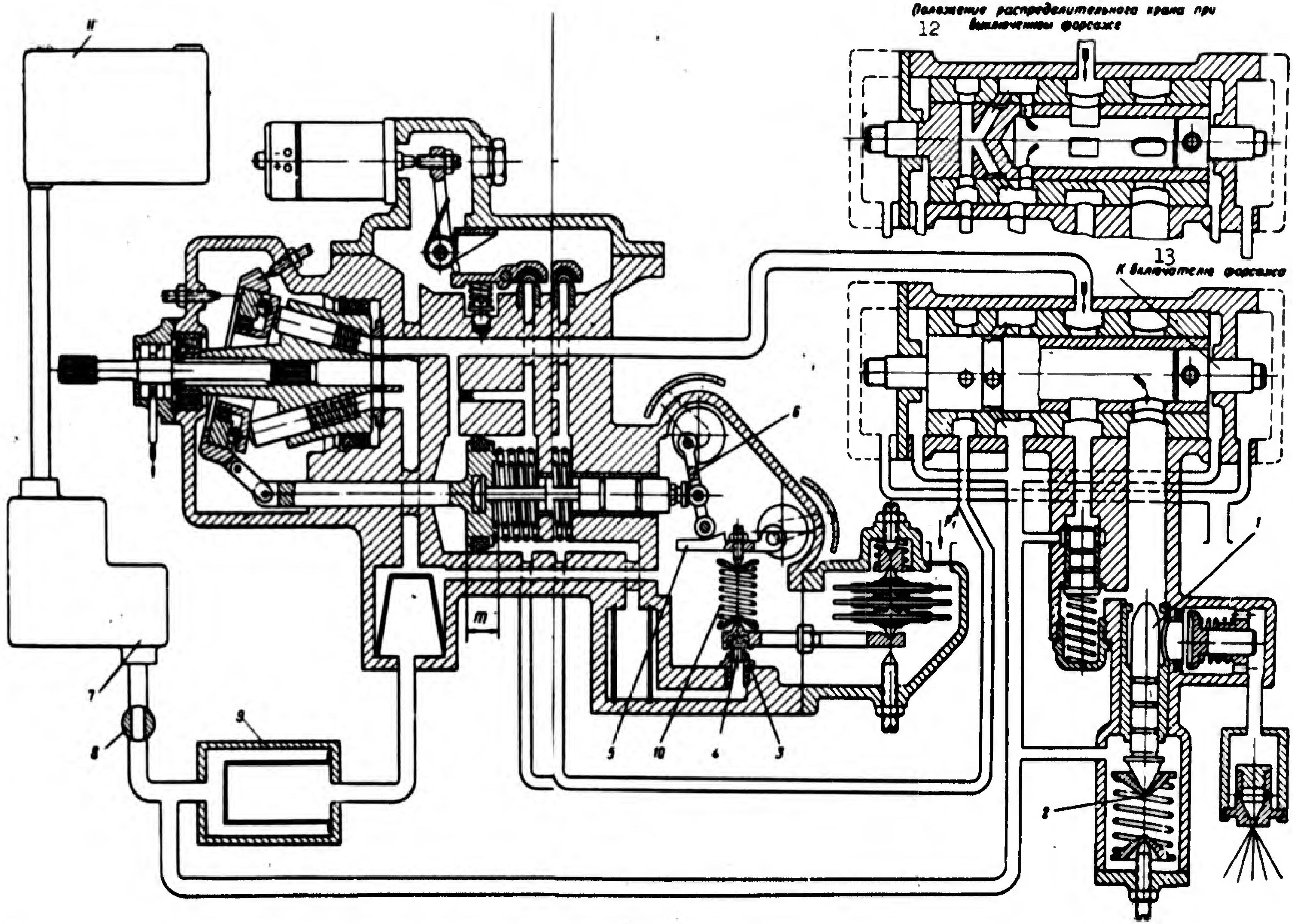

Fig. 297. Design of open-loop afterburner fuel-supply control system. 1) Cycle valve; 2 and 10) springs; 3) valve; 4) nozzle; 5) cam; 6) lever; 7) booster fuel pump; 8) valve; 9) filter; 11) fuel tank; 12) position of distributor valve with afterburner off; 13) to afterburner switch.

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of the fuel-pump final control element (Fig. 297).

The static characteristic of such a regulator is the curve (Fig. 298) representing the position of the pump final control element (in this case, the inclined-disk servopiston) as a function of the air pressure at the compressor intake. In order to find the required regulator characteristic, we use the curve showing the required fuel feed rate as a function of the air pressure p^*_1 (see Fig. 294) and the fuel-pump characteristic (Fig. 299). The required regulator characteristic is provided by proper selection of the shape of cam 5 (see Fig. 297) in the mechanical feedback loop.

Fig. 298. Static characteristic for afterburner fuel-supply regulator.

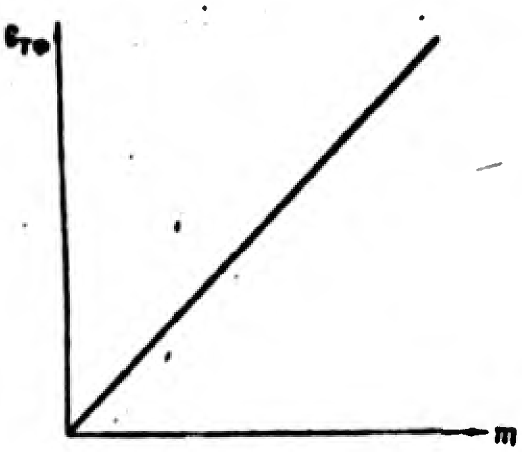

Fig. 299. Characteristic of afterburner fuel pump.

This system operates as follows. Under an increase, for example, in the pressure p^*_1, the aneroid element is compressed, causing valve 3 to close. The reduction in the rate at which fuel flows out of nozzle 4 increases the fuel pressure in the right-hand chamber of the servopiston. The latter begins to move to the left, increasing the rate at which fuel is supplied to the afterburner (the coordinate m increases here). As the servopiston moves to the left, the roller of lever 6 moves along cam 5. The upward motion of the latter results in the same displacement of the stop of spring 10 together with valve 3. Here the flow of fuel out through nozzle 4 increases to the point that the servopiston stops in some definite position. As a consequence, a larger fuel feed rate and a larger value of the coordinate m will correspond to the higher pressure p^*_1.

Since in this control system, the coordinate of the fuel-pump final control element depends on the pressure p^*_1 and not on the speed, when the engine is throttled down with the afterburner on by readjustment of the speed regulator there will occur together with a reduction in rotor speed a decrease in the rate at which fuel is supplied to the afterburner. This advantage of the given control system distinguishes it favorably from that considered previously.

When an open-loop control system is used for the afterburner, a change in nozzle throat area with the afterburner on will affect the operation of the engine turbocompressor in exactly the same way as when the afterburner is off, i.e., an increase in F_{sf} leads to a drop in the temperature T^*_3, while the converse is also true.

3. CLOSED-LOOP AFTERBURNER FUEL-SUPPLY CONTROL SYSTEMS

The unacceptably large static error, as well as the fact that the operation of open-loop control systems does not depend on the operation and condition of the engine itself has led to the appearance of closed-loop fuel-supply control systems. In a closed-loop system, the rate at which fuel is fed to the afterburner is controlled on the basis of a variable that depends on afterburner operation. When the controlled variable is held constant, its control system acts as an automatic-stabilization system.

Figure 300 shows the block diagram of a closed-loop fuel-supply control system in which the temperature T^*_3 is the controlled variable. The regulator has a detector for the temperature T^*_3, and when this temperature departs from the prescribed value, the regulator changes the fuel feed rate. If, for example, with an increase in flight altitude the temperature T^*_3 rises, the regulator moves the fuel-pump final control element so as to reduce the rate at which fuel is fed to the

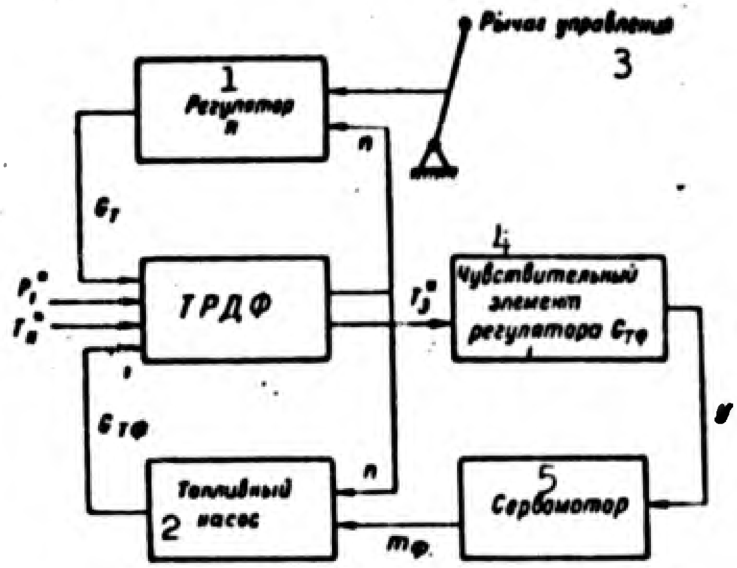

Fig. 300. Block diagram of closed-loop afterburner control system using temperature T^*_3. 1) Regulator; 2) fuel pump; 3) control lever; 4) G_{tf} regulator sensing element; 5) servomotor.

afterburner; it continues to operate until the deviation from the prescribed temperature T^*_3 has been eliminated. During the control process, the speed regulator comes into play to eliminate speed deviations from the prescribed value produced by fluctuations in the temperature T^*_3 and the pressure drop at the turbine. This is a typical case of the interaction of two closed-loop control systems, effected through the engine.

We may also use the gas temperature T^*_4 at the turbine outlet or the afterburner temperature T^*_f as the control parameter. The difficulties involved in creating closed-loop gas-temperature control systems limit the application of such an afterburner control method, however. Fewer difficulties appear in designing closed-loop systems in which the ratio p^*_2/p^*_4 of the air pressure following the compressor to the gas pressure behind the turbine is taken as the controlled variable [27]. A block diagram of such a control system is shown in Fig. 301. This system has a sensing element that responds to a deviation of the ratio p^*_2/p^*_4 from the theoretical maximum value $(p^*_2/p^*_4)_{max}$; the sensing element acts on the fuel feed rate G_{tf} (through a servomotor) so that when the ratio p^*_2/p^*_4 increases, the fuel feed rate will be increased, while when the ratio decreases, the feed rate decreases as much as is needed to eliminate the error in the ratio p^*_2/p^*_4 that has appeared.

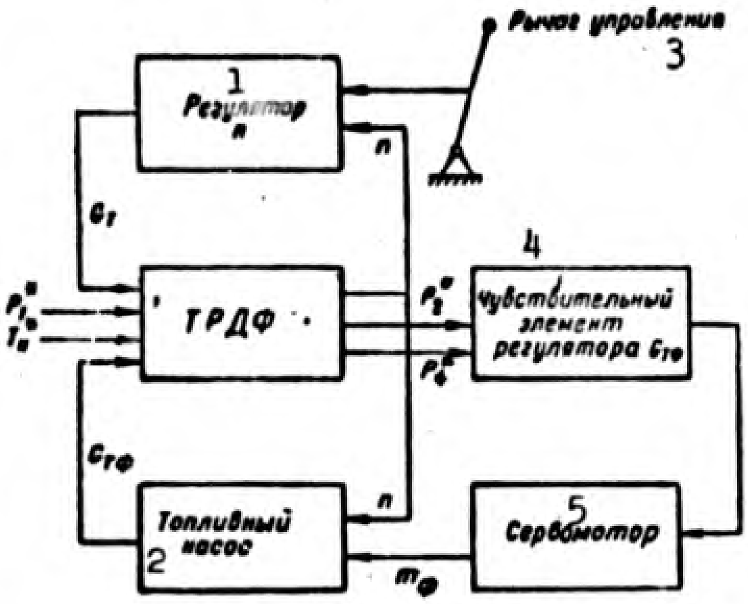

Fig. 301. Block diagram of closed-loop afterburner control system using p^*_2/p^*_4. 1) Regulator; 2) fuel pump; 3) control lever; 4) G_{tf} regulator sensing element; 5) servomotor.

Figure 302 shows the basic arrangement of a lever-type sensing element. Identical aneroid elements 1 and 2, located in chambers A and B, are connected to the ends of a lever 3. Gas at the pressure p^*_4 is supplied to chamber A, and air at the pressure p^*_2 to chamber B. Arms a and b of the lever are so chosen that their ratio will equal the prescribed value of $(p^*_2/p^*_4)_{max}$. If the actual value of p^*_2/p^*_4 equals $(p^*_2/p^*_4)_{max}$, the lever and attached servomotor control element will be in the neutral position. For the system to function properly, an increase in the ratio p^*_2/p^*_4 should cause the servomotor control element to move so as to increase the rate at which fuel is supplied to the afterburner, while a drop in p^*_2/p^*_4 should cause a corresponding displacement of the servomotor control element so as to reduce the fuel feed rate.

