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MAIN MARINE GAS TURBINE ENGINES: STRUCTURAL FEATURES OF THEIR COMPONENTS

MINISTRY OF EDUCATION AND SCIENCE OF UKRAINE

Admiral Makarov National University of Shipbuilding

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Textbook

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A U T H O R S :
SERHIY I. SERBIN – D.Sc. (Engineering), Professor, Admiral Makarov National University of Shipbuilding, Ukraine;
XIANRUI ZHAO – Ph.D., Professor, Jiangsu Maritime Institute, China;
VOLODYMYR M. PATLAICHUK – Ph.D. (Engineering), Associate Professor, Admiral Makarov National University of Shipbuilding, Ukraine

R E V I E W E R S :
I. V. Bilousov – D.Sc. (Engineering), Professor, Professor of the Department of Ship Technical Systems and Complexes, Odesa National Maritime University;
M. R. Tkach – D.Sc. (Engineering), Professor, Head of the Department of Engineering Mechanics and Manufacturing Technology, Admiral Makarov National University of Shipbuilding;
O. K. Cherednichenko – D.Sc. (Engineering), Professor, Professor of the Department of Marine Power Plant Operation and Thermal Engineering, Admiral Makarov National University of Shipbuilding

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The textbook examines the design, operating principles, and typical configurations of marine gas turbine units used in maritime transport. It analyzes the structural features and applications of the principal components of marine gas turbine units, including compressor and turbine flow-path components, axial turbomachine rotors, bearings and seals, as well as gas turbine casings. An overview of the designs currently employed in marine propulsion systems is also presented. The publication is intended for students majoring in G11 Mechanical Engineering, as well as for students enrolled in the Marine Power Engineering Technology educational programme.

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1. GENERAL STRUCTURE OF MARINE GAS TURBINE ENGINES

1.1. Marine application of gas turbines

Immediately after the end of the Second World War, many manufacturers of aviation gas turbines began work on the creation of gas turbine power plants for ships of the Navy. For this, mass-produced designs of turboprop and turbojet aircraft engines were used as base engines, which underwent special refinement (conversion). The first ship in the world to be equipped with a gas turbine propulsion power plant was the Motor Gun Boat MGB 2009 (Great Britain), on which in 1947 it was successfully tested as a 1.86 MW G.1 Gatric acceleration engine of the Metropolitan-Vickers Electrical Co. Ltd. (Fig. 1.1) [14]. (The gas turbine engine replaced one of the three diesels (central) of the base MGB 509 three-screw boat.)

Fig. 1.1. MGB 2009 gunboat (Great Britain)
(displacement – 95 tons; maximum speed – 31 knots)

The use of gas turbine engines in the navy was due to their positive properties, such as:

1. Small weight and size.

For water tonnage vessels with the same power, the mass of a power plant with a simple cycle gas turbine is 10-16 times less than the mass of a plant with a low-speed diesel engine. For the same generated power, a marine plant with simple cycle gas turbines occupies an area approximately 12 times smaller than a power plant with a low-speed engine [3].

2. High maneuverability.

The duration of preparation for the start-up of steam turbine power plants of transport vessels is at least 4 hours. Warm-up before starting heavy low-speed diesel engines requires 2.0-2.5 hours. Start-up of a simple-cycle gas turbine is performed without preliminary warm-up. The time of preparation for start-up in this case is 5-10 minutes [3].

The time required to gain full power from idle is approximately 1.0-1.5 hours for steam turbine plants and low-speed engines, about 30 minutes for medium-speed engines, 3-5 minutes for a simple cycle gas turbine and high-speed engines [17].

3. Low installation and maintenance costs.

The compactness and low weight of simple cycle gas turbines allow them to be fully assembled and tested at the turbine plant with subsequent delivery as a single module to the destination by rail, road, or air transport. According to [3], the cost of installing a ship-based gas turbine power plant is five times less than the cost of installing a diesel power plant.

4. Low lubrication costs (compared to diesel engines).

With the same power and operating time, lubrication costs in a gas turbine power plant are 3-7 times lower than in a low-speed diesel engine, and 20-30 times lower than in a high-speed engine.

5. Lower concentration of harmful substances in exhaust gases.

Compared to other types of power plants, the concentration of harmful substances in the exhaust gases of gas turbine plants is significantly lower. This is explained by the continuity of working processes over time, the use of light fuel grades, and a significant excess coefficient of air pumped through the engine.

6. Lower vibration level (compared to diesel engines).

Modern gas turbines are characterized by a constant combustion mode over time; they do not have reciprocating moving masses, which ensures smooth operation of gas turbines in any operating mode.

* * *

The development of naval gas turbine engineering in the second half of the 20th century led to the fact that currently the gas turbine engine is part of the power plants of a significant number of surface naval ships (patrol and missile boats, patrol and anti-submarine ships, frigates, corvettes, destroyers, cruisers, light aircraft carriers, hydrofoils and hovercraft), successfully competing in this area with diesel engines. Gas turbine engines are used in the navies of 40 countries in Europe, Asia, and America. According to data for 2026, more than 400 naval ships of this type were in operation.

Figures 1.2 and 1.3 present images of various types of naval ships equipped with power plants with gas turbine engines.

In the 1950s, the first gas turbine power plants appeared on commercial vessels.

From October 1951 to 1960, the tanker Auris of the English company Anglo-Saxon Petroleum Co. Ltd. underwent experimental operation, equipped first with a gas turbine generator with a capacity of 955 kW (with exhaust gas heat regeneration; in addition to it, three more diesel generators were operating), and after replacing the power plant in 1956, with a gas turbine generator with a capacity of 3955 kW (a completely gas turbine plant). Both versions of the power plant were manufactured by the British Thomson-Houston Co.

Fig. 1.2. Arleigh Burke-class guided missile destroyer (USA)
(4 LM2500 engines; full displacement – 8315 tons, full speed – 30 knots)

Fig. 1.3. Project 1232.2 Zubr hovercraft (Ukraine)
(5 UGT 6000 engines; full displacement – 555 tons, full speed – 63 knots)

In the 1950s, the first gas turbine power plants appeared on commercial vessels.

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In 1956, in connection with the US Maritime Administration's campaign to select the most suitable type of power drive for transport vessels, General Electric Co. installed a 4.4 MW gas turbine on the USS John Sargeant (one of four Liberty-class vessels designated for this purpose).

At that time, gas turbine units with free-piston gas generators designed by R. Pescara, which were manufactured by the French company S.I.G.M.A. and were also produced under license by some American and European companies, were more widely used in the commercial fleet. By 1963, 47 transport vessels were equipped with these units.

The 1970s were marked by a fairly high rate of construction of gas turbine ships. According to data for 1978, 24 commercial ships with a turbocompressor-type gas turbine power plant were in operation. Ship gas turbine plants for tankers, gas carriers, container ships, ferries, and ore carriers were mass-produced during this period by the companies Zorya-Mashproekt (Ukraine), General Electric Co. (USA), Pratt & Whitney Corp. (USA), General Motors Corp. (Allison Division) (USA), and others.

However, starting from the second half of the 1980s, after a series of energy crises and a significant increase in oil prices, the use of gas turbines (as engines less economical compared to low-speed diesel engines) on commercial vessels began to decline. Currently, these power plants are equipped mainly with high-speed cargo and passenger ferries, as well as cruise liners (Figs. 1.4 and 1.5). The scheme of a combined diesel-gas turbine plant with electric power transmission from both engines to the propellers – *Integrated Electric Propulsion* (*IEP*) is becoming popular. The scheme of a gas turbine plant with additional power in a steam turbine by utilizing exhaust gas heat and electric propulsion – *COGES* is also finding some application.

Fig. 1.4. Container ships of project 1609 Atlantic (Ukraine)

(2 UGT 16000 engines with heat recovery; full displacement – 35,000 tons, full speed – 25 knots)

Fig. 1.5. Millennium-class cruise ship

(2 LM2500+ engines with heat recovery; full displacement – 90,238 tons, speed – 21 knots, crew – 920 people, passenger capacity – 1,950 people)

The leading positions in the field of naval and commercial ship gas turbine engineering are currently held by the companies GE Aerospace (USA), Rolls-Royce Holdings plc (Great Britain), and Zorya-Mashproekt (Ukraine). Engines created by Vericor Power Systems Inc. (USA), Mitsubishi Power Ltd. (Japan), and Siemens Energy AG (Germany) also find some periodic use.

Currently, about 1% (in value terms) of all gas turbine engines manufactured in the world are intended for naval and commercial ship applications [12; 13].

1.2. Structure and operation of simple cycle gas turbine units

A distinctive feature of gas turbine units is the presence in their composition of a main source of mechanical energy, which operates on a gaseous (except for water vapor[1]) working fluid. In the vast majority of cases, such a working fluid is the combustion products of hydrocarbon fuel, although in units made according to a closed thermodynamic cycle, inert gases (helium, argon), nitrogen, carbon dioxide, and other substances can also be used.

The structural scheme of the simplest gas turbine unit is shown in Fig. 1.6, *a*.

The first main element of a gas turbine unit in terms of the movement of the working fluid is the **compressor**. The purpose of the compressor is to increase the pressure of the working fluid (atmospheric air)[2]. In modern units, the air pressure at the compressor outlet exceeds atmospheric pressure by 7-35 times [5].

After the compressor, the air enters the second main element of the gas turbine unit – **the combustion chamber**. The purpose of the combustion chamber is to heat the working fluid. As a rule, this process is carried out by burning hydrocarbon (liquid or gaseous) fuel directly in the flow of air moving through the

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[1] The name "steam turbines" was adopted for turbines powered by steam.

[2] It is possible to create a gas turbine unit without a compressor.

combustion chamber at a practically constant pressure. The temperature of the flue gases formed in modern gas turbine units is 1300-1770 K [5].

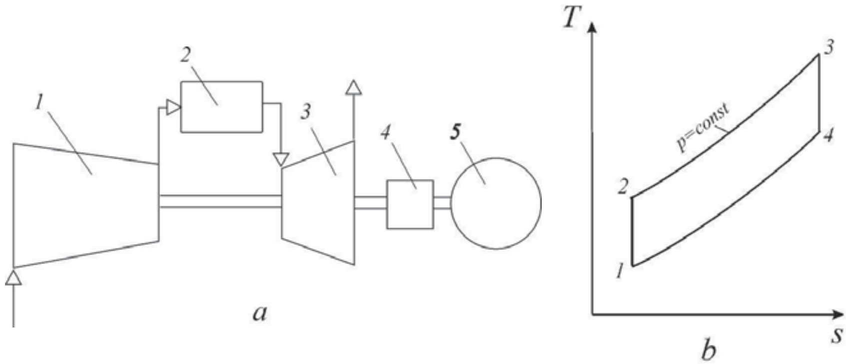


Fig. 1.6. Structural scheme of the simplest gas turbine unit (*a*) and the corresponding ideal cycle on the heat diagram (*b*):

1 – compressor; 2 – combustion chamber; 3 – turbine; 4 – gearbox; 5 – consumer

After the combustion chamber, the working fluid, which already has a certain reserve of potential energy (due to the increase in pressure and temperature), enters **the turbine**. The purpose of the turbine is to generate mechanical rotational energy.

Part of the power generated by the turbine is spent on rotating the compressor of the gas turbine unit itself, and the remaining part (**useful power**) is given to the consumer. The power consumed by the compressor is relatively large, and in simple schemes at a moderate temperature of the working fluid, can exceed the useful power of the unit by 1.5-2.0 times.

After working in the turbine, the working fluid is released into the environment (atmosphere). The temperature of the gases at the turbine outlet in modern gas turbine units is 650-900 K [5].

The ideal cycle corresponding to the simplest gas turbine unit is shown on the heat diagram in Fig. 1.6, *b*.

The process between points 1 and 2 takes place in the compressor. Under ideal conditions (excluding compression losses), it occurs at a constant entropy value, i.e., it is **isentropic**, and is depicted as a vertical line on the heat diagram.

The process between points 2 and 3 takes place in the combustion chamber. For the vast majority of modern gas turbine units, it is implemented in combustion chambers that are constantly open to the moving air flow, which means that the pressure in them is maintained practically unchanged (in ideal conditions that do not take into account aerodynamic losses, it is absolutely unchanged). Such a process is called **isobaric** in the literature.

The expansion in the turbine (process 3-4) in an ideal cycle is also given as a vertical segment, i.e., it is **isentropic**.

The process between points 4 and 1 is called the **conditional cooling process, which closes the cycle**. From a thermodynamic point of view, after the turbine, the working fluid must be cooled and only then directed back to the compressor inlet. However, in most modern gas turbine units, this cooling process is carried out by mass exchange with the environment, i.e., the hot gases exhausted in the turbine are discharged into the atmosphere, and clean cold air is continuously supplied from the atmosphere to the compressor at this time.

The thermodynamic cycle of a gas turbine unit with heat supply at constant pressure has become known in the literature as **the Brayton cycle**. This cycle was used in an automobile internal combustion engine (piston), patented and built in 1872 by engineer George Brayton (Boston, USA).

To denote the totality of the compressor part of the unit, its combustion chamber, and turbine, we will use the term **gas turbine engine**. The totality of the gas turbine unit and the complete complex of systems, devices, mechanisms, and machines serving it will be designated as **a gas turbine power plant**.

The speed of the rotor of a gas turbine engine is several thousand revolutions per minute, so the transfer of useful power to the consumer can in some cases (especially in ship power plants) be carried out through a speed-reducing device – **a gearbox**.

Figures 1.7 and 1.8 show the world's most common marine gas turbine engine, the LM2500, with a capacity of 24 MW, manufactured by GE Aerospace (USA).

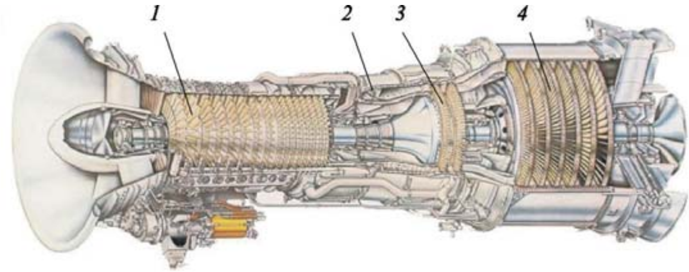


Fig. 1.7. GE Aerospace LM2500 engine [13]:

1 – 16-stage axial compressor; 2 – annular combustion chamber; 3 – two-stage axial compressor turbine; 4 – 6-stage axial power turbine

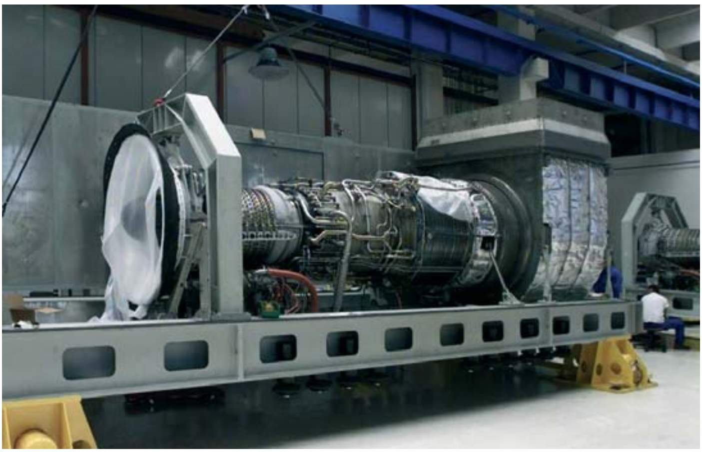


Fig. 1.8. LM2500 engine before delivery to the customer

For the normal operation of a gas turbine unit, a complex of systems is required. **The main systems serving the gas turbine unit** include:

– starting system;

– fuel system;

– oil system;

– blowing and unloading system;

– cooling system;

– compressed air system;

– air intake and gas exhaust systems.

Starting system. Since a gas turbine can only operate in the presence of compressed air, obtained only from a compressor, which is driven into rotation by the same turbine, it is obvious that the start of the gas turbine unit must be carried out from an external energy source – **a starting motor (starter)**, with the help of which the compressor rotates until gas of certain parameters and in an amount sufficient to start the gas turbine starts to flow from the combustion chamber. For example, the start of a ship's gas turbine unit is usually 80-100 s.

The starting motor is connected to the compressor rotor of the gas turbine unit during start-up. When starting a gas turbine unit with a two-spool compressor, the starter can spin the rotor of the first spool, or the rotor of the second, or both rotors simultaneously. As a rule, they strive to spin the lightest rotor, since this will require less power from the starting motor.

Starting motors use DC and AC electric motors, gas turbine engines of lower power, and turbostarters of different operating principles (gas, air, powder, etc.).

The fuel system is designed to store, prepare, and supply fuel to the combustion chamber of a gas turbine unit.

The oil system serves to store, prepare, and supply oil for lubrication and cooling of the friction surfaces of the engine and gearbox. Oil is also used as a working fluid in some mechanisms of the gas turbine unit and the gearbox control system.

In a gas turbine unit, the oil cavities are separated from the flow part by several rows of labyrinth seals. To ensure the normal operation of these seals, as well as the oil system as a whole, the engine oil cavities have an outlet to the atmosphere. **The blowing system** connects the engine oil cavities with the atmosphere in order to reduce the pressure in them, prevent oil from escaping into the flow part through the gaps in the labyrinth seals, and also ensures the separation of oil from the air that is released into the atmosphere.

The blowing system is often considered as one whole with **the unloading system**, which ensures the maintenance of the required pressure in some air cavities of the gas turbine unit, thereby reducing axial loads on the engine supports.

The high temperature of the gas in front of the turbines of modern gas turbine units necessitates the use of special **cooling systems** in them. In most existing designs, the cooling of the nozzles and blades, disks, and turbine casings is carried out using air, which is taken from the flow part of the compressor of the gas turbine unit itself. After cooling the parts, this air enters the flow part of the turbine, where it mixes with the main gas flow (open-air cooling system).

In gas turbine plants, **the compressed air system** is used to control the gearbox and reverse, for emergency engine shutdown, operation of tire-pneumatic and disconnecting friction clutches, for engine cleaning, etc. In addition, on ships, compressed air is used to start diesel generators and for the needs of the engine room and the ship.

Compressed air systems usually include reciprocating electric compressors, air storage tanks, fittings, instruments, pipelines, etc.

Axial compressors, which are part of a gas turbine unit, are very sensitive to the quality of the air supplied to them. The air is cleaned before being fed into the gas turbine unit by **an air intake system**.

The exhaust system serves to discharge exhaust combustion products from the gas turbine unit into the atmosphere.

1.3. Typical schemes of marine gas turbine engines

The multifunctionality of use and various operating conditions have led to the presence of different structural schemes of ship gas turbine engines. And the first feature by which they can be classified is **the type of power turbine**. Accordingly, gas turbine engines are distinguished:

– with free power turbine;

– with blocked power turbine.

It should be noted that in the general case, there may be several turbines in a gas turbine engine. Each of them has its own functional purpose. The turbine that drives the compressor is called **the compressor turbine**; the turbine that transmits power to the consumer (drives the consumer rotor) is called **the power turbine**. Various gas turbine engine schemes by type of power turbine are shown in Fig. 1.9.

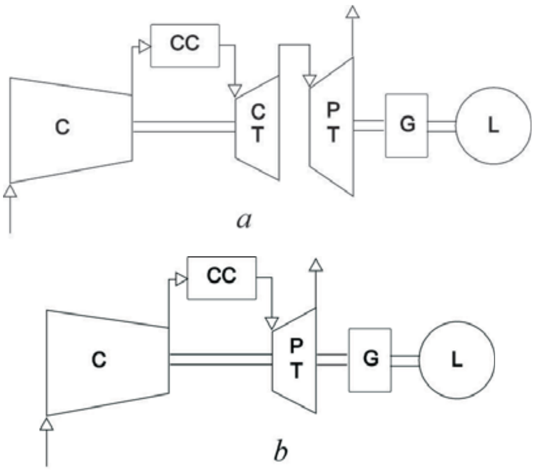


Fig. 1.9. Schemes of a gas turbine engine with different types of power turbine: *a* – two-shaft scheme with a free power turbine; *b* – single-shaft scheme with a blocked power turbine; C – compressor; CC – combustion chamber; CT – compressor turbine; PT – power turbine; G – gearbox; L – consumer

A feature of the design of an engine with a free power turbine is that in it the power turbine is free from a rigid (mechanical) connection with the compressor rotor (Fig. 1.9, *a*). Accordingly, in the engine, there are separate turbine(s) that drive the compressor(s), and there is a turbine that works only for the consumer.