Figure 303 shows a diagram of a pneumatic sensing element. Air from the compressor is supplied to chamber A above the diaphragm through nozzle 1 at pressure p^*_2; it leaves this chamber through nozzle 2 and goes to the atmosphere. Here a certain pressure p_x, less

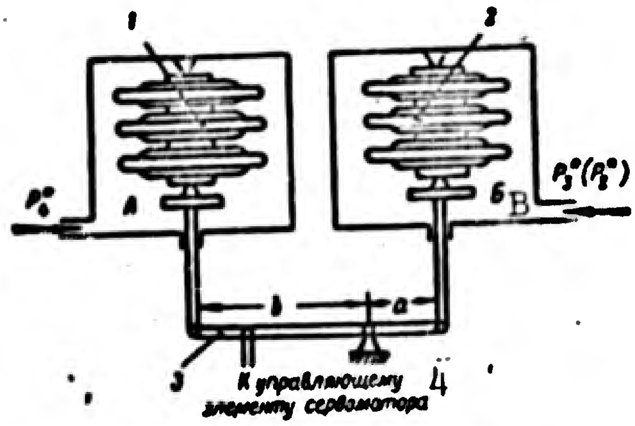

Fig. 302. Basic arrangement of lever-type sensing element for p^*_2/p^*_4 = const regulator. 1 and 2) Aneroid elements; 3) lever; A and B) chambers; 4) to servomotor control element.

than p^*_2, is set up above the diaphragm. If the pressure ratio p_N/p_x is below the critical value at nozzle 2, p_x will not depend on the pressure p_N but will be proportional to p^*_2. By proper selection of nozzles 1 and 2, it is possible to make the pressure p_x equal to pressure p^*_{4max} at the maximum steady operating speed of the turbocompressor. Gas at pressure p^*_4 is applied to cavity B under the diaphragm. The diaphragm is connected to the control element for the servomotor of the afterburner fuel feed-rate regulator. This control element may either be a slide valve or a valve similar to that used in the control system shown in Fig. 297.

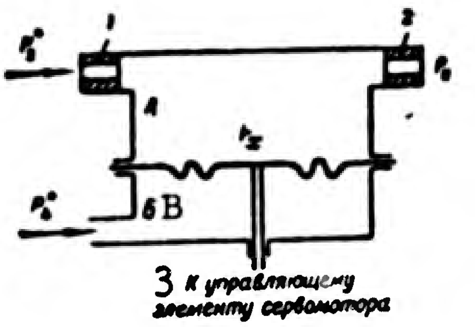

Fig. 303. Basic arrangement of pneumatic sensing element for p^*_2/p^*_4 = const regulator. 1 and 2) Nozzles; A and B) chambers; 3) to servomotor control element.

If the actual pressure p^*_4 in chamber B equals the prescribed value, i.e., $p_x = p^*_{4max}$, the diaphragm will be in the equilibrium neutral position and the control element will exert no effect on the servomotor. On the other hand, if, for example, the rate at which fuel is supplied to the afterburner becomes greater than the required rate owing to an increase in flight altitude or a decrease in nozzle throat area, the pressure p^*_4 in chamber B will exceed the pressure in chamber A, the diaphragm will bend upward and move the servomotor control element, decreasing the fuel feed rate and, consequently, decreasing the pressure p^*_4 until the prescribed value of the ratio $(p^*_2/p^*_4)_{max}$ is restored.

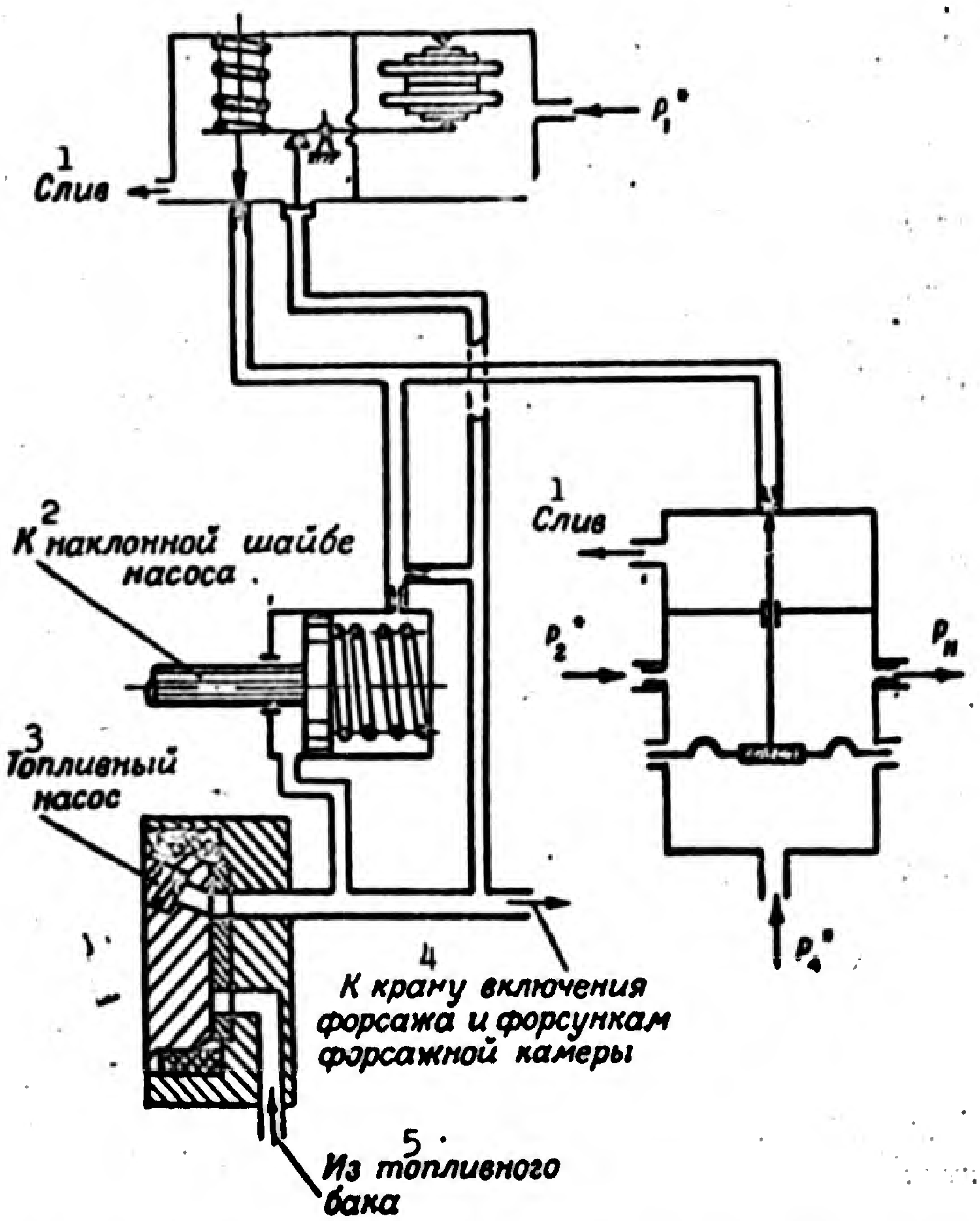

Fig. 304. Basic arrangement of closed-loop afterburner control system using p^*_2/p^*_4; the system employs an open-loop fuel feed-rate limiter using p^*_1. 1) Discharge line; 2) to pump inclined disk; 3) fuel pump; 4) to valve to turn on afterburner system and afterburner injectors; 5) from fuel tank.

In evaluating the operating characteristics of closed-loop afterburner control systems, we should remember that if fuel combustion efficiency is impaired for any reason and the temperature T^*_f is reduced, the regulator, in increasing the fuel feed rate, may so enrich the mixture as to extinguish the flame in the afterburner. This is especially probable in high-altitude flights. In order to eliminate this effect, some afterburner fuel-supply systems supplement the closed-loop control of the ratio p^*_2/p^*_4 with open-loop regulation as shown in Fig. 304. Here the open-loop regulator is adjusted for a somewhat greater fuel feed rate than is required by the condition $(p^*_2/p^*_4)_{max}$ = const, and in normal afterburner operation, the regulator is inoperative. It comes into play as a fuel feed-rate limiter when the afterburner combustion process becomes degraded.

Where a closed-loop control system of the $(p^*_2/p^*_4)_{max}$ = const type is used with a TRD afterburner system, the thrust may be throttled down when the afterburn on by reducing the rotor speed. Here in addition to the drop te at which fuel is fed to the main chambers, there is also a decrease in the rate at which fuel is fed to the afterburner to the degree required to maintain the prescribed ratio p^*_2/p^*_4 constant.

A very substantial advantage of a closed-loop afterburner control system based on p^*_2/p^*_4 is the fact that turbocompressor operation does not depend on the nozzle throat area. A variation in the latter affects only the temperature T^*_f and the rate at which fuel is supplied to the afterburner. This property of a closed-loop control system is used to control engine thrust with the afterburner on; here the nozzle flaps are the control elements: in order to reduce thrust, the flaps are closed, while they are opened to increase thrust.

Since when a TRD with afterburner is throttled by closing the flaps the rotor speed and temperature T^*_3 remain constant, and the thrust is reduced solely owing to the reduced rate at which fuel is fed to the afterburner, this throttling method is the most economical of all the methods discussed above.

Part Nine

STARTING SYSTEMS FOR GAS-TURBINE ENGINES

Chapter 26

STARTERS

1. GENERAL INFORMATION ON STARTING SYSTEMS. TYPES OF STARTERS

Rapid and reliable starting of the engine is one of the chief conditions for ensuring the combat effectiveness of an aircraft.

Starting of a GTD [gas-turbine engine] is usually taken to mean the process during which the engine goes from the stationary state (state of rest) to a state of steady independent operation.

Steady independent operation of a GTD is possible only at speeds for which the turbine power is adequate to turn the engine rotor and the temperature of the gases ahead of the turbine does not exceed the maximum permissible value.

The minimum speed for steady independent operation of a GTD under given external conditions will depend on the starting characteristics of the engine. By the starting characteristics of a GTD, we usually mean the relationships between the power N_p needed to turn the rotor, the power N_t developed by the turbine, and the speed during the starting process where the gas temperature T^*_3 ahead of the turbine follows a given law.

Figure 305 shows a sample starting characteristic for a TRD. It is clear from the figure that even where $T^*_3 = T^*_{3max}$, the minimum speed n_r for independent TRD operation is fairly high. At speeds below the equilibrium speeds ($n < n_r$), the engine cannot operate independently; it is therefore necessary to have a constant source of power (starter) in order to start a TRD; the starter must be able to turn

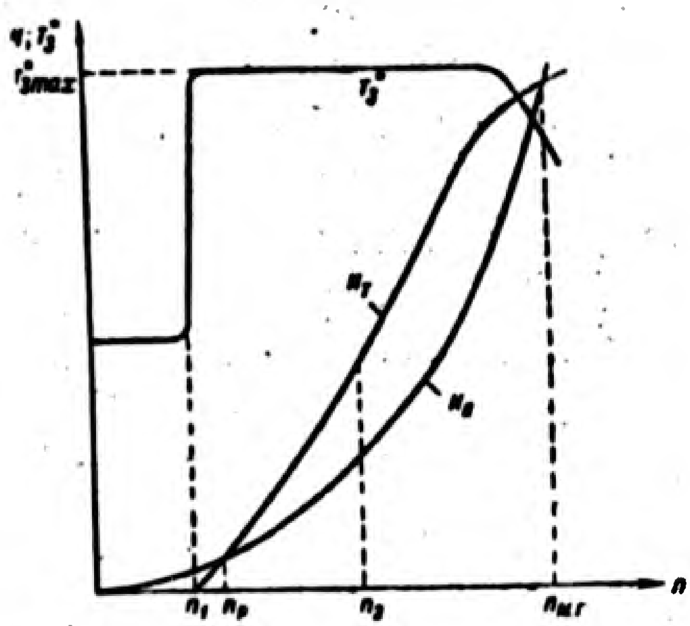

Fig. 305. Starting characteristics of TRD.

the engine rotor at least until a speed $n = n_r$ is reached. It is impossible to disconnect the starter at these speeds, since a slight impairment of engine operating conditions appearing, for example, through the action of a wind blowing from the aircraft tail, an increase in losses at the engine intake and exhaust sections, unstable compressor operation, etc. may cause the power developed by the turbine at $n = n_r$ to drop below the power needed to turn the rotor, and the engine will stall.