It should be noted that the layout of a gas turbine engine with a free power turbine allows it to be represented as two main modules – a gas generator and a free power turbine. The free power turbine produces useful power, while the gas generator (this is the entire previous part of the engine) prepares a high-pressure and high-temperature working fluid for the power turbine. A typical example of a ship engine of this scheme is the LM2500 (see Fig. 1.7).

In gas turbine engines with a blocked power turbine, the power turbine simultaneously performs the functions of a compressor turbine (i.e., it is blocked by the compressor, as it were) (Fig. 1.9, *b*).

Each of the two schemes presented has certain advantages and disadvantages, which determine the specifics of their application.

The free power turbine design is used in gas turbine engines that require flexibility in operation, i.e., often operate at part-load conditions. These are engines used to drive marine engines, natural gas compressor station superchargers, ground vehicle engines, and helicopters.

This scheme is especially effective for transport gas turbine engines. This is due to the characteristic change in torque at partial modes (the so-called **traction characteristic**), which makes a clutch gearbox unnecessary or allows you to minimize the number of its gears. As can be seen from Fig. 1.10, single-shaft engines (curve "*1*") are characterized by a sharp drop in torque when switching to a partial power mode, which makes them completely unsuitable for use in transport (with mechanical transmission of torque to the consumer). For comparison, the dashed line shows the change in torque for a piston engine.

A gas turbine engine with a free power turbine has a torque characteristic (curve "*2*") even better than that of piston engines. Moreover, in such engines, it is quite possible to achieve a starting torque value (starting torque) two or even three times greater than at nominal operation.

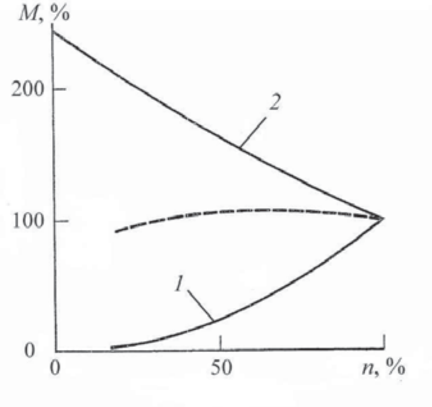


Fig. 1.10. Dependence of torque on the engine output shaft on its rotation speed [2]: *1* – for a single-shaft gas turbine engine; *2* – for a twin-shaft gas turbine engine with a free power turbine; – – – – for a piston engine

The advantage of a gas turbine engine with a free power turbine is also the significantly lower power of the starting device, since only the gas generator rotor needs to be spun up during start-up, the mass of which is less than the total mass of the rotor part of the engine.

According to [1], the absence of a direct mechanical connection between the compressor and the consumer has a positive effect (from the point of view of the stability margin) on the compressor when it operates in transient and partial modes.

The free power turbine scheme is convenient to use when creating stationary and marine gas turbine engines based on converted turbojet engines (a free power turbine is added to the original engine instead of a jet nozzle).

The scheme under consideration is also convenient from the point of view of the multipurpose use of the created gas turbine engine. Depending on the specific field of application (propeller drive, natural gas supercharger, current generator), only the free power turbine of the base engine is modified; the gas generator part is practically unchanged. The latter is considered important, since transport and drive engines are often produced in relatively small series and multipurpose use allows you to significantly expand and thereby reduce the cost of production.

In conclusion, the free power turbine scheme is used to create propulsive marine gas turbine engines that drive a propeller through a mechanical transmission.

Gas turbine engines with a blocked power turbine are simpler to manufacture and cheaper. The smaller number of individual structural elements compared to the previous scheme results in a smaller total number of shafts, supports, transitional parts of the casing, a simpler design of the lubrication system and, ultimately, better reliability.

Gas turbine engines with a blocked power turbine are used in power units that operate mainly in one (nominal) mode of operation. They are ideally suited for driving electric generators. As a conclusion, in marine transport these engines are used for generating electric current, which can then be used either to ensure the movement of the vessel or to meet general ship needs.

The second characteristic by which marine gas turbine engines can be classified is **the number of compressors**.

Accordingly, marine gas turbine engines are distinguished by the number of separate, mechanically unrelated compressors:

– engines with single-spool compressor;

– engines with two-spool compressors.

It should be noted that one of the directions of increasing the efficiency of a gas turbine engine is to increase the initial gas temperature before the turbine. As follows from the calculations, an increase in the initial gas temperature necessitates a simultaneous increase in the initial gas pressure. However, the greater the pressure in the last stages of the compressor at nominal mode, the higher the probability of surge in an axial multistage compressor when it operates at partial load modes.

To prevent surge phenomena, various design solutions are used: rotary vanes of the inlet guide device and the guide devices of the first several stages of the compressor, air release from the intermediate stages into the atmosphere, etc.

One of the most effective design solutions is to divide the compressor of a gas turbine engine into several separate, mechanically unrelated parts. Each of these stages has its own rotor, supports, its own speed, and its own turbine, which drives it into rotation. The connection between the stages is aerodynamic - by the flow of air passing from one part to another.

The higher the pressure increase rate in the compressor, the more cascades it is advisable to divide it into. According to [21], this can be done starting from a pressure increase rate of 8.

The use of a multi-spool scheme provides a self-regulating effect for the compressor, which in some cases makes it possible to do without other means of preventing surges.

Reducing the inconsistency of individual compressor stages in this design **at partial operating modes**:

– increases the compressor's stability margin (i.e., reduces the likelihood of surge);

– improves engine efficiency (since the need for other, costly control means, such as air bypass from the compressor, is completely or partially eliminated);

– reduces vibration stresses in the blades.

It should be noted that the multi-spool scheme has no advantages when operating in nominal mode. Its **disadvantages** include:

– significant complication of the engine design (an increase in the number of individual structural parts leads to an increase in the total number of shafts, supports, transitional parts of the casing, a complication of the lubrication system, etc.);

– increasing the size and weight of the engine;

– reduction in the reliability of the structure and increase in its cost (inevitable consequences of increasing complexity).

The structural schemes of marine gas turbine engines with one and two mechanically unconnected compressors are presented in Fig. 1.11.

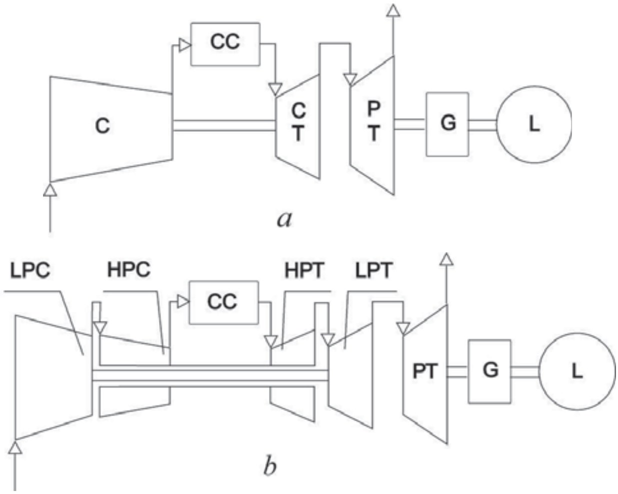


Fig. 1.11. Schemes of a gas turbine engine with different number of compressors: *a* – single-spool scheme with a free power turbine; *b* – two-spool scheme with a free power turbine; C – compressor; CC – combustion chamber; CT – compressor turbine; PT – power turbine; G – gearbox; L – load (consumer); LPC – low pressure compressor; HPC – high pressure compressor; HPT – high pressure turbine; LPT – low pressure turbine

Scheme of a single-spool gas turbine engine with a free power turbine (Fig. 1.11, *a*) is currently realised in such marine gas turbine engines as [9]:

– LM500, LM1500, LM2500, LM2500+ and LM2500G4 from GE Aerospace (USA);

– TF40 and TF50 from Vericor Power Systems Inc. (USA).

The scheme of a two-spool gas turbine engine with a free power turbine (Fig. 1.11, *b*) is currently the most popular among marine power plants.

In such engines, the air sequentially passes through a mechanically unconnected low-pressure compressor and a high-pressure compressor. The rotation frequencies of these compressors are different and are accepted based on the conditions of economic operation. The rotation frequency of the high-pressure compressor is taken to be slightly higher, which allows to reduce its diameter and thereby increases the length of the blades of the last stages in order to reduce final losses and increase their efficiency.

After heating in the combustion chamber, the working fluid enters the turbine section, which consists of a high-pressure turbine, a low-pressure turbine, and a free power turbine arranged in series.

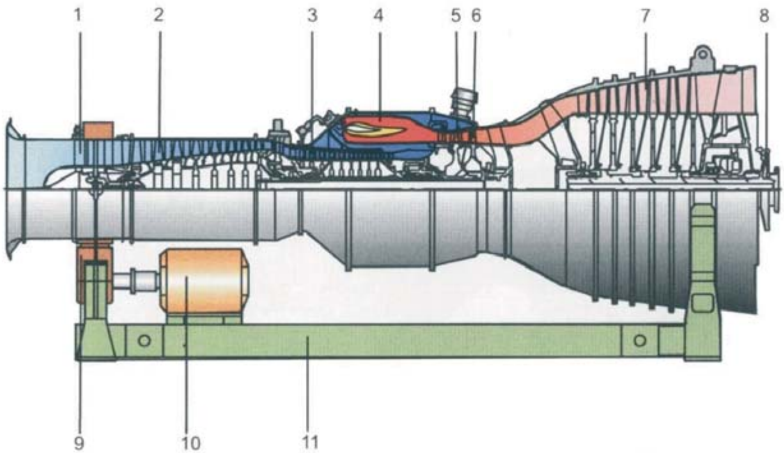


Fig. 1.12. Zorya-Mashproekt UGT6000 engine (cross-section):
1 – inlet guide vanes; 2 – low pressure compressor; 3 – high pressure compressor; 4 – combustion chamber; 5 – high pressure turbine; 6 – low pressure turbine; 7 – power turbine; 8 – output flange; 9 – gearbox; 10 – electric starter; 11 – frame

The high-pressure turbine uses all the generated power to drive the high-pressure compressor, which is mechanically connected to it by a shaft, forming the so-called "high-pressure turbocompressor". The low-pressure turbine uses the generated power to drive the low-pressure compressor. The shaft connecting the turbine and the low-pressure compressor passes inside the hollow shaft of the high-pressure turbocompressor. This "shaft-in-shaft" layout is a traditional design solution in aircraft gas turbine engines, and from there it moved to engines for other purposes.