As a rule, the speed n_2 at which the starter cuts out is a speed such that the turbine power is sufficient to bring the engine reliably to the idle-running speed.

The idle-running speed n_{mg} is selected on the basis of the condition that will provide minimum thrust with reliable and stable engine operation. Here, in order to provide the necessary acceleration, the gas temperature ahead of the turbine should be less than the maximum permissible value.

The speeds n_r, n_2, and n_{mg}, as well as the speed n_1 at which the turbine begins to create a positive torque will depend on many factors (the type of engine, the efficiencies of compressor and turbine, the gas temperature ahead of the turbine during starting, and the ambient temperature) and will be different for different types of engine. Processing of statistical data indicates, however, that for the majority of engines of the same type the ratios of these speeds to the maximum speed n_1/n_{max}, n_r/n_{max}, etc., i.e., the relative speeds, will vary over small ranges (see the table).

1 Тип двигателя	$n_в/n_{max}$	$n_т/n_{max}$	$n_в/n_{max}$	$n_{мг}/n_{max}$
2 ТРД с осевым компрессором	0,08—0,11	0,11—0,15	0,2—0,33	0,28—0,38
3 ТРД с центробежным компрессором . .	0,06—0,09	0,08—0,11	0,13—0,15	0,2—0,23
4 ТВД с осевым компрессором	0,08—0,11	0,11—0,15	0,3—0,4	0,6—0,8

1) Type of engine; 2) TRD with axial compressor; 3) TRD with centrifugal compressor; 4) TVD [turboprop engine] with axial compressor.

In accordance with what we have said, the entire starting process may be divided into three fundamental stages (see Fig. 305).

<u>The first stage</u> extends from time $n = 0$ (the starter is connected to the engine rotor) to time $n = n_1$ (ignition of the fuel in the combustion chamber and commencement of turbine operation). In this stage, the starter alone turns the rotor.

<u>The second stage</u> occupies the period from time $n = n_1$ (commencement of turbine operation) to time $n = n_2$ (starter dropout). During this stage, the rotor is turned by the starter and engine turbine together.

<u>The third stage</u> extends from time $n = n_2$ (starter dropout) to the instant at which the engine arrives at idling speed. During this stage, the rotor is turned by the engine turbine alone.

Starting systems are provided to start GTD. The starting system should satisfy the following basic requirements:

– it must provide reliable automatic starting of the engine on the ground and in the air for all operating altitudes and flight speeds;

– it should provide independent starting with low starting-system weight and size;

– it should be simple and safe to operate and have high operating reliability;

– it should satisfy the required flying and mechanical conditions for duration of the starting period.

Starting systems for GTD include the following basic elements: the starter, ignition devices to create the initial flame site for ignition of the fuel in the combustion chambers, and fuel regulators to control the rate at which fuel is supplied to the combustion chamber during the starting process.

In addition, the starting system contains electrical systems that make the entire starting process automatic, as well as power sources for the starter.

The starter is an engine designed to turn the GTD rotor during the starting process. In accordance with the specifications for starting systems, the starter should be simple in construction, should develop the highest possible power with small size and low weight, should be safe to handle, have high operating reliability, and should provide for multiple starts.

Starters used for present-day GTD may be classified into the following basic types: electrical, turbine, and rotary.

Turbine starters are in turn classified as gasturbine, compressed-air, powder, vapor, and turborocket starters, and rotary starters as compressed-air, gas, and hydraulic starters.

In some cases, systems employing no starter are used to start low-power GTD.

2. ELECTRICAL STARTERS

An electrical starter is a direct-current electric motor powered by an on-board or airport storage battery with a rated voltage of 24-28 v.

In addition to electrical starters, electrical starter-generators are also widely employed with present-day engines.

The starter-generator is an electrical machine that operates as a starter during the starting process, and then switches to generator operation and subsequently acts as a source of direct current for the aircraft. By combining the starter and generator into one unit, we are able to reduce the weight of power-plant elements by an amount nearly equal to the weight of a starter. The required electrical characteristics of a starter generator in the starter and generator modes can be obtained by connecting into the starter-generator drive train an automatic two-speed transmission or by connecting an additional winding into the electrical circuit of the starter-generator that is automatically switched as we go from one operating mode to another. The first method is most commonly used at this time. Here the two-speed transmission automatically provides different gear ratios in the starter and generator modes. As a rule $i_{st} = (3\text{-}4)i_{gen}$, where $i_{st} = n_{st}/n_{dv}$ and $i_{gen} = n_{gen}/n_{dv}$.

In order to increase the efficiency with which the storage batteries are used, starting systems are sometimes used in which the batteries are switched from parallel to series connection (24-48 v system).

The utilization of a 24-48 v starting system increases starter power during the last stage in which the engine is turned over, but it leads to rapid discharge of the storage batteries and reduces the number of repeated starts that can be made without recharging the storage batteries.

Electrical starters are simple in operation and provide reliable engine-rotor turnover during the starting process. The electrical starting system easily makes it possible to provide a fully automatic starting cycle.

The main drawbacks to the electrical starting system are the great weight of the storage batteries, which amount to 120-180% of the weight of the starter, and their drop in capacity as the ambient temperature drops.

The weight-to-power ratios of electrical starting systems used with present-day GTD are 5-8 kg/hp, and they basically depend on the type and number of storage batteries. A reduction in the weight-to-power ratio by a reduction in the number of batteries of the same type will lead to rapid discharge of the batteries and a reduction in the number of repeated starts available.

The possibilities of an electrical starting system may be expanded by using a new type of storage battery that possesses high capacity per unit weight as compared with the lead storage batteries in current use, and also by using auxiliary on-board turbogenerator units.

3. GAS-TURBINE STARTERS

A gas-turbine starter is a small gas-turbine engine whose basic elements are: a compressor (usually centrifugal), combustion chamber, gas turbine, reduction gear, coupling to engine shaft, as well as fuel-supply and automatic-control systems. An electrical starter is used to start the starter itself. When in operation, the starter turbine develops more power than is needed to drive the compressor and other elements of the starter. This excess power is transmitted through a reduction gear to the shaft of the main engine, which is turned over. Thus, a gas-turbine starter is similar in operating principle to a turboprop engine. It differs in several structural details from usual full-size TVD designs, however, owing both to the small dimensions and specific features of starter operation.

First, the main structural elements and systems are designed on the basis of brief intermittent operation of the engine during the starting process only.

Second, the structural shapes and arrangement of elements are chosen so as to provide the maximum possible power with small size and low weight. Here efficiency is of secondary importance.

Third, gas-turbine starters use simplified automatic control systems that provide only for acceleration and operation of the starter in one working regime.

Fourth, certain types of gas-turbine starters employ an open lubrication system in which fuel (kerosene) is used as the lubricant.

Figure 306 shows the arrangement of a single-spool gas-turbine starter.

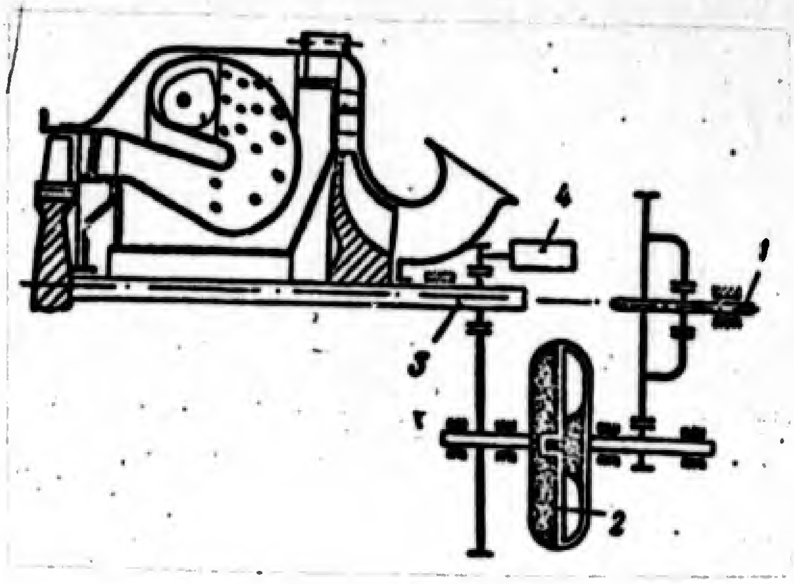

Fig. 306. Diagram of single-spool gas-turbine starter. 1) Starter drive shaft; 2) hydraulic clutch; 3) shaft of starter turbocompressor; 4) electrical starter.

A single-stage centrifugal compressor with intake on one side is installed on the starter. Such a compressor is simple in construction, inexpensive to produce, and is short. The combustion chamber is hook-shaped and provides a normal combustion process with short length. The turbocompressor rotor is doubly supported. Journal bearings are used as the supports, which helps to reduce the starter diameter.

In order to reduce the length of the starting process, provision is made for starting the starter itself and bringing it up to working speed while under no load, i.e., it is brought up to speed before the starter rotor engages the engine rotor. Measures are also taken to reduce the dynamic loading at the instant of engagement. In the arrangement shown in Fig. 306, these functions are performed by a hydraulic clutch that is mounted in the reduction-gear structure.

While the gas-turbine starter is being started and accelerated, no oil is supplied to the hydraulic clutch and the starter, disconnected from the engine, rapidly reaches its design operating speed. When the starter turbocompressor speed is close to the maximum value, a special regulator fills the hydraulic clutch with oil, so that the hydraulic clutch begins to transmit torque to the shaft of the engine to be started, turning it over.

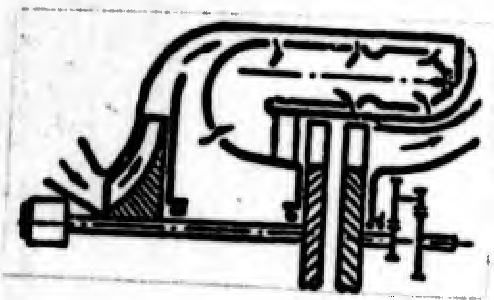

Fig. 307. Diagram of two-spool gas-turbine starter.

Two-spool gas-turbine starters have also found application (they use a free turbine). Figure 307 shows one such starter. Here the gas-turbine starter has two turbines placed one after the other; the first drives the compressor and the second turns the rotor of the engine being started. There is no mechanical connection between the turbines. Thus when the starter is started, its turbocompressor rapidly reaches working speed and then runs as a gas generator for the second turbine.

Gas-turbine starters may develop considerable power with comparatively small size and low weight of the starting system; they also easily provide autonomous starting with a nearly unlimited number of starts.

The weight-to-power ratios of modern starting systems using gas-turbine starters are 1.3-1.8 kg/hp, while for the starters themselves the ratios range from 0.6 to 1.1 kg/hp.

We should take note of the following drawbacks to starting systems using gas-turbine starters: first, the gas-turbine starter itself is complicated in construction as compared with other types of starters and, as a consequence, it is less reliable; second, the time required to start the GTD is fairly long, since in this case it also includes the time needed to start and accelerate the gas-turbine starter itself. For modern gas-turbine starters of 100-200 hp, this time amounts to 15-20 sec.

4. POWDER TURBOSTARTERS

In powder turbostarters, the turbine operates on gases generated by powders. Figure 308 shows the arrangement of a powder starter.

The powder charge is located in a thin-walled cartridge. Nitro-base powder is ordinarily used for the charge, with special added substances for flegmatization, which retard the rate of charge combustion. Powder ignited by an electrical charge is usually used for ignition. The gases evolved during combustion of the powder charge are directed against the blades 6 of the gas-turbine rotor; after passing through the turbine, the powder gases are exhausted to the atmosphere. The power developed by the turbine is transmitted through a reduction gear and clutch to the rotor of the engine to be started.

The temperature of the gases evolved by powders depend on the composition of the powder charge and reach 1600-1700°C. The weight of powder in one charge is chosen on the basis of calculations for a single charge. For repeated starting, the cartridge must be recharged. To provide for multiple starts, some powder-turbostarter designs have several cartridges or multicharge cartridges of the revolver type (with a rotating drum). A characteristic feature of powder-turbostarter operation is the fact that once the powder has started to burn, combustion cannot be stopped until it is complete. Thus if the engine to be started reaches the starter dropout speed before powder combustion has ceased, the turbine speed may exceed the maximum permissible value owing to the sharp decrease in load. In order to avoid unacceptable turbine acceleration, special regulators are installed that halt the supply of gas to the turbine when the started engine reaches the starter-dropout speed. The powder gases are then exhausted outside through special protective vents.

A basic advantage of powder turbostarters is the fact that they can create a high starting impulse that considerably reduces the time needed to start a TRD.

In addition, a starting system using a powder turbostarter is small in size and light in weight (the weight-to-power ratio of powder turbostarters is 0.1-0.15 kg/hp), and in addition, there is a considerable reduction in airport-equipment costs.

Together with these advantages, powder turbostarters have the following substantial disadvantages that prevent them from coming into wide use.

First, the powder gases cause extensive carbon deposition on the turbostarter flow-passage elements, contributing to rapid degradation of its characteristics during service.

Second, the use of powder charges as an energy source involves the danger of explosion. At low negative temperatures (below $-40^{\circ}C$), the powder charge often cracks, which during starting leads to an in-

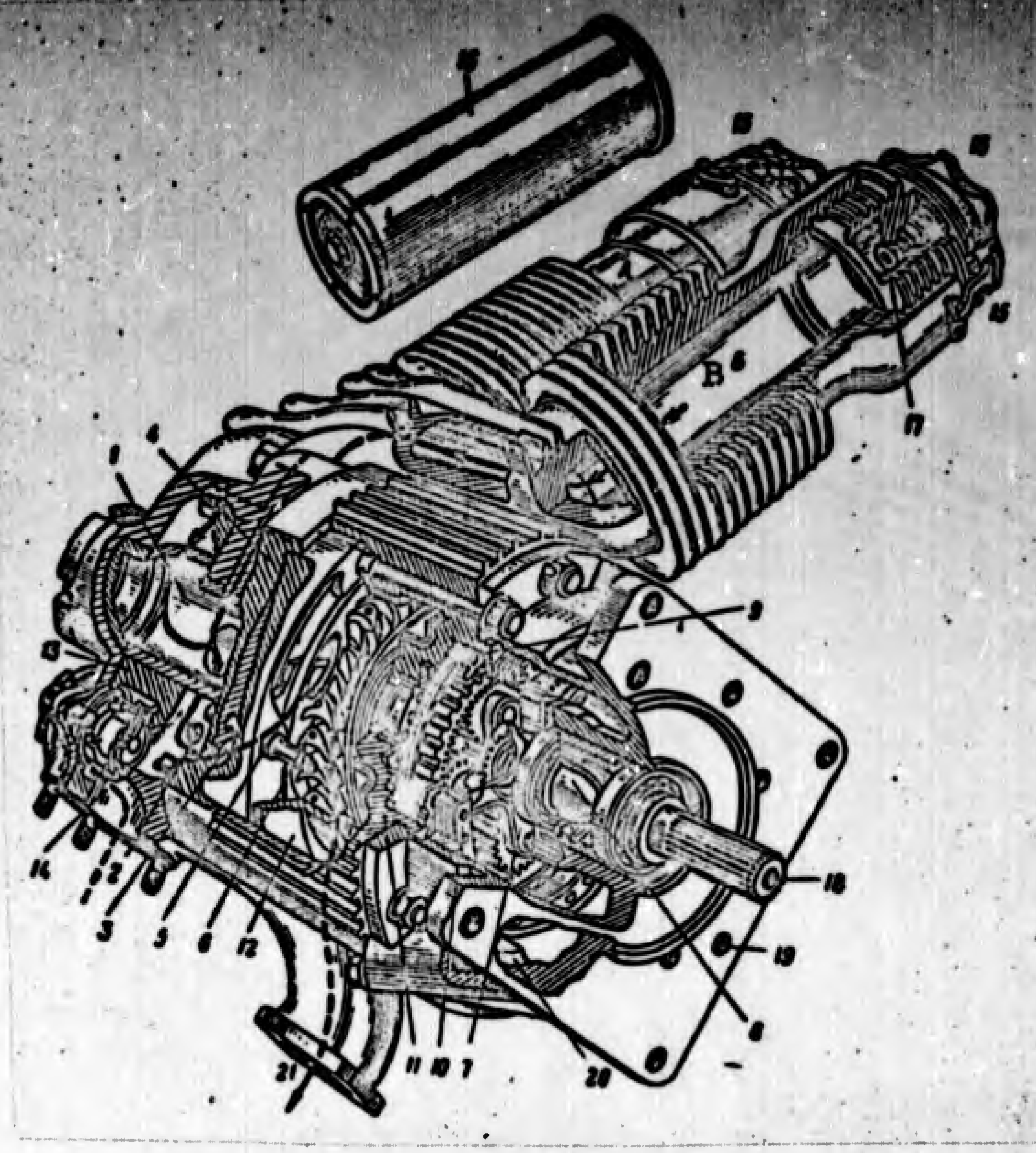

Fig. 308. Arrangement of powder turbostarter. 1) Duct; 2) powder vents; 3) holes in diaphragm; 4) diaphragm; 5) stop diaphragm with ports; 6) turbine-rotor blades; 7) reduction gear, i = 5; 8) shaft; 9) multiple-disk clutch; 10) elastic ring; 11) oil and gas seals; 12) exhaust duct; 13) disk for releasing gas at excess pressure; 14) port for discharging excess gas flow under emergency conditions; 15) lock cap with locking ratchet (to reload, it is necessary to unscrew the cap and insert the cartridge); 16) cartridge; 17) electrical contact; 18) drain vent for engine oil; 19) vent for oil supplied from engine; 20) oil-filling orifice; 21) exhaust-gas vents; A and B) cartridge cylinders.

crease in the combustion surface and may cause an explosion.

Third, as the ambient temperature decreases, the energy of the powder charge will be reduced, which reduces the available starter power, while at the same time the power required for starting rises. Thus in winter and in summer and under various climatic conditions it is necessary to use different powders and cartridges of various powers. This, naturally, complicates the operation of powder turbostarters.

Fourth, there is great difficulty in designing the starting fuel system and combustion-chamber ignition system that will provide normal combustion for 2-3 sec after starting.

5. VAPOR AND TURBOROCKET STARTERS [27]

The turbine of a vapor starter uses vapor obtained by the decomposition of hydrogen peroxide.

Figure 309 shows a vapor starter. The hydrogen peroxide (H_2O_2) is supplied from tank 2 to the vapor generator with the aid of compressed air delivered to the tank from cylinder 1. In the vapor generator, the hydrogen peroxide is decomposed in the presence of a liquid or solid

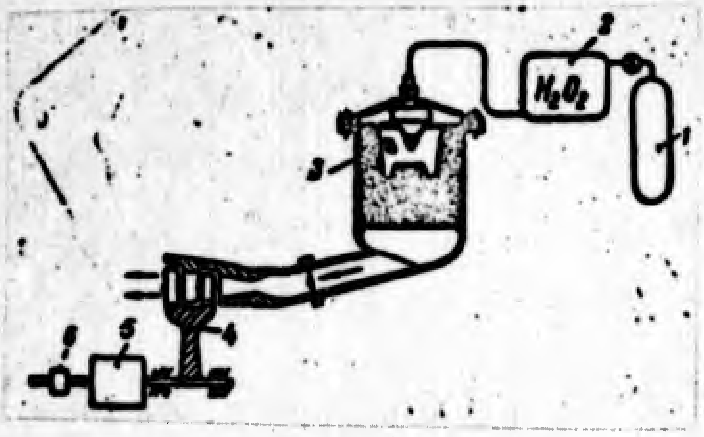

Fig. 309. Vapor starter. 1) Air cylinder; 2) tank with hydrogen peroxide; 3) vapor generator with solid catalyst; 4) turbine; 5) reduction gear; 6) clutch.

catalyst; a considerable amount of heat is then evolved. When the peroxide decomposes, a mixture of hot water vapor and oxygen is formed — a gas-vapor mixture; this is used as the working fluid for the turbine 4. The torque developed by the turbine is transmitted through reduction gear 5 and clutch 6 to the engine shaft.

A starting system using a vapor starter is extremely inconvenient to operate, since hydrogen peroxide presents the danger of explosion and has a high freezing point (−10°C at 90% concentration). In addition, where hydrogen peroxide is used as a fuel, the decomposition temperature is fairly low (400-500°C for an 80% concentration); as a result, the weight-to-power ratio of the turbine is low.

The weight-to-power ratio of the turbine may be increased and the starter size and weight thus decreased by additional combustion of a fuel in the peroxide decomposition products or by using other monopropellant fuels as the working fluids that give higher decomposition-product temperatures.

Starters in which the turbine uses the decomposition products of monopropellant fuels or the combustion products of a combustible and oxidizer are usually called turborocket starters. This is an arbitrary designation and is used since the process by which the working fluid is obtained in the starters is similar to the process occurring in ZhRD [liquid-fueled rocket engine] chambers.

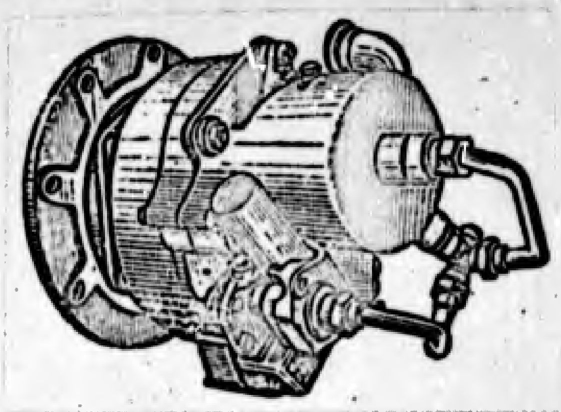

Fig. 310. Over-all view of turborocket starter.

The starting system will be simpler if a monopropellant is used. Isopropyl nitrate, hydrazine, etc. may be used as the fuel.

Figure 310 shows an over-all view of a turborocket starter using isopropyl nitrate, while Fig. 311 shows a starting system employing this starter [27].

To start the engine, an electrical circuit is closed to ignite a small powder charge in a cartridge set in drum 3, which is located at an easily accessible point. The powder gases pass through duct 11 to the starter combustion chamber. Since the pressure of the powder gases is low, the turbine will begin to turn smoothly. The gases leave the chamber through duct 12 and go to cylinder 7; the gas forces isopropyl nitrate out of the cylinder into the combustion chamber, where the heat from the powder gases causes the isopropyl nitrate to decompose

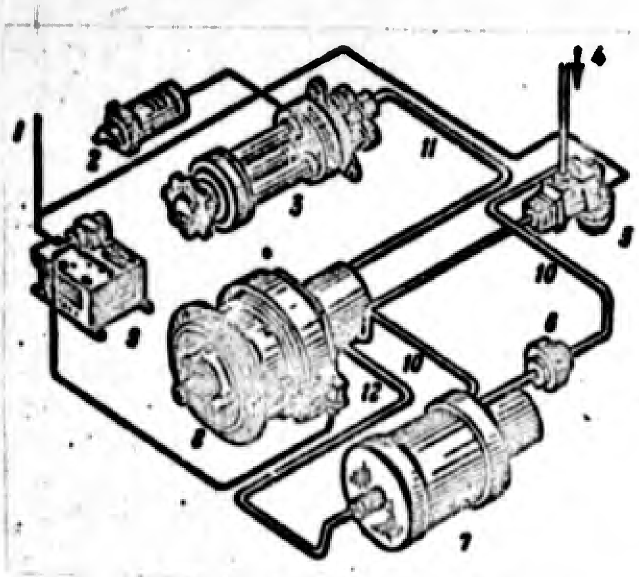

Fig. 311. Starting system using turborocket starter. 1) Power from storage battery (24 v, 0.5 amp); 2) cockpit switch; 3) drum with three cartridges; 4) inlet for fuel from tank; 5) bypass valve; 6) return valve; 7) cylinder with fuel; 8) turbostarter; 9) control box; 10) fuel lines; 11 and 12) gas-supply lines.

(burn). The chamber pressure increases, leading to an increased fuel-supply rate, which increases the speed and power of the turbine.

After starting, the piston of cylinder 7 is returned by a spring to its initial position; this pulls out a new portion of isopropyl nitrate from the tank through the return valve 6 for the next starting cycle. When the cylinder is pressed all the way over to the left, the capacity of cylinder 7 corresponds to the amount of isopropyl nitrate needed for one start cycle.

The basic advantage of turborocket starters is their ability to develop high power with small size and weight.

6. OTHER TYPES OF STARTER

Compressed-air turbostarters. In compressed-air turbostarters, the turbine uses compressed air supplied from cylinders or from a gas-turbine generator. Figure 312a shows the basic arrangement of a compressed-air turbostarter with air supplied from cylinders. The advantages of this arrangement are structural simplicity, low weight and small size of the starter itself.