The useful power transmitted to the consumer is generated using a free power turbine.

This is a structural scheme realised in such marine gas turbine engines as [9]:

– Olympus, Tyne, Spey and MT30 from Rolls-Royce Holdings plc (Great Britain);

– UGT3000, UGT6000 (Fig. 1.12), UGT15000, UGT16000 and UGT25000 from Zorya-Mashproekt (Ukraine).

Control tasks and questions

1. How do gas turbine power plants compare to low-speed diesel engines regarding weight and space requirements?

2. Why do gas turbine engines produce a lower concentration of harmful substances in their exhaust gases compared to other power plants?

3. What is the functional purpose of the compressor and the combustion chamber within a gas turbine unit?

4. Describe the ideal thermodynamic cycle (Brayton cycle) that governs the operation of a gas turbine.

5. What is the difference between a "gas turbine engine" and a "gas turbine power plant" according to the provided definitions?

6. Why is a gas turbine engine with a free power turbine preferred for transport and propulsion applications?

7. What is the primary motivation for using a multi-spool (two-spool) compressor design in a marine gas turbine?

2. DETAILS OF THE FLOW PATH OF GAS TURBINE ENGINES

The flow path of a gas turbine engine is understood as its internal space through which the working fluid flows. Details of the flow path include all elements located in the path of the working fluid. These include the blades and guide vanes of the compressor, the nozzle vanes and blades of the turbine, as well as their fastening elements.

2.1. Details of axial multistage compressors

Currently, the dominant type of compressor for gas turbine engines is the multistage axial compressor. The use of other types of compressors is limited to some narrow applications where the use of an axial compressor is undesirable for one reason or another [18].

The scheme of a multistage axial compressor is shown in Fig. 2.1. Its main components are: rotor 11 with blades 5 fixed to it; casing 7, to which guide vanes 6 and end seals 2 are attached; rotor supports (bearings) 3.

The combination of one row of rotating blades and one row of stationary guide vanes located behind them is called **a compressor stage**.

The air sucked in by the compressor passes sequentially through the following elements: inlet casing 1, inlet guide vanes 4, group of stages 5 and 6, exit straightening vanes 8, diffuser 9, and outlet casing 10. Let's consider the purpose of these elements.

The inlet casing is designed to supply air from the atmosphere to the compressor blades uniformly in a circle. The type of inlet casing is selected from the layout conditions. If the axial size of the compressor is not limited, then an axial inlet casing is taken (as the most efficient); in other cases, an annular, spiral, or elbow-shaped one is used.

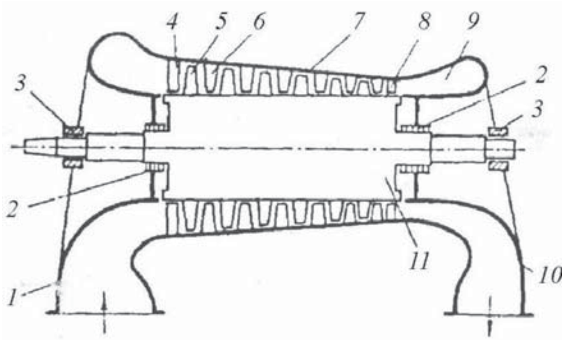


Fig. 2.1. Scheme of a multistage axial compressor:

1 – inlet casing; 2 – end seals; 3 – supports (bearings); 4 – inlet guide vanes; 5 – blades; 6 – guide vanes; 7 – casing; 8 – exit straightening vanes; 9 – diffuser; 10 – outlet casing; 11 – rotor

The inlet guide vanes must provide the necessary flow direction before entering the first stage (to increase its efficiency). The vanes of this device form confusing channels (with a passage cross-section that decreases as it moves). When passing through them, as well as earlier through the inlet casing, the flow velocity increases, and the pressure decreases. The vanes of the inlet device can be made rotary to regulate the compressor operation.

In the compressor stages, the air pressure increases due to the use of mechanical energy from the rotation of the rotor. From the last stage, the air enters the straightening device, which is designed to untwist the flow, i.e., to give it an axial direction before entering the diffuser. In the diffuser (a channel with a through-section that increases as it moves), the air pressure continues to increase due to a decrease in the speed of its movement (the so-called kinetic energy is transformed into potential energy). The outlet casing is designed to supply air from the diffuser to the combustion chamber.

2.1.1. Blades of axial compressors

The blades of an axial compressor form diffuser channels (i.e. channels with an increasing cross-section), in which the mechanical energy of the drive is converted into the potential energy of the working fluid (increasing air pressure) (Fig. 2.2).

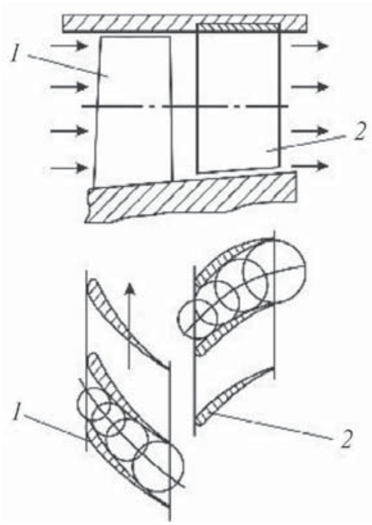


Fig. 2.2. Axial compressor stage scheme:
1 – blades; 2 – guide vanes

Compressor blades have a profile part (**feather**) 1 and a locking part (**tail**) 2 (Fig. 2.3).

The thickness of the blade profile is usually 2.5-8 % of the chord length. Thinner profiles with lower resistance are not used due to manufacturing difficulties and insufficient blade stiffness.

Due to the different angles of incidence of the flow on the blade profile along its length, the blades are made twisted, that is, they have a different orientation of the profile relative to the incident flow along its length. The twist of the blades increases with increasing their length. For this reason, the blades of the first compressor stages (as the longest) are most twisted.

The blades experience significant centrifugal and bending forces, and are also subject to vibration loads of various natures. Very high requirements are imposed on the accuracy of processing of the profile part and the cleanliness of its surface, since these factors determine the efficiency of the compressor and the strength of the blades. The blades are polished, and the notches and scratches are removed so that they have very smooth transitions. The radius of the transition from the profile part to the tail should be as large as possible to avoid increased stress concentration.

Fig. 2.3. Blades of an axial compressor:
1 – profile part (feather); 2 – tail

To reduce the mass of the blades and tensile stresses and loads on the rotor, the blades are made with a chord and thickness that decrease along the length, so that the peripheral cross-sectional area can be significantly smaller than the root cross-sectional area. At the same time, the rigidity of the profile part must be sufficient to maintain the angles of installation of the profiles in the most severe operating conditions of the blade.

The materials for the blades of axial compressors are alloy steels, heat-resistant alloys, titanium, and aluminum alloys.

Aluminum alloys are quite strong, well amenable to forging, carbing, and machining. They are used at temperatures up to 250 0C. To prevent corrosion, aluminum alloy blades are carefully anodized. (In [6], it is noted that aluminum alloys, which are widely used in aircraft engines, are subject to increased erosion wear and corrosion when operating in ground conditions.)

At higher temperatures (up to 500 ^{o}C), blades made of titanium alloys work well. The specific strength of such alloys in the temperature range of 150-500 ^{o}C exceeds the specific strength of aluminum, magnesium alloys, and steel.

The density of aluminum and titanium alloys is significantly lower than the density of steel, so blades with practically the same (in the operating temperature range) strength limits have a lower mass and are subject to lower tensile stresses from centrifugal forces.

Reinforced plastics are used for parts of compressor casings and blades. Their advantages are low density, high corrosion resistance, and low production cost. The main disadvantages of fiberglass are a low modulus of elasticity (8-9 times lower than that of steel), low fatigue strength (5 times lower than that of steel), and accelerated erosion failure.

The advantage of steel blades and blades made of heat-resistant alloys is their high mechanical strength, less tendency to form pits, nicks, and roughness when sand and small objects enter the compressor, and good performance at high temperatures.

The blades of axial compressors do not have a bandage (see Fig. 2.3). In their manufacture, instead of mechanical processing, electrochemical processing is increasingly used, which provides a higher quality of the blade surface, as well as a reduction in residual stresses.

The fastening of the blades to the rotor must meet the following requirements:

– have high strength;

– to ensure the possibility of precise installation of the blades;

– allow easy assembly and disassembly of blades;

– be easy to make;

– provide the possibility of placing a significant number of blades;

– to have a small mass.

Both circular (in annular-shaped grooves) and axial (in longitudinal individual grooves) fastenings of compressor blades are used.

Circular fastening is used in some designs of stationary engines that have a solid or welded compressor rotor, as well as in stages of aircraft engines (Fig. 2.4).

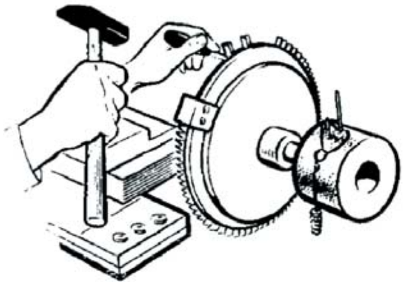


Fig. 2.4. Installation of blades in annular shaped grooves

Fig. 2.5 shows **the T-shaped fastening method** used by the English Electric Co. for EM-27P engines [15]. Each standard blade 1 is rotated 90° before installation, lowered through the keyway into a deep groove, then returned to its normal position and wound around the groove circle. The last three blades – the standard, the locking 2 and the adjustment 3 – are installed in the sequence shown in the figure. The locking blade has a rectangular tail. A gasket 4 is provided between the locking and adjustment blades to maintain the pitch. Plates 5 are installed in the tail of the last blade, the antennae of which are bent to fix the blade.

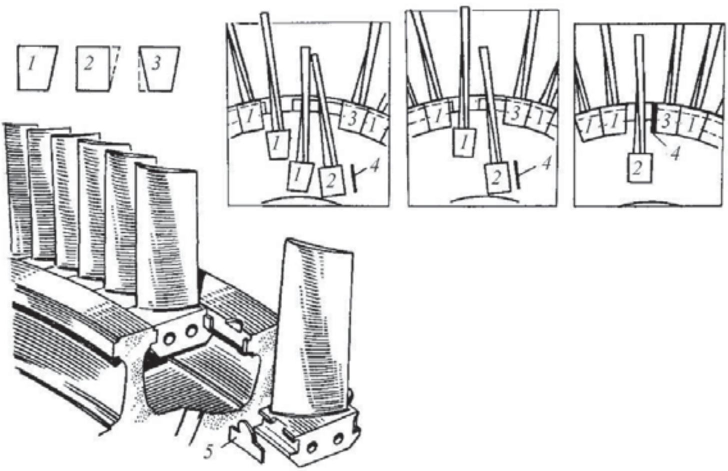


Fig. 2.5. Circular fastening of compressor blades:
1 – standard blade; 2 – locking blade; 3 – adjustment blade; 4 – gasket; 5 – plate

A modification of the T-shaped fastening is **the stream fastening method**, which has become widely used in aircraft engines in recent decades (Fig. 2.6 and 2.7).

The blades are inserted into the annular groove of the disk through one loading window, while they are fixed from rotation in a circular direction by several latches (Fig. 2.8). Each latch holds its sector of the blades, between the shelves of which a gap of 0.3-0.5 mm is provided for temperature expansions [15].

Fig. 2.6. Stream-type fastening

Fig. 2.7. Blades with stream-type tails

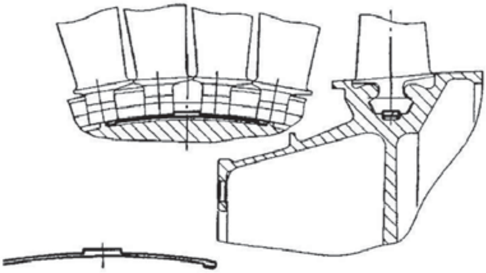


Fig. 2.8. Fixing the blades using plate elastic locks

Fig. 2.9 and 2.10 show **a hinged circular fastening** (a type of fork mount), which is used for blades of the first (longest) stages of axial compressors as a means of damping vibrations. The hinged fastening allows you to install the blade in the plane of rotation so that under the action of gas-dynamic and centrifugal forces it, turning on the hinge, is located at a certain angle to the radial direction. Under the influence of alternating forces, the blade will oscillate, converting the energy of mechanical vibrations into thermal energy by friction [6].

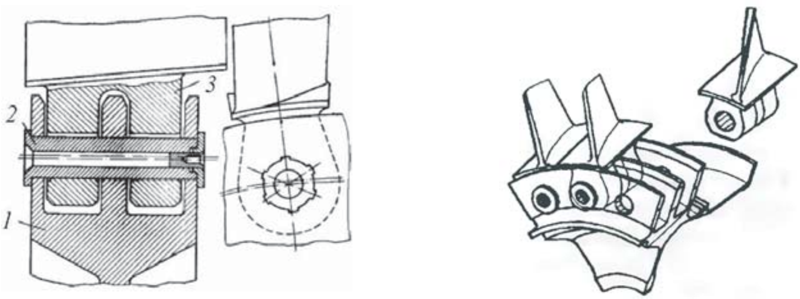


Fig. 2.9. Hinged fastening:
1 – disc rim; 2 – pin (hinge); 3 – blade

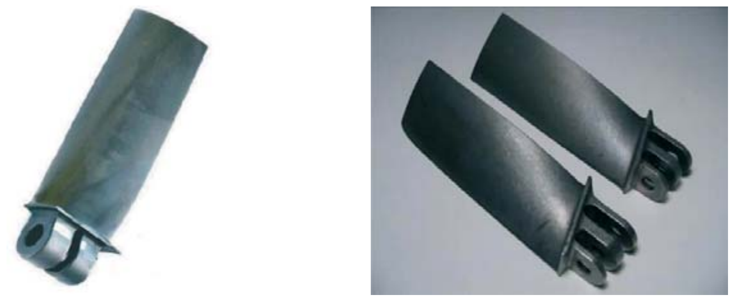


Fig. 2.10. Hinged compressor blades

The disadvantages of this connection include the large crushing and shear stresses that occur when working in the joints, as well as the large mass.

A rigid (fixed) fastening of the forked tail is also found in compressors.

To eliminate wear and seizure in the hinge, a so-called "solid lubricant" is used. The blade lug from the inside and from the ends, and the outer surface of the axis are rubbed with molybdenum disulfide powder (MoS_2).

Axial mounting of compressor blades has become more widespread. Among them, the fastening of the blades with the help of a trapezoidal groove ("dovetail") and fir-tree (toothed) fastening are most often used.

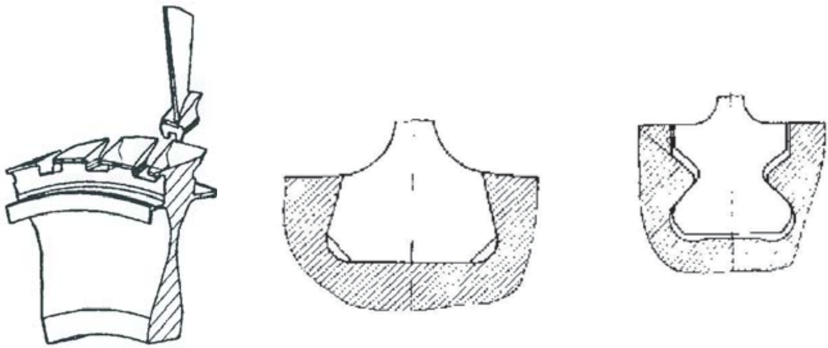


Fig. 2.11. Dovetail-type fastening

The dovetail type of fastening (Figs. 2.11 and 2.12) most fully meets the requirements for fastening compressor blades. The grooves in the disk are made by broaching, and the blade tails are milled. The blade is installed in the groove either with a tension of up to 0.08 mm or with a gap of 0.035-0.060 mm. When fitting with tension, its value is chosen so that there is no overstress in the locking part of the rim. A loose fit (with a gap) is more often used. It provides easy installation and removal of the blade, as well as better damping properties [8].

The blade fastening slots are usually placed at an angle to the rotor axis, close to the angle of the blade root sections. This increases the strength of the connection (by reducing the specific pressure on the contact surfaces).

The dovetail fastening allows for a large number of blades to be placed on the wheel. This joint is simple in design, has a low mass and sufficient strength.

Fig. 2.12. Blades of the GTN-25 engine compressor

With a large length of blades, fasteners resembling a "dovetail" are used as a constructive means of damping vibrations by friction (damping), but with rounded working surfaces (Fig. 2.11, *c*). The blades are installed in a disk with large gaps, which provides swinging at the end of the blade up to 5-8 mm. This allows the blades to self-align and provides some "hinging" of the connection [15].

Fastening the blades in the longitudinal grooves requires **fixing** them **against axial movements**. The fixing can be common for all blades or individual and can be carried out using plates, pins, etc. (Fig. 2.13).

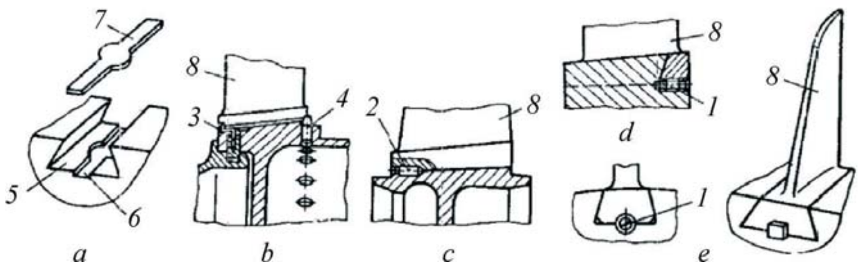


Fig. 2.13. Fixation of the blades with dovetail-type fastening [15]:
a – general view of the fastening, *b*, *c*, *d* – methods of fixing the blades against axial displacements; *e* – blade together with the disk; 1 – threaded pin; 2 – axial pin; 3 – plate stopper; 4 – radial pin, 5 – disk groove; 6 – milling for the locking plate in the disk; 7 – locking plate; 8 – blade

The fir-tree fastening (Fig. 2.14) is very common in gas turbines. It is rarely used in axial compressors, since at relatively low compressor blade loads, simpler designs can be used, such as the "dovetail" design, which does not require expensive shaped broaches to machine the blade tails and the grooves in the disk.

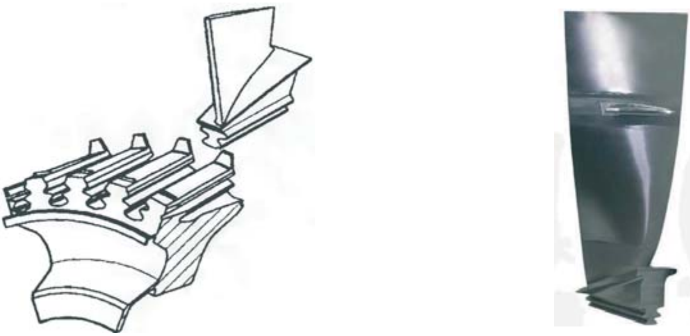


Fig. 2.14. Fir-tree fastening of compressor blades

Fir-tree fastening in compressors can only be used with steel blades and discs. When using aluminum or titanium alloys as the material, the connection may break due to insufficient plasticity of these alloys and chipping of the teeth.

2.1.2. Guide vanes of axial compressors

The guide vanes of axial compressors form diffuser (in the general case) channels (see Fig. 2.2), designed for:

– reducing the speed of movement and increasing air pressure (i.e., to convert the kinetic energy of the flow into potential energy);

– redirection of air at a given angle into the working channels of the next stage.

The inlet guide vanes are installed before the first stage of the compressor, which provides the initial swirl of the air. At the outlet of the compressor, there are the exit straightening vanes designed to spin the flow and its axial movement behind the compressor.

Guide vanes are manufactured by precision stamping, precision casting or from rolled profiled strip. The profile part of such vanes does not require mechanical processing, but is only polished and subjected to anti-corrosion treatment. The main geometric characteristics of the vanes are determined in the same way as for the blades by gas-dynamic calculation.

There are many different ways to attach guide vanes to the compressor casing. There is single-support (cantilever) vane fastening (without an internal banding tape) and double-support fastening (with an internal banding).

Guide vanes can be attached directly to the compressor casing or in rings (half rings) – bandages that form guide devices of the frame type.

By design, guide devices are collapsible and non-collapsible (welded, riveted).

With **cantilever fastening** (Fig. 2.15), the cross-sections of the vanes in terms of strength and vibration resistance should be larger than those of the vanes with double-sided fastening. Cantilever fastening is rarely used, since with it significant air flows occur through the gap between the ends of the vanes and the rotor, which worsens the efficiency of the compressor. Reducing the flow is achieved by reducing the radial gap. However, a number of factors (rotor runout, etc.) do not allow reducing the gap to an acceptable value.