A basic disadvantage to this system is the large amount of air

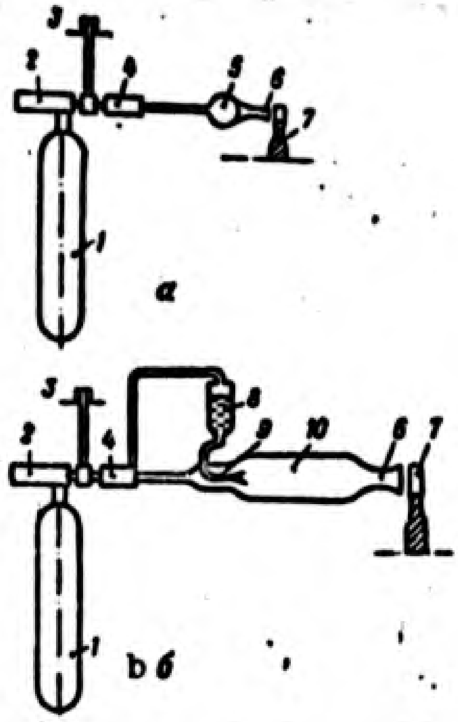

Fig. 312. Compressed-air starters. a) Without heater; b) with air heater. 1) Compressed-air cylinder; 2) reducing valve; 3) inlet for connecting airport cylinder; 4) second-stage reducing valve; 5) accumulator; 6) nozzle; 7) turbine; 8) fuel tank; 9) injector; 10) combustion chamber.

consumed, which in practical terms means the starting process is not self-contained.

This difficulty may be eased by heating the compressed air as shown in Fig. 312b. Even here, however, considerable amounts of air are needed and, consequently, the air cylinders must be large and heavy, thus complicating the design of a self-contained starting system.

Multiengined aircraft may use an auxiliary gas-turbine engine as a source of power for compressed-air turbostarters the auxiliary turbine is installed on board the aircraft. Here compressed air is taken off the auxiliary GTD (Fig. 313) and supplied through lines to the compressed-air turbines installed on the engines to be started. Where needed, compressed air may be heated in a special chamber.

It is advantageous to use this starting system in that the GTD may also be used on board the aircraft to power emergency-system units, for engine heating at very low ambient temperatures, and to meet other needs.

Compressed-air turbostarters supplied by an auxiliary GTD clearly will find their widest use on board heavy supersonic aircraft, where independent drive is used for accessory units in order to provide normal temperature conditions (an auxiliary GTD is used). Here during engine starting, the auxiliary GTD may be used as an air generator, which makes it possible to reduce the weight of the entire power plant.

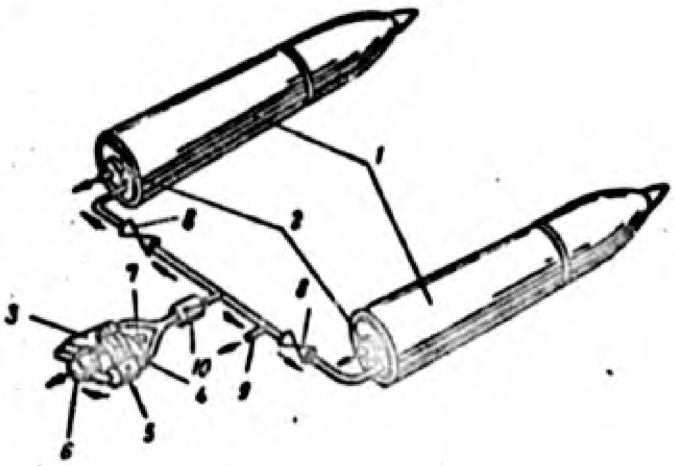

Fig. 313. Basic arrangement for starting multiengine aircraft with the aid of auxiliary GTD. 1) Aircraft GTD; 2) starter turbine; 3) auxiliary GTD; 4) compressor; 5 and 6) combustion chambers; 7) electrical starter of auxiliary GTD; 8) stop valves; 9) intake for supply of compressed air from constant source (the same intake may be used to take compressed air from the auxiliary GTD); 10) return valve.

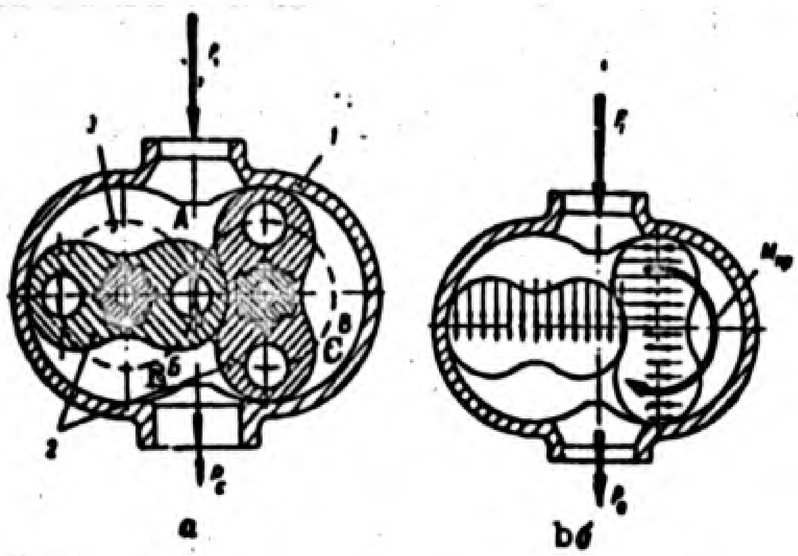

Fig. 314. Rotary compressed-air starter. a) Starter operation; b) torque generation; 1) starter housing; 2) rotors; 3) gears; A, B, and C) chambers.

Compressed-air rotary starters. In a rotary-type starter, as in a compressed-air turbine, the energy of compressed air is used to start the engine. Figure 314a shows a compressed-air rotary starter. Two rotors 2 mechanically linked by gears 3 turn within a housing 1 whose inside surfaces are precision-machined. The rotors are metal cylinders with figure-eight cross sections.

Air under pressure p_1 enters chamber A of the starter and creates a torque M_{kr} at one or both rotors, depending on their positions. Figure 314b illustrates torque generation. The air pressure causes the rotors to turn continuously, and high-pressure air moves to the low-pressure side. The gear mechanism produces a resultant torque at the starter drive shaft which is transmitted to the GTD shaft through a reduction gear and coupling mechanism.

An advantage of the compressed-air rotary starter over compressed-air turbostarters is structural simplicity and small size. In order to obtain the same power, 20-30% more air must be used than in a compressed-air turbostarter, however.

Starting systems using no starting motor. In starting systems using no starting motor, the engine rotor is turned by the engine's own turbine. As a rule, compressed air supplied through special nozzles to the turbine moving blades is used as the working fluid. Thus, for example, in the starting system shown in Fig. 315a, compressed air supplied to the moving blades of turbine 1 through nozzles 2 mounted among the blades of the main nozzle-box assembly 3.

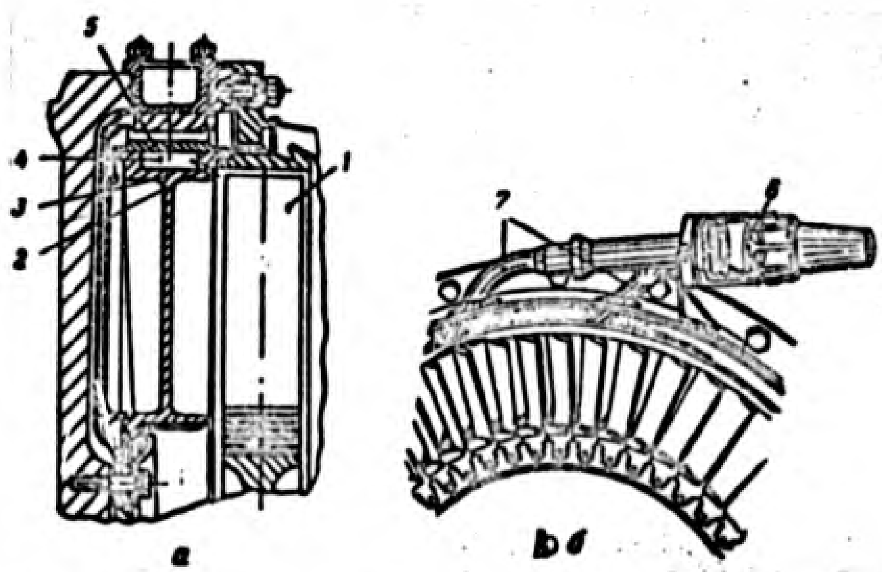

Fig. 315. Arrangement for supplying air to moving blades of GTD turbine during starting. 1) Turbine; 2) nozzles; 3) nozzle-box assembly; 4) annular channel; 5) ring; 6) return valve; 7) intakes.

Figure 315b shows another arrangement for supplying air to the moving blades. In this arrangement, air from a cylinder or other source is fed through return valve 6 to two intakes 7 mounted on the outside housing of the turbine and through oblique holes in the housing to the moving turbine blades.

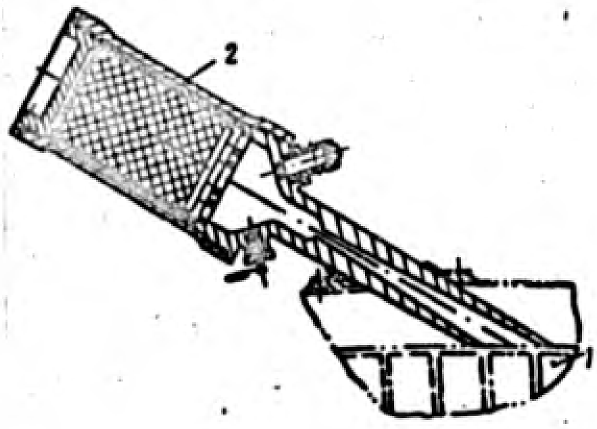

Fig. 316. Cartridge starter. 1) Blades of GTD turbine; 2) powder generator.

Structural and operating simplicity are the main advantages of a starting system that does not use a starting motor. Owing to the low efficiency of the turbine, however, the initial turnover requires a large amount of air for each start cycle. Starting systems not using starting motors are thus used only for low-power GTD at present.

The air consumption may be reduced by heating the air before it is supplied to the turbine, but this greatly complicates the design and operation of the starting system.

In addition to compressed air, powder-generated gases may be used for the working fluid. The basic arrangement for a no-motor starting system using a powder generator is shown in Fig. 316. This system has received almost no practical application.

7. REDUCTION GEARS. COUPLING MECHANISMS. TRANSMISSION STRUCTURAL ELEMENTS

Reduction gears. On the basis of acceptable efficiencies, maximum speeds of starter motors range from 25-40 thousand rpm for turbostarters, 12-14 thousand rpm for rotary starters, to 5-6 thousand rpm for electrical starters. Rotor speeds of GTD at starter dropout are 1200

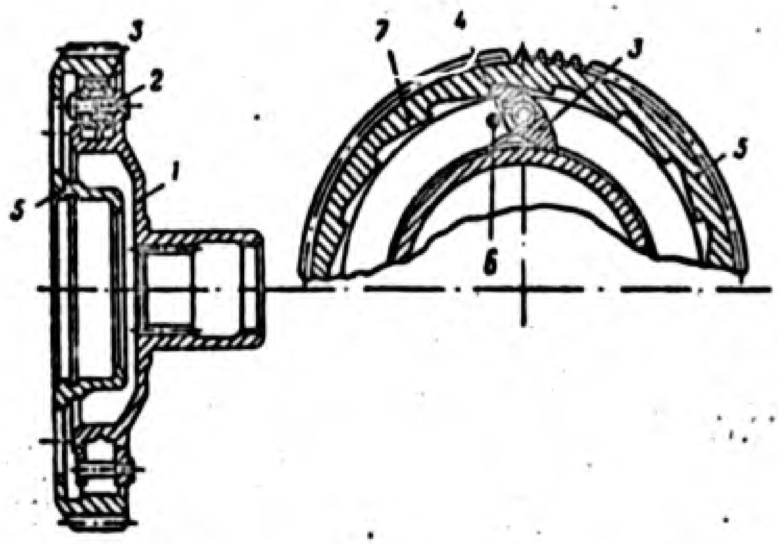

Fig. 317. Ratchet-type centrifugal clutch. 1) Driven portion of clutch; 2) dog axis; 3) dog; 4) spring; 5) driving portion of clutch; 6) stop; 7) angular tooth.

to 2000 rpm for TRD and 2000-4000 rpm for TVD. Thus a reduction gear is installed between the starter-motor shaft and the starter output shaft in order to reduce the output-shaft speed to a value below the starter-motor speed.

For turbostarters, the reduction-gear ratio is 0.04-0.15, for rotary starters, 0.10-0.30, and for electrical starters, 0.3-0.4.

For turbine and rotary starters, the reduction gear is usually made in the form of a unit attached structurally to the starter motor. In electric starters, the reduction gear may be separate and may be contained in the GTD drive box.

Types and mechanical arrangements of starter reduction gears and their basic structural elements are similar to those found in TVD reduction gears.