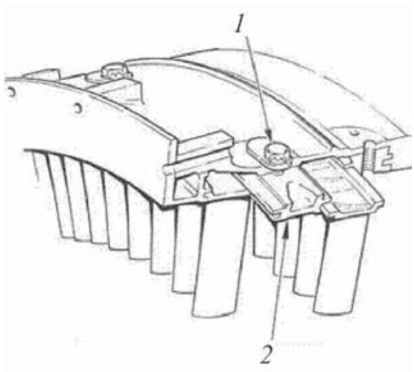


Fig. 2.15. Cantilever fastening of guide vanes [10]:
1 – fastening screw; 2 – guide vane band ring

Fig. 2.16, *a* shows a design in which the vanes are cantilevered in an outer shroud, which in turn is attached to the casing by means of tract rings (see also

Fig. 2.15). This fastening facilitates assembly, but slightly increases the weight of the compressor.

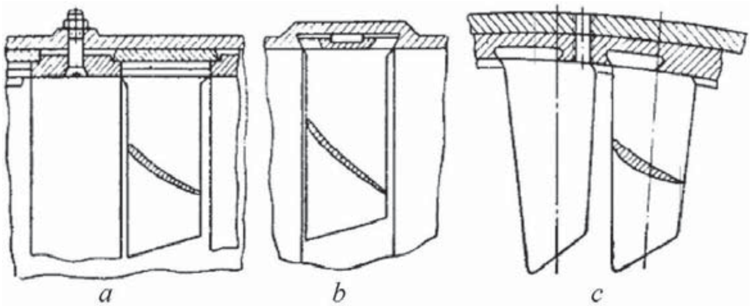


Fig. 2.16. Cantilever fastening of guide vanes

Fig. 2.16, *b* shows a design in which the vanes are attached directly to the compressor casing in annular trapezoidal grooves (similar to the circular fastening of the blades). To facilitate assembly, the groove for the vanes is made deeper, which allows the vanes to be freely inserted and then secured using clamping wedges.

Cantilever fastening of vanes is also possible in steel rings or half rings with longitudinal trapezoidal grooves (Fig. 2.16, *c*) (similar to axial mounting of blades). Each ring is attached to the casing with screws.

Fig. 2.17 shows the fastening of the guide vanes of the 17-stage single-spool compressor of the GE Aerospace J79 turbojet engine. Fig. 2.18 shows the fastening of the guide vanes of the 10-stage single-spool compressor of GE Aerospace T58 turboshaft engine.

The double-support fastening (Fig. 2.19) of the vanes allows for higher efficiency of the compressor. The guide devices of such fastening can be of two types:

– with rigid fastening of the vanes to both bandages;

– with rigid fastening of the vanes to one, usually outer, bandage and a loose fastening, which allows radial movement of the vanes, to the other bandage.

Fig. 2.17. Guide vanes of the 17-stage single-spool compressor of the J79 engine

Fig. 2.18. Guide vanes of the 10-stage single-spool compressor of the T58 engine

Rigid fastening is used in guide devices made of welded steel, with relatively short vanes, where thermal stresses caused by uneven heating of parts are small.

With a split compressor casing, the bands are made split – consisting of two halves, each of which is independently attached to the casing using radial pins or screws.

Fig. 2.20 shows the guide device of the low-pressure compressor of the Zorya-Mashproekt DI59 engine. At one end, the vane 2 has a shelf that fixes it in a certain position in the outer band ring 1, at the other – a cylindrical pin for fastening the vane in the inner band 4 (see also Fig. 2.21). Grooves are made in the inner rings, into which copper-graphite inserts 3 are inserted, which, in combination with the labyrinths made on the rotor disks, form seals that prevent air flow between the compressor stages. The guide device is fixed inside the casing in cylindrical grooves with screws.

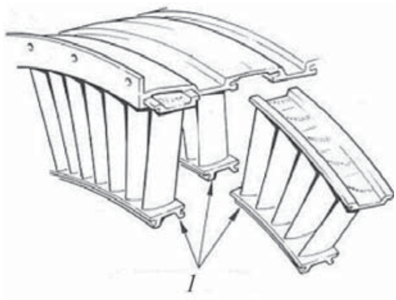


Fig. 2.19. Double-support fastening of guide vanes [10]:
1 – guide vanes with banding rings

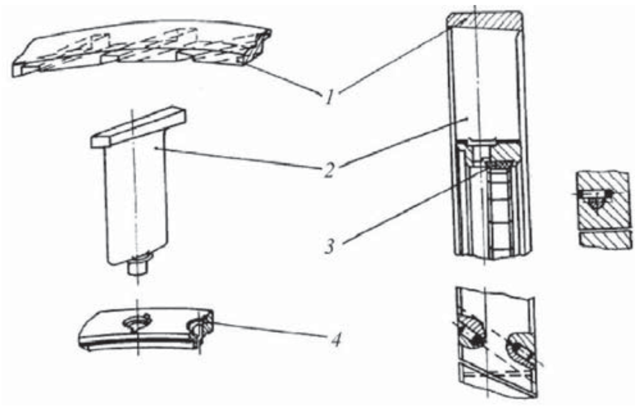


Fig. 2.20. Low-pressure compressor guide device of the Zorya-Mashproekt DI59 engine [6]

This type of fastening allows for easy removal and replacement of the vanes. However, the design of the guide device becomes more complicated and its weight increases.

Fig. 2.21. Guide vanes of an axial compressor

Fig. 2.22 shows the guide device of the low-pressure compressor of the Zorya-Mashproekt D050 engine. It consists of an outer 1 and an inner 4 band rings and guide vanes 2 located between them. The outer ring has dovetail-type grooves, where the guide vanes are inserted with their upper shelves. The inner ring has notches into which the ends of the vanes enter. The protruding parts of these ends are riveted from the inside of the ring. The inner ring has an annular groove in which copper-graphite inserts 3 are installed. The inserts, in combination with labyrinths made on the rotor disks, form a seal for the flow part of the engine. The guide device is cut horizontally into two parts, each of which is attached to its own half of the compressor casing.

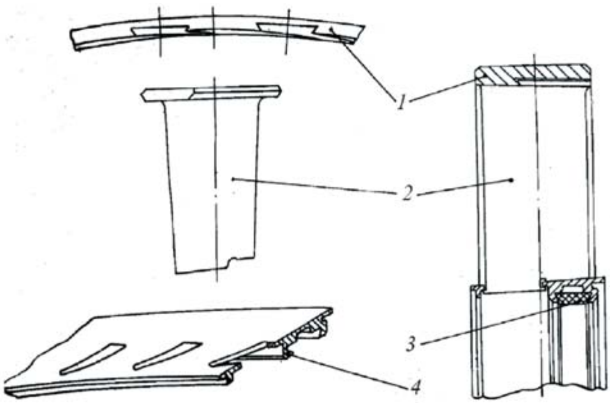


Fig. 2.22. Low-pressure compressor guide device of the D050 engine [15]:
1 – outer band ring; 2 – guide vanes; 3 – copper-graphite inserts; 4 – inner band ring

With a large swirl of the air flow in the working channels of the last stage, an additional row of fixed guide vanes (**exit straightening vanes**) is sometimes installed, which ensures the axial exit of the air flow into the diffuser.

In order to reduce losses caused by unfavorable angles of attack of the blades at reduced air flow rates in partial operating modes and to prevent blade surge, the inlet guide vanes and the guide vanes of the first stages of the compressor are often made rotary.

Thanks to the installation of rotary vanes, it is possible to use a single-spool compressor with a high pressure increase rate and good performance characteristics.

The rotary vanes of the guide devices (Fig. 2.23) are made of stainless steel or aluminum alloy. Their leading edge can be heated by hot air, which is taken from one of the compressor stages. The vane pins are placed in bearings. Steel bushings with a cylindrical or barrel-shaped outer surface are pressed onto the aluminum alloy pins. The latter eliminate the possibility of the vane jamming in case of deformation of the casing.

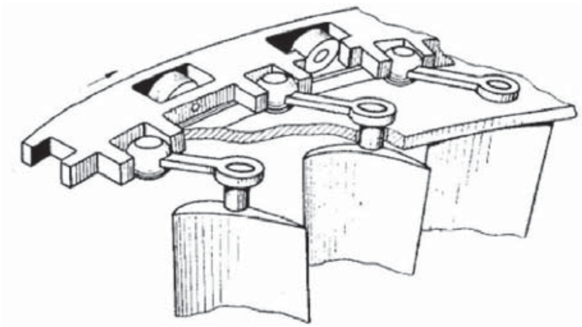


Fig. 2.23. Rotary guide vanes of the compressor

The simultaneous rotation of all vanes is carried out either by means of a steel rotary ring located outside the casing and connected by its cutouts (forks) to levers fixed on the upper trunnions of the vanes (as in Fig. 2.23), or by means of a gear wheel connected to toothed sectors fixed on the lower trunnions of the vanes. In both cases, the rotation of the vanes is controlled by means of power cylinders.

Figs. 2.24 and 2.25 show the design of a 17-stage axial compressor of a GE Aerospace J79 turbojet engine (also known as the LM1500 marine engine after conversion), the inlet guide device and the first six stages of which are equipped with rotating guide vanes.

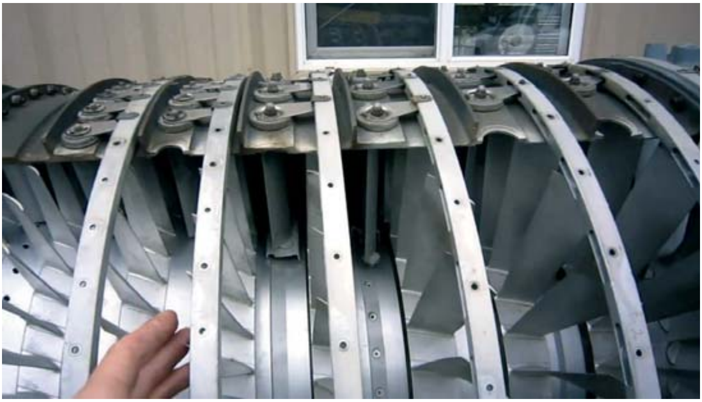


Fig. 2.24. Compressor part of the J79 engine

Fig. 2.25. Inlet guide vanes of the J79 engine

2.2. Details of axial turbines

In the vast majority of cases, axial turbines are used in gas turbine engines; in low-power engines, centripetal radial turbines are sometimes used.

We will consider the structure and principle of operation of the simplest axial turbine using the example of a design, the diagram of which is shown in Fig. 2.26.

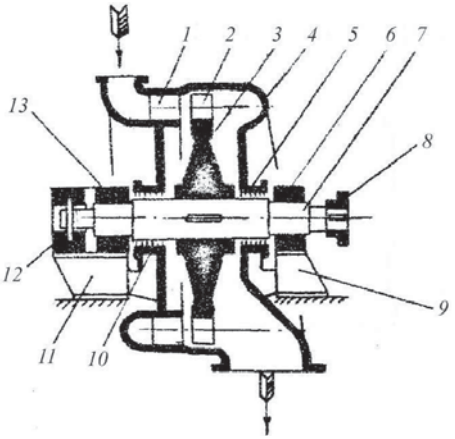


Fig. 2.26. Structure of an axial turbine [11]

The main elements of the axial turbine are: nozzle device 1, formed by vanes fixed immovably in the casing 4, and blades 2, assembled around the rim of the disk 3. A row (around the circumference) of blades forms a working crown. The disk can be manufactured as a single part with the shaft 7 or can be made separately and then rigidly connected to it by welding or fasteners. The shaft together with all the parts installed on it forms the rotating part of the turbine during operation – the rotor.

The position of the rotor relative to the casing is fixed by support (radial) 6 and 13 and thrust 12 bearings. The bearings are installed in bearing casings 9 and 11, connected to the lower half of the casing. With the help of these casings, the turbine is attached to the foundation. The casing has pipes for supplying (inlet) and removing (exhaust) the working fluid. External seals 5 and 10 are provided at the points where the shaft exits the casing to isolate the internal cavities from the environment. The set of all fixed parts forms the turbine stator. The power generated by the turbine is transmitted to the consumer through a flange or connecting coupling 8.

2.2.1. Nozzle vanes of axial turbines

The nozzle vanes of gas turbines, placed on the band rings, form an annular lattice (nozzle device), in which the potential energy of the gas is converted into kinetic energy (flow acceleration) (Figs. 2.27 and 2.28). The inner and outer band rings limit the annular channel of the flow part of the turbine. In some designs, the role of the outer band ring is performed by the turbine casing.

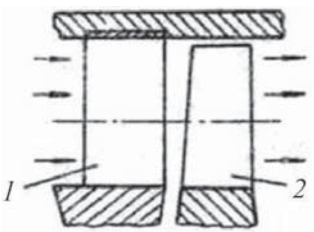


Fig. 2.27. Turbine stage scheme:
1 – nozzle vanes; 2 – blades

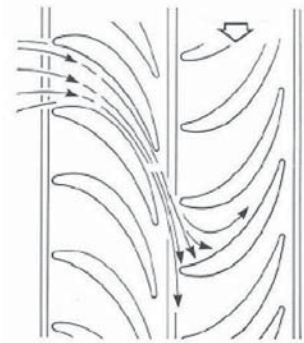


Fig. 2.28. Scan of sections of nozzle vanes and blades

Fig. 2.29. Nozzle vane of the GTK-10-4 engine

Fig. 2.30. Nozzle vane of the Turboatom GTE-45-3 engine

The nozzle vanes of the first stage are made wider, and the following ones are made with the aim of reducing the axial dimensions and mass of the turbine. The vanes are twisted along the length and have a constant or variable profile and chord along the length (Figs. 2.29 and 2.30). In nozzle vanes with a chord that increases towards the periphery, the relative pitch of the grid along the entire length of the vane is closer to the optimal value.

The design of nozzle vanes depends on the method of their fastening and manufacture. The vanes can be with shelves, with trunnions, lugs, etc., solid or hollow (Fig. 2.31).

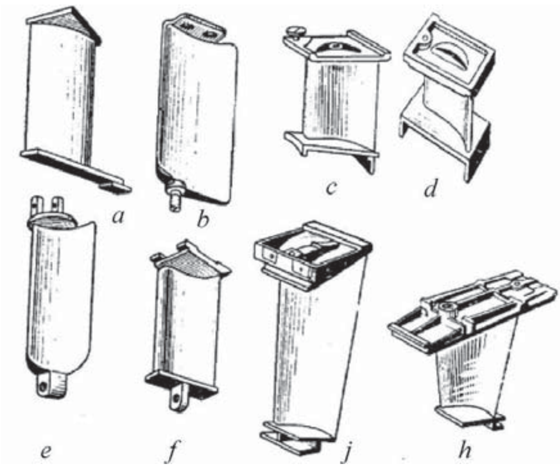


Fig. 2.31. Nozzle vanes:
a, *b*, *c*, *d* – from first stage; *e*, *f*, *j*, *h* – from subsequent stages

Nozzle vanes experience significantly lower mechanical loads compared to blades, but are exposed to high-temperature stresses caused by uneven heating. When starting a gas turbine engine and during rapid load discharges, the vanes experience significant stresses associated with a sharp change in the temperature of the inlet edges and a greater thermal inertia of the massive middle part. This temperature difference, even with a relatively small number of heat transfer cycles, can lead to fatigue failure of the vanes. The use of hollow vanes minimizes this difference. In addition, the presence of a cavity in the vanes creates the possibility of their cooling and also reduces their mass and metal consumption. However, hollow vanes have a large thickness of the profile and outlet edges, which somewhat worsens their gas-dynamic qualities.

To increase the heat resistance of nozzle vanes, technological processes are used during their manufacture that do not produce final tensile deformations, such as broaching, electromechanical processing, electropolishing, etc.

Nozzle vanes are made of foundry alloys by precision casting from investment casting patterns. Both solid and hollow-cooled vanes are produced by casting.

In cooled cast vanes, the coolant (most often compressed air) exits either through slotted holes on the side of the trough near the trailing edge (which allows the trailing edges to be kept thin) or through slotted holes in the trailing edge.

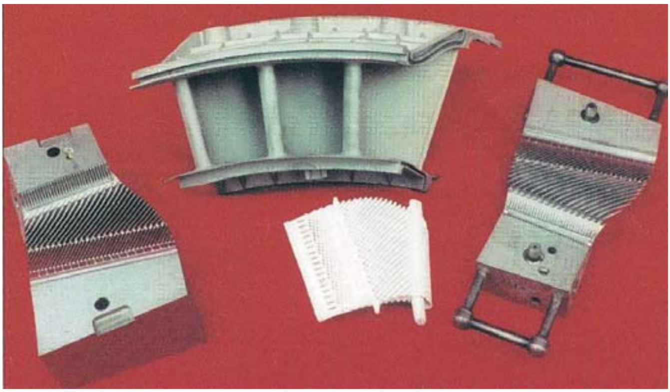


Fig. 2.32. Cooled nozzle device, its casting rod, and a device for manufacturing

Fig. 2.32 shows a cast three-vane package of a cooled nozzle device manufactured by Zorya-Mashproekt, as well as a casting core and a device for manufacturing this core.

Hollow nozzle vanes are also made of deformable alloys by flexible stamping of sheets with subsequent welding along the leading edge. This significantly reduces material consumption, reduces the complexity of manufacturing, and lowers the cost of the vane. Compared to solid vanes, such a vane has 3-4 times greater heat resistance, as well as higher reliability and resource.

Thermocouples can be mounted on the leading edges of the nozzle vanes to regulate the gas turbine engine based on gas temperature.

Nozzle vanes, due to significant temperature deformations, are usually not included in the force connection of the turbine casing with the bearing casing. For this purpose, other elements (racks, pins, spokes, etc.) are used.

Nozzle vanes can have a single-support (cantilever) or double-support fastening.

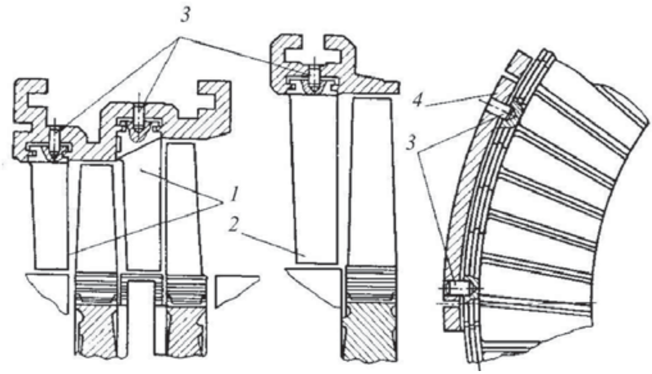


Fig. 2.33. Nozzle vanes of the GT-700-5 engine:

1 – high-pressure turbine vanes; 2 – low-pressure turbine vanes; 3 – pin; 4 – segment insert

Cantilever fastening, in which the vane loads are transmitted to only one band ring (usually the outer one, rigidly connected to the engine casing), is sometimes used in nozzle devices of the last stages. In Fig. 2.33, the vanes are cantilever-mounted in split-segment inserts by T-shaped shelves of the same profile (similar to the circular fastening of the blades). A gap of 0.2 mm is provided between the shelves to compensate for expansion. The end vanes of each segment are fixed by radial pins 3.

In the case of **a double-support fastening** (Fig. 2.34), the forces from the vane are transmitted to both the outer band ring and the inner one. The double-support fastening of vanes is widely used in nozzle devices of all stages, where the band rings are rigidly attached to the corresponding power elements of the casing.

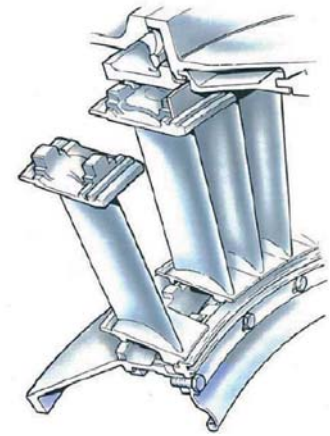


Fig. 2.34. Nozzle device with double-support vane fastening [10]

Double-support and cantilever vane fastenings are made in the form of fixed or detachable connections. Accordingly, nozzle devices are divided into fixed and detachable.

Nozzle devices with non-detachable fastening can be manufactured by casting, welding, and with riveted vanes.

For the manufacture of cast nozzle devices, the precision casting method is used. Vanes and rims are usually cast as one piece, in sections of 2-4 vanes each (Fig. 2.35) or as a joint casting. In the latter case, to reduce thermal stresses, the internal band is cut in the gaps between the vanes in several places. The advantage of such a design is significant rigidity. Its disadvantages include: difficulties in grinding the vanes and obtaining high-quality surface treatment; the presence of thermal stresses in the structural elements; difficulties in repair.