Coupling mechanisms. Coupling mechanisms provide a power link between the starter rotor and engine rotor during the start cycle. As a rule, ratchet and roller clutches are used as the coupling mechanism. At present two types of ratchet clutch are in wide use: centrifugal and axial clutches.

A centrifugal ratchet clutch (Fig. 317) consists of driving and driven sections. The driving section 5 of a ratchet clutch is turned by the starter and has angled (oblique) teeth 7 on the inside of its rim. The driven section 1 of a ratchet clutch, which is connected to

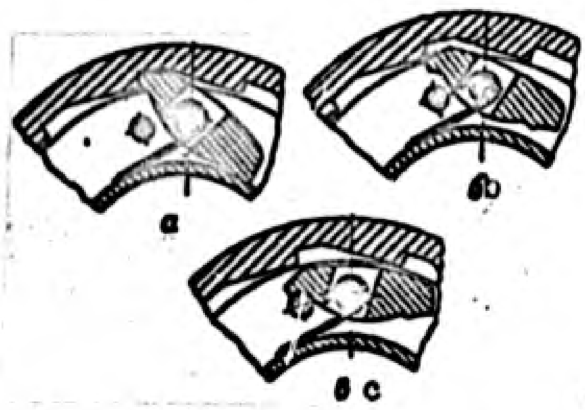

Fig. 318. Operation of a centrifugal ratchet clutch. a) Dogs engaged; b) dogs in intermediate position; c) dogs disengaged.

the engine rotor, takes the form of a housing with an annular groove in which dogs 3 are mounted on pivots 2. As a rule, three dogs are used. The dogs are free to turn on the pivots and are at all times pressed by springs 4 into a position in which the driven and driving parts of the clutches are engaged when torque is transmitted from the starter to the engine. Spring elasticity is so chosen that at a speed close to the starter-dropout speed, the heavier part of the dog lifts owing to the moment caused by centrifugal force, turning the dog about pivot 2 until it reaches stop 6 and takes up the position shown in Fig. 318c. The dogs can disengage only when the starter is off, since the peripheral force acting on the dogs when the starter is running keep them in engagement. By an appropriate choice of spring elasticity, it is possible to ensure rapid dog disengagement when the speed of the driven section becomes greater than that of the drive section. This protects the dogs against damage while they are becoming disengaged.

A basic drawback to a ratchet clutch is the possibility of large impact loads on its elements at the instant of engagement. Shock engagement can occur if the dogs occupy the position shown in Fig. 318b with respect to the teeth at the instant of starter actuation.

It is especially dangerous from the viewpoint of shock loading of ratchet-clutch parts to repeat the start cycle while the engine rotor is turning. In this case, the way in which the ratchet clutch engages will depend on the relative positions of dogs and teeth at the instant at which the speeds of the driven and driving portions of the clutch become equal. If the dogs are directly under the ratchet teeth, as shown in Fig. 318a, there will be smooth engagement with no shock loading. It is also possible for the driven and driving parts of the clutch to reach the same speed when the position shown in Fig. 318b occurs.

In the time required for the driving part of the clutch to overtake the driven part (reaching the next tooth), the speed of the starter, which is running at this instant without load, will increase rapidly, while the speed of the engine, which is turning with deceleration will drop. Thus at the instant of engagement, the difference in the angular velocities of the driving and driven parts of the clutch will cause a shock.

The case is also possible in which the starter is cut in at an engine-rotor speed such that the centrifugal force on the dogs overcomes spring elasticity and the dogs become disengaged (Fig. 318c). In this case, the starter will rapidly reach the maximum no-load speed and will turn at this speed until the starting-system control mechanism disconnects it. If during this time the engine rotor speed drops so much that the dogs begin to move, the ratchet clutch will engage with a large difference in angular velocities, i.e., with a severe clash.

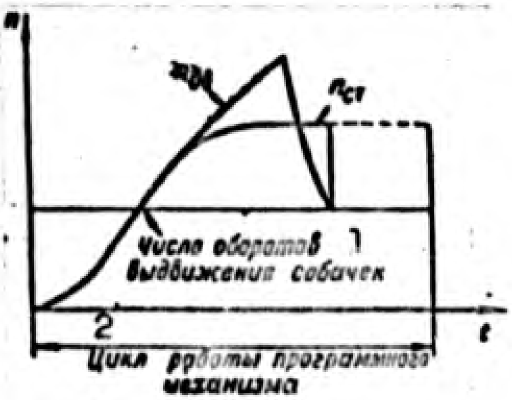

Fig. 319. Shock engagement of ratchet clutch in interrupted starting. 1) Speed at which dogs change position; 2) control-mechanism operating cycle.

Shock engagement of the starter with the engine rotor cannot be permitted, as this can lead to failure of either the clutch or the drive mechanism. Thus where a ratchet clutch is used, the engine cannot be restarted until it has come to a full stop.

The starter drive mechanism can also fail where owing to surging or an increase in gas temperature above normal values the start cycle is interrupted by a halt in fuel supply. If the ratchet clutch is already disengaged (the dogs are not engaged), and the starting-system control mechanism has as yet not completed a cycle of operation, the starter will run up to the maximum no-load speed. The engine-rotor speed will begin to drop, and at the speed at which the dogs change position (Fig. 319) the moving dogs will clash with the teeth of the clutch driving section which is turning at the no-load speed.

In order to eliminate excessive loads on drive-mechanism elements when the clutch engages, various design methods are employed. In electrical starting systems, the starting current at the instant the starter is turned on is limited by connecting an additional starting resistance into the electrical circuit. In starting systems using turbostarters, friction clutches are introduced into the drive mechanism.

In addition to the centrifugal ratchet clutch, axial ratchet clutches are also used (they use an axial ratchet wheel). The construc-

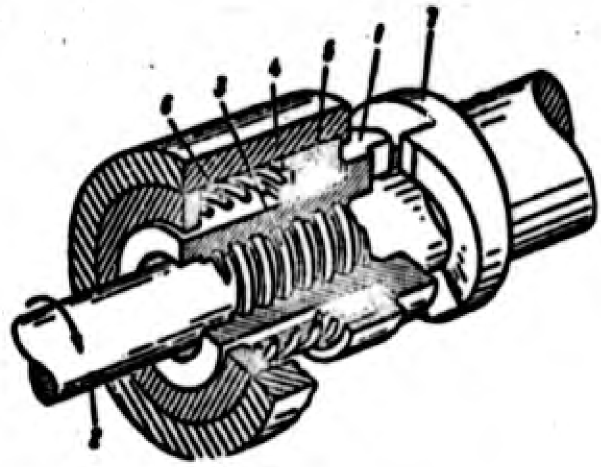

Fig. 32^. Axial ratchet clutch. 1) Drive ratchet of clutch; 2) shaft; 3) threads; 4) clutch housing; 5) felt ring; 6) spring; 7) driven ratchet clutch.

tion of such a clutch is shown in Fig. 320.

The drive ratchet 1 of the clutch is connected to shaft 2 by the triple square thread 3. Shaft 2 is turned by the starter. Felt ring 5 at all times presses spring 6 against the driving ratchet, creating a friction moment when it turns. The driven ratchet 7 is connected through a transmission to the engine rotor.

At the initial instant of starting, shaft 2 is turned by the starter. Owing to the braking effect of felt ring 5, the drive ratchet 1 advances along the triple thread and engages driven ratchet 7.

Fig. 321. Operation of roller clutch.

When the engine is operating and the starter is disconnected, the drive ratchet drops out of engagement with the angle teeth of the driven ratchet and comes to a stop.

Ratchet clutches with an axial ratchet possess the same disadvantages as centrifugal ratchet clutches, i.e., large impact loads may appear at the instant of engagement.

Roller clutches. In contrast to ratchet clutches, roller clutches

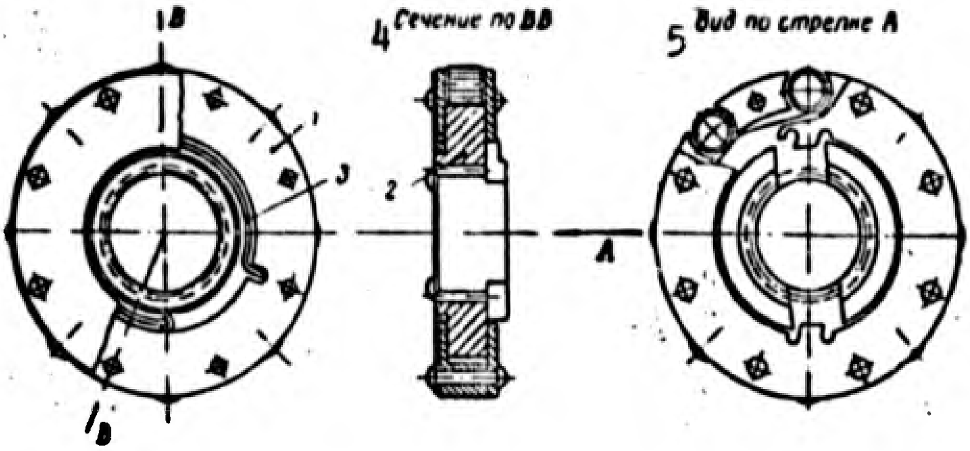

Fig. 322. Construction of roller clutch. 1) Cage; 2) clutch sprocket; 3) spring; 4) view along BB; 5) view along arrow A.

automatically ensure smooth engagement of the shafts. When there is constant contact between a roller and the working surfaces at points A and B (Fig. 321), engagement occurs gradually, as soon as the angular velocity ω_1 exceeds the angular velocity ω_2 (in the absence of slipping). Constant contact between the roller and the working surfaces may be provided by the structure through the installation of a spiral spring (Fig. 322), one end of which rests against the cage 1 and the other against sprocket wheel 2. This spring at all times tends to bend the cage with respect to the sprocket so as to jam the rollers.

The basic drawback to the roller clutch is the fact that its elements wear rapidly since they run continuously when the clutch is disengaged. Since after the start cycle has finished the starter does not operate, the driven clutch element turns at high speed relative to the drive element for the entire time that the engine is running. This leads to rapid wear of roller-clutch parts and, as a consequence, to failure.

Thus roller clutches are used as coupling mechanisms only where the clutch runs briefly when disconnected. Thus, for example, it is used in a drive mechanism from an electrical starter to a gas-turbine starter or in a two-speed transmission with a starter-generator.

Two-speed transmissions. Two-speed transmissions, as we have already said, are used in starter-generator drive mechanisms. The mechanical arrangement of one type of two-speed transmission is shown in Fig. 323. The transmission includes two free-running clutches, a ratchet clutch 1 and roller clutch 2, as well as a friction clutch 3 designed to reduce dynamic loading.

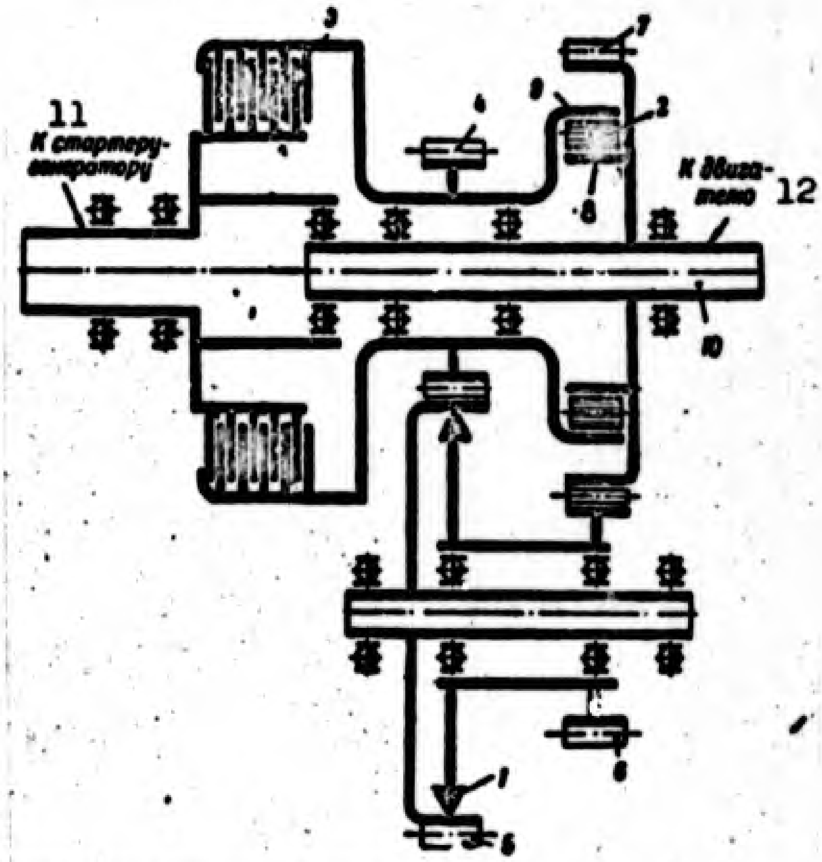

Fig. 323. Mechanical arrangement of two-speed transmission for starter-generator drive mechanism. 1) Ratchet clutch; 2) roller clutch; 3) friction clutch; 4, 5, 6, and 7) gears; 8) roller-clutch sprocket; 9) outer race of roller clutch; 10) shaft; 11) to starter-generator; 12) to engine.