In welded nozzle devices, the vanes are welded to both bands, one of which has cuts, or to one of the bands, forming movable joints with the other. This design is simple, but it is very laborious to manufacture due to the welding of a large number of vanes. However, with such fastening of the vanes, their precise installation before welding is difficult, and, in addition, during the welding process, the vanes may be bent, and the shape of the flow part may be distorted. Therefore, the welding of the vanes must be performed in special devices. The replacement of defective vanes is completely excluded.

Fig. 2.35. Cast nozzle devices

Technologically simpler is the method of fastening the vanes to the bandage by riveting. The vanes have protrusions that, after being inserted into the corresponding grooves of the bandages, are riveted.

The main advantage of a non-detachable fastener is the relatively small number of technological operations for machining the fastener parts. The disadvantages include the difficulty of replacing defective vanes during repairs.

Collapsible nozzle devices with double-support fastening are made either with rigid fastening of the vanes in one of the bandage rings and movable in the other, or with movable fastenings in both bandage rings.

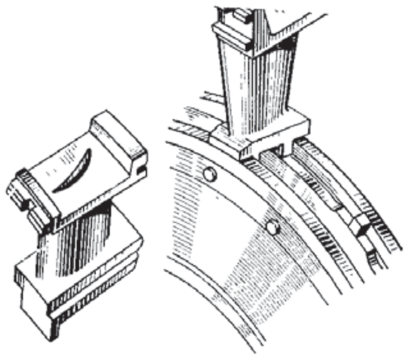


Fig. 2.36. Nozzle vane of the Centrax CS600 engine [4]

Rigid fastening of the vanes (Fig. 2.36) is usually performed in the outer band ring or turbine casing:

– by screws that are passed through the vane shelf;

– using a shelf, the front and rear edges of which enter the grooves of the collapsible bandage ring and are fixed relative to the body with a radial pin;

– using an eyelet on the vane shelf, through which the vane is attached to the turbine casing flange.

With movable supports in both band rings (floating vanes), circular and axial fixation of the vanes can be carried out in various ways, for example:

– using shoes that are attached to the body of the nozzle device with radial screws; each of the shoes has cuts corresponding, on one side, to the profile of the back, and on the other, to the profile of the trough; thus, two adjacent shoes form a cutout that has the shape of a vane profile, into which the vane is installed at one end; its other end is fixed in a similar cutout formed by the shoes on the opposite side;

– using special notches that have the shape of a vane profile and are made in the outer and inner band rings;

– using oblique grooves in the outer and inner casings of the nozzle device, into which the vanes enter with their shelves with a radial gap, which ensures freedom of thermal expansion of the vanes.

The radial fixation of the floating vanes is carried out either by abutting the vane against the casing wall, or by means of a locking ring, the protrusion of which enters special cutouts in the vanes, fixing them all at the same time, or by means of cuffs welded to the bandage ring and cotter pins that are inserted into the holes in the cuffs and vanes.

Collapsible nozzle devices have the following advantages:

– possibility of good surface treatment of vanes and obtaining a high-precision profile;

– lower thermal stresses resulting from temperature differences between vanes and bands;

– possibility of quick replacement of a damaged vane during repair.

As a result, collapsible nozzle devices are currently the most widespread.

It should be noted that cantilever-mounted nozzle devices are usually manufactured with rigid fastening of the vanes to the turbine casing (or to the outer shroud ring), for example, by seam welding. Collapsible nozzle devices with cantilever-mounted vanes, compared to non-collapsible ones, are more difficult to manufacture, more expensive, and heavier.

The nozzle device fastening perceives and transmits the torsional moment arising on the nozzle device, the axial forces arising from the pressure difference before and after the device, and the acceleration of the gas flow, as well as the mass and inertial forces of the nozzle device.

Nozzle devices are attached to nozzle casings, which in turn are attached to the power elements of the engine casing.

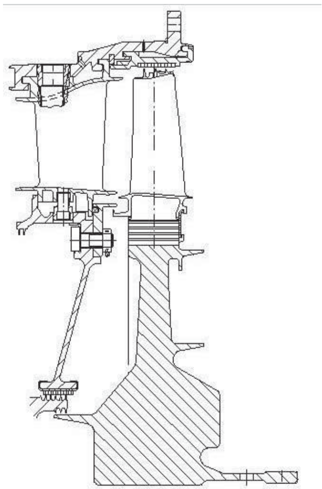


Fig. 2.37. Turbine stage with diaphragm-type nozzle device

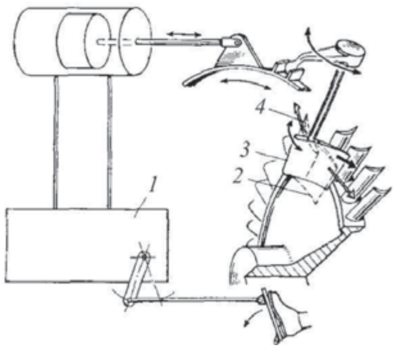


Fig. 2.38. Rotary nozzle device of a gas turbine:
1 – regulation system; 2 – vane position at low loads; 3 – vane position at full power; 4 – vane position during engine braking

The use of **nozzle devices of diaphragm design** (Fig. 2.37) allows for significantly reducing losses associated with the flow of the working fluid between stages, due to the possibility of installing the seal on a sufficiently small diameter.

To ensure higher turbine efficiency at partial operating modes, some gas turbine engines provide for the possibility of changing the angular position of the vanes in the nozzle device according to a certain program. An example is the rotating nozzle device of the gas turbine of the Chrysler automobile engine (Fig. 2.38).

2.2.2. Blades of axial turbines

The blades form convolutional (in the general case) channels (see Fig. 2.28), in which the kinetic energy of the gas is converted into mechanical work.

Due to the different angles of incidence of the flow on the blade profile along its length, the blades are made twisted, that is, they have a different orientation of the profile relative to the incident flow along the length (Fig. 2.39). The twist of the blades increases with increasing their length. For this reason, the blades of the last turbine stages (as the longest) are most twisted [16].

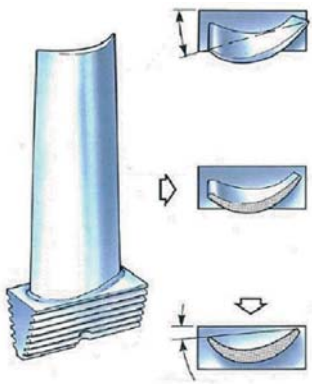


Fig. 2.39. Twisted turbine blade

Blades experience significant centrifugal and bending forces, and are also subject to vibration loads of various nature. To reduce the mass of the blades, tensile stresses and loads on the rotor, blades are made with a chord and thickness that decrease in length, so that the peripheral cross-sectional area can be 3-5 times smaller than the root cross-sectional area [15]. In this case, the rigidity of the profile part must be sufficient to maintain the angles of installation of the profiles in the most severe operating conditions of the blade.

Turbine blades, in comparison with compressor blades, operate at higher temperatures, experience significant static, vibrational and temperature stresses. They are simultaneously exposed to corrosive and erosive effects of gases. Considering the difficult operating conditions and their role in the engine (the reliability and resource of the blades largely determine the reliability and resource of the entire engine), particularly stringent requirements are imposed on the structural shapes of the blades and the methods of their fastening in the disk, on their materials and manufacturing technology.

Turbine blades have the same structural elements as compressor blades, but their geometric shapes and proportions are different. Turbine blades are characterized by thicker, more curved profiles and more developed transition parts and tails. For reasons of strength and rigidity, the blade tip is made with a more pronounced decrease in cross-sectional area from the root to the periphery.

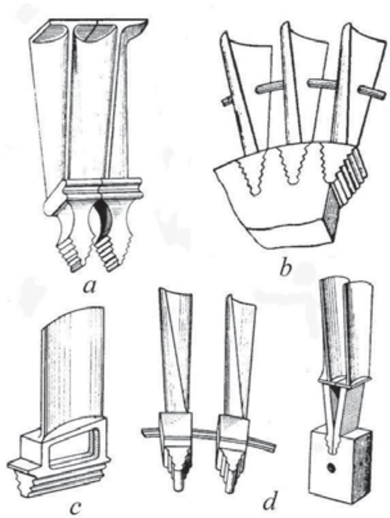


Fig. 2.40. Blades of turbines

To reduce losses due to flow through radial gaps, which is especially important for short blades, the blades are supplied with **banded shelves** (Figs. 2.40-2.42). Long blades are also supplied with banded shelves, but mainly to reduce stresses during vibration. Blades with shelves have thicker peripheral profiles. After the blades are installed on the disk, the shelves form a split-banded ring. A small gap (0.15-0.25 mm) is assumed between the shelves in the cold state. In the working state, the shelves are in contact with each other, while the blade tip of each blade is slightly untwisted under the action of centrifugal force. The shelves are joined together by oblique (see Fig. 2.41, *a*) or straight (see Fig. 2.41, *b*) cuts and, taking into account the significant thermal expansion of the blades, they are very rarely welded. For the same reasons, a band bandage is used extremely rarely, mainly in low-power engines with low temperatures.

On turbine blades, the feather of which is strongly twisted and undergoes significant untwisting under the action of centrifugal forces during turbine operation, **Z-shaped band shelves** are usually used (see Fig. 2.41, *a*). However, such shelves are characterized by increased crushing stresses on the contact surfaces, as well as the presence of stress concentrators in the radii of the "zigs". Therefore, for turbine blades, which have a relatively rigid feather, which is slightly prone to untwisting, it is more expedient to use band shelves with straight side ends of the contact surface (the so-called **zig-free shelves**) (see Fig. 2.41, *b*).

With very high spring stiffness, it is possible to use a zig-free band shelf with an additionally installed special damper, which rests on conical pads at the joints.

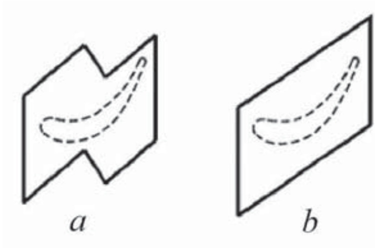


Fig. 2.41. Scheme of a blade with Z-shaped (*a*) and zig-free (*b*) band shelves

Banded shelves are often equipped with **sealing ridges** to reduce fluid leakage. The mass of the banded