In the starter mode, the torque from the starter is transmitted through friction clutch 3 and gears 4 and 5 to the drive section of ratchet clutch 1; the torque is transmitted from the drive section of the ratchet clutch through dogs 3 (see Fig. 317) to the driven section of the ratchet clutch and through gears 6 (see Fig. 323) and 7 to shaft 10, which connects the two-speed transmission to the engine gear box. Here the free-running roller clutch is disconnected, since sprocket 8, which is an integral part of gear 7, turns at a speed less than that of the outer race 9, which is connected by splines to gear 4, and the rollers are not jammed. The gear ratio in starter mode will equal

$$i_{ст}=\frac{z_7}{z_6}\cdot\frac{z_5}{z_4}\cdot i_{кп},$$

where i_{kp} is the gear ratio of the engine gear box.

When the starter is disconnected, its speed and the speed at which outer race 9 of the free-running roller clutch turns will decrease, and the speed of sprocket 8 will increase as the engine turns over owing to the appearance of excess torque at the turbine. As soon as this speed exceeds the speed of the outer race, the free-running clutch rollers will be jammed, and the rotor of the starter-generator and shaft 10 will then begin to turn at the same speed. At this instant the ratchet clutch drops out, and the starter-generator goes over to the generator mode. Torque will be transmitted from the engine to the starter-generator through the free-running roller clutch and the friction clutch.

The gear ratio in generator operation will equal the gear ratio of the engine gear box, i.e., $i_{gen} = i_{kp}$.

<u>Hydraulic clutch.</u> A hydraulic clutch consists of a drive portion, the pump 1 (Fig. 324), and a driven portion, the turbine 2; they are not rigidly linked. The hydraulic-clutch pump is driven through a gear transmission by the shaft of the starter turbocompressor, while the turbine is connected by the same transmission to the shaft of the engine to be started. The inside toroidal chamber formed between the pump and the turbine is divided by radial baffles 3, which are connected together by rings 4 (at the impeller) and 5 (at the turbine).

The hydraulic clutch is filled with oil which is circulated in each section while the clutch is in operation in the directions shown

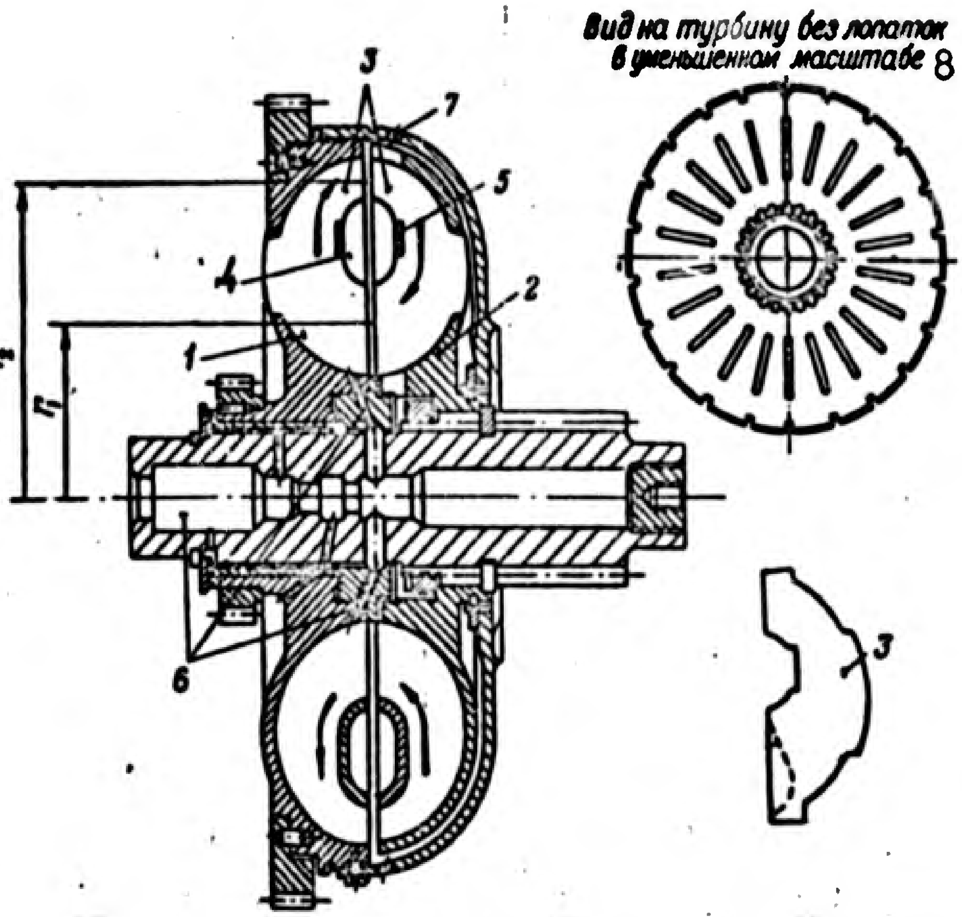

Fig. 324. Hydraulic clutch. 1) Drive section of hydraulic clutch (pump); 2) driven section of hydraulic pump (turbine); 3) radial baffles; 4 and 5) rings; 6) oil ducts; 7) oil-discharge vents; 8) view of turbine without blades, reduced scale.

by the arrows. This circulation is maintained by the difference in oil centrifugal forces in the drive (pump) and driven (turbine) sections which is produced by the difference in angular velocities. The angular velocity of pump rotation will always be greater than the angular velocity of turbine rotation.

The circulating oil acts on the radial baffles 3 of turbine 2 and produces a torque on the driven section of the hydraulic clutch equal to the torque transmitted by the hydraulic clutch to the shaft of the engine to be started.

The magnitude of torque transmitted by the clutch depends on the amount of oil circulating (on the degree to which the inside clutch chamber is filled with oil) and on the speed at which the oil circulates in the chamber, which in turn depends on the difference in the angular velocities of the pump and turbine. The more oil that circulates and the greater the difference between the angular velocities of pump and turbine, the greater will be the torque transmitted by the hydraulic clutch.

Chapter 27

IGNITION EQUIPMENT

1. STARTING IGNITERS

The majority of present-day GTD use starting igniters as ignition equipment. A starting igniter (Fig. 325) includes a spark plug 1, starting injector 2, and chamber 3. Vents 4 are drilled through the chamber walls. Air is supplied through these vents to the starting-fuel combustion zone. The length of the chamber and the number of vents is so chosen that the fuel supplied to the starting igniter burns partly within the igniter chamber and partly in the main combustion chamber. As a result, a flame is formed that ensures reliable ignition of the main fuel. The number of igniters installed on an engine depends on the engine dimensions, type of combustion chamber, and usually equals 2 to 8.

In can-type and cannular combustion chambers, the flame is conducted from one flame tube to another through connecting ducts. Two igniters are usually installed in annular chambers, and to improve propagation of the flame throughout the entire chamber, additional starting injectors are sometimes provided. The point of installation of a starting igniter in the combustion chamber is so chosen that the flame reaches the flow reversal zones and heats the fresh mixture throughout the entire stabilization zone.

In order to ensure stable combustion of the starting fuel, various types of stabilizing devices are installed in the igniter chambers.

Figure 326 shows a starting igniter that has a stabilizing device

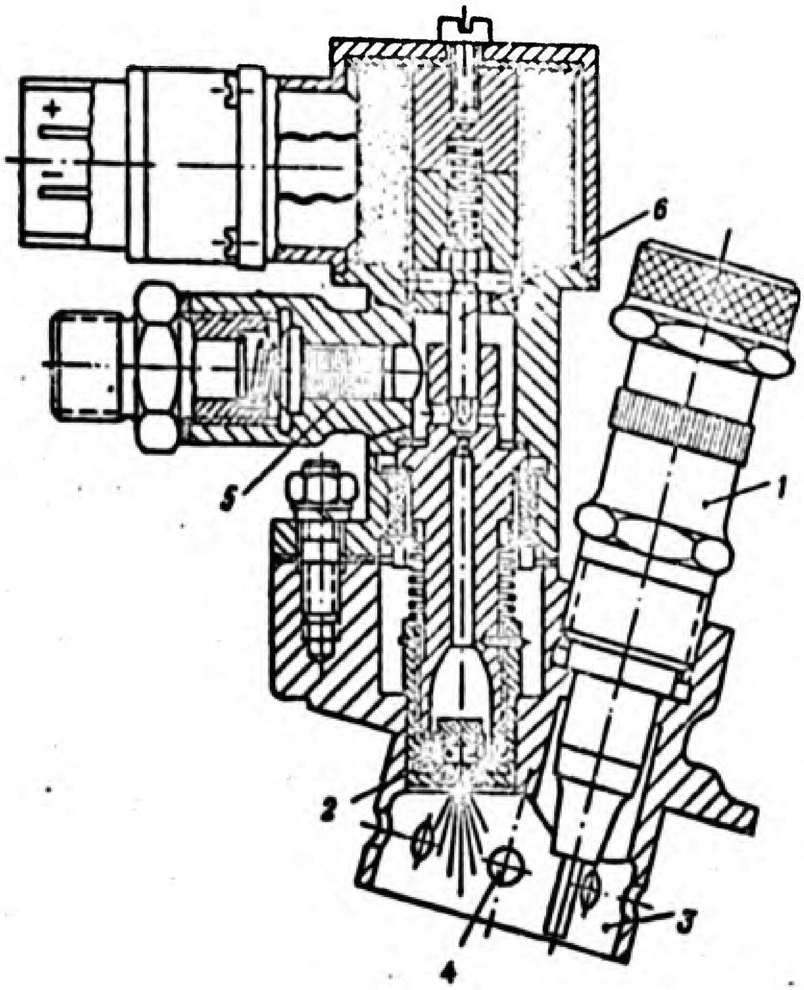

Fig. 325. Starting igniter. 1) Spark plug; 2) starting injector; 3) chamber; 4) air vents; 5) filter; 6) electromagnetic valve.

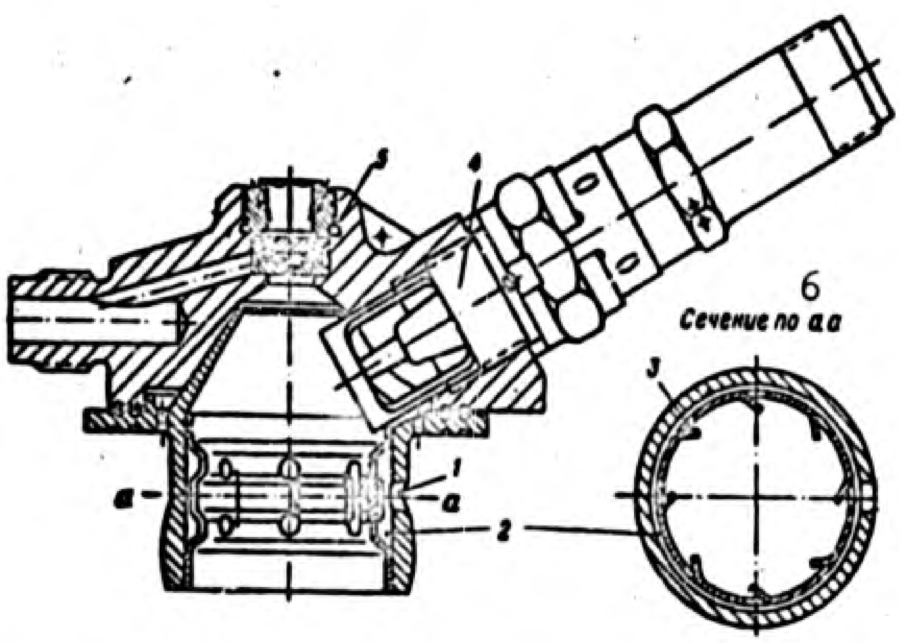

Fig. 326. Starting igniter with swirler. 1) Air vents; 2) annular chamber; 3) tangential slots; 4) spark plug; 5) starting injector; 6) section through aa.

in the form of a swirler. As we can see from this figure, air passes from the combustion chamber through vents 1 in the housing to the annular chamber 2 formed by the housing and the swirler. From the annular chamber, the air enters the igniter chamber through tangential slots 3 (see the section through aa), cut in the swirler. The resulting eddying of the air helps to improve mixture formation in the igniter chamber and to provide stable ignition of the starting fuel. Owing to the low rate of motion of the air through the swirler slots, however, the resulting vortex strength is inadequate to produce good flame stabilization.

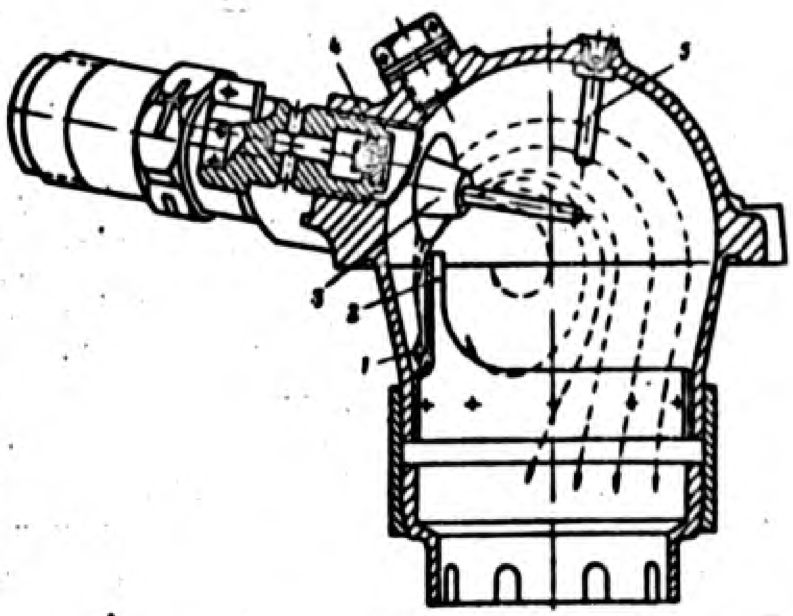

Fig. 327. Starting igniter with baffle-type stabilizing device. 1) Air vents; 2) deflector (baffle); 3) spark plug; 4) starting injector; 5) spark conductor.

Starting igniters with a baffle-type stabilizing device are frequently used in engines. Figure 327 shows such an igniter. Air from the combustion chamber enters the igniter through vents 1 in the housing and is directed by deflector 2, made in the form of a baffle, to the ignition zone, as shown by the arrows in Fig. 327. This motion of the air provides good conditions for mixture formation and, in addition, creates a flow-reversal zone that ensures a constant supply of gas to the zone in which the starting fuel is mixed with the air.

It is also possible to design a starting igniter with a stabilizing device that takes the form of a combination of a baffle-type stabilizer and a swirler. Such an igniter design provides better combustion stabilization for the starting fuel than the igniters described above.

Starting igniters either use the same fuel as the main engine, i.e., kerosene, or else they use gasoline. In the latter case, the greater volatility of the gasoline improves the starting characteristics of the igniter, especially at low ambient temperatures and high altitudes.

Fuel is supplied to the starting injectors with the aid of the starting fuel system (see Fig. 200). It includes the starting-fuel tank, usually installed aboard the aircraft, the starting-fuel pump 20, driven by an electric motor, a starting-fuel manifold with starting injectors 2 and return valve 1.

The starting-fuel system operates only during a start cycle. When the engine is started, at the same time as the ignition is turned on, the starting-fuel-pump electric motor is turned on. Fuel is delivered by the pump through the return valve to the igniter injectors. After ignition has occurred in the main chambers, the supply of starting fuel is cut off together with the power to the spark plugs.

The return valve closes the fuel line, eliminating the possibility of fuel flowing out of the fuel tank through gaps in the gear pump to the starting manifold. While the engine is in operation, this valve prevents compressed air from the combustion chamber from reaching the starting-fuel line.

The normal working fuel pressure ahead of the starting injectors is 1.6-2.2 kg/cm^2, and it is maintained within this range by a reducing valve installed on the fuel pump.

Where the igniter uses kerosene, the starting pump takes fuel directly from the line that supplies the main pumps.

Starting igniters provide reliable ignition for combustion chambers when the engine is started on the ground and in flight. As the start cycle is initiated at higher altitudes, the starting characteristics become poorer. This is explained by the fact that the formation and development of the starting flame in the starting igniter depends on the temperature, pressure, and velocity of the air in the combustion chamber, as well as on the strength of the spark discharge.

As we know, an inoperative engine will rotate by itself in flight. Here the air pressure and temperature in the combustion chamber will differ little from the pressure and temperature of the outside air, while the velocity of the air in the chamber will be considerably greater than the velocity observed during test-stand starts. As a consequence, as the altitude at which a start is attempted increases, the operating conditions of the igniter become worse. The low air pressure impairs mixture-formation conditions, while the low temperature and high stream velocity increase the amount of heat taken from the starting flame. In addition, owing to the reduced air density, the energy of the spark discharge decreases. Thus, as the altitude goes up, the amount of heat obtained in the starting flame is reduced. And if we take account of the fact that conditions for stable combustion in the main chamber also are degraded as the altitude increases, we see that starting of an engine at altitudes exceeding 8-10 km is an unreliable operation.

A marked improvement in the high-altitude performance of a starting igniter and, therefore, a considerable increase in the reliable-starting altitude may be gained by supplying additional oxygen to the fuel-air mixture in the combustion chamber [27].

An improvement in the high-altitude performance of a starting igniter is also gained if we increase the spark power.

2. OTHER TYPES OF IGNITION DEVICES

In addition to starting igniters, other types of igniting devices are used for the main combustion chambers. Thus, for example, the fuel is frequently ignited from a spark plug installed directly in the main combustion chamber.

In this case, reliable fuel ignition during a high-altitude start may be provided by the use of an ignition system with a high-power spark.

Chapter 28

AUTOMATIC STARTING-FUEL SUPPLY SYSTEMS

1. THE NEED FOR AUTOMATIC FUEL SUPPLY DURING STARTING

The fuel feed rate required during engine starting is limited by the maximum permissible gas temperatures, as well as by the conditions that must be satisfied to ensure reliable ignition and stable combustion of the fuel. The latter is especially important in a high-altitude start, where the chamber mixture-composition tolerance range narrows sharply and a mixture that is slightly too rich will cause the flame to go out.

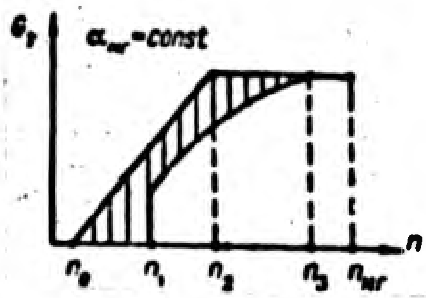

Fig. 328. Fuel feed rate during engine starting.

The optimum maximum permissible flow rate and the way in which it changes with rotor speed depend on the starting characteristics of the engine. Figure 328 shows a sample curve (below) for the maximum permissible fuel feed rate when an engine is started. The speed n_1 corresponds to the beginning of fuel supply.

If the engine control system is provided with a fuel flow-rate control and the throttle valve is set to the idling position when the engine is started, in the absence of a special automatic device, the actual supply of fuel during the starting process (top curve) will begin at a speed n_0 at which the fuel pressure developed by the pump opens the distributor valve installed ahead of the injectors. At speed n_2, the constant flow-rate control comes into play.

By comparing the two curves, we see that in order to ensure normal starting it is necessary to eliminate the excess supply of fuel indicated by the hatched area. The excess fuel supply may be eliminated manually or automatically. As a rule, with manual control the throttle valve or stop cock will begin to be opened smoothly only when the rotor speed reaches a value near n_1. With manual control, however, it is difficult to ensure reliable starting and it is even more difficult to bring the engine up to idling speed rapidly. This shows that it is necessary to use an automatic fuel supply system in starting.

2. AUTOMATIC STARTING-FUEL CONTROLS (TAZ)

In designing automatic devices to control the fuel feed rate in starting, it is usual to employ either variation of the fuel feed rate as a function of rotor speed or variation of the fuel feed rate in proportion to the air flow rate. It is clear that in the latter case, the automatic device will change the fuel feed rate not only with rotor speed but also with external conditions (p^*_N and T^*_N) thus ensuring that a given gas temperature ahead of the turbine will be maintained during the start cycle when external conditions vary. This is a fundamental advantage of automatic devices that control the fuel feed rate in proportion to the air flow rate over automatic devices controlling the fuel feed rate on the basis of rotor speed; this is why the former type of control is widely used on engines.

The air pressure behind the compressor or following one of its stages may be used as a measure of the air flow rate with an accuracy sufficient for the purpose at hand. This fact is also usually employed when automatic devices are designed to control the fuel flow rate during starting, i.e., when automatic fuel starting controls (TAZ) are designed.

The basic arrangement of a TAZ with a sensing element that measures the air pressure behind the compressor is shown in Fig. 329.

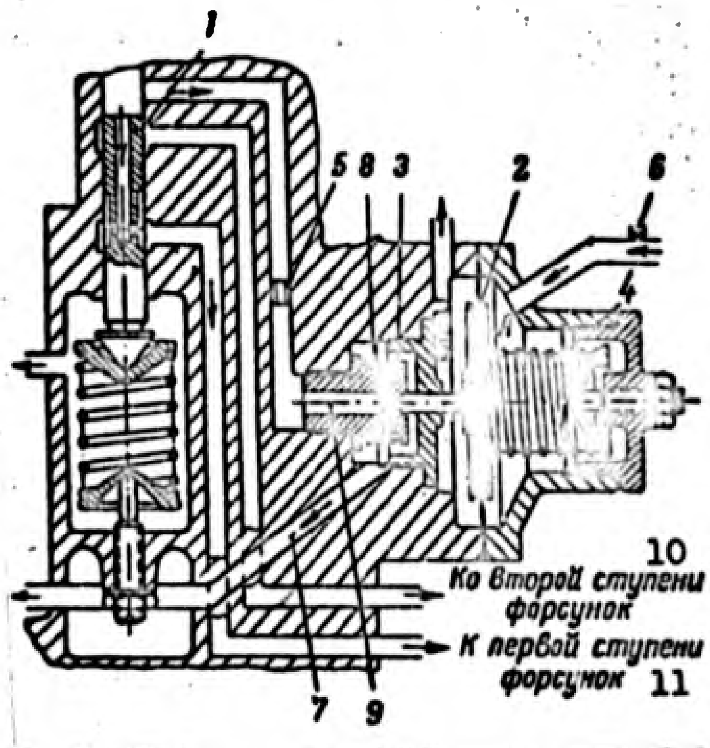

Fig. 329. Basic arrangement of automatic starting-fuel control. 1) Distributor valve; 2) diaphragm; 3) valve; 4 and 8) springs; 5) fuel nozzle; 6) scavenging nozzle; 7 and 9) valves; A and B) chambers; 10) to second-stage injector; 11) to first-stage injector.

The automatic starting-fuel control includes a valve 3 connected to a diaphragm 2, a fuel nozzle 5, scavenging nozzle 6, and spring 4. The diaphragm separates chambers A and B. Chamber A receives the pressure of the air after the compressor, and chamber B communicates with the atmosphere. On one side, valve 3 is acted on by the forces due to the fuel pressure in the duct ahead of distributor valve 1 and the force due to spring 8, and on the other side by the force due to spring 4 and the force that appears owing to the differential air pressure across the diaphragm, which is proportional to the air pressure p_2 behind the compressor.

In all steady engine operating regimes, the force due to spring 4 together with the force that appears at the diaphragm owing to the difference in pressures on its two sides will be greater than the sum of the force due to spring 8 and the force due to the fuel pressure on valve 3. Thus the latter will completely close duct 9. When the engine is not running, duct 9 will also be completely closed, since the force due to spring 4 is greater than that due to spring 8.

When the engine is started, the fuel pressure opens valve 3, which causes fuel to be transferred from the high-pressure line to the pump intake through duct 7, thus limiting the rate at which fuel is fed through distributor valve 1 to the combustion chamber.

As the engine-rotor speed increases and the air pressure behind the compressor rises, valve 3 will close down duct 9, thus decreasing the amount of fuel transferred and, consequently, increasing the rate at which fuel is fed to the engine. At a speed somewhat below idling speed, the air pressure in chamber A will rise to the point that valve 3 closes altogether and the TAZ drops out of service.

By a suitable choice of the dimensions of nozzle 6, valve 3, and diaphragm 2, as well as of the stiffnesses of springs 4 and 8, it is possible to provide a TAZ performance characteristic that will match the required characteristic very well.

In order to limit the amount of fuel bypassed during high-altitude starts, where the air pressure behind the compressor is low, a nozzle 5 is installed ahead of valve 3; its flow area is chosen so as to provide high-altitude engine starting.

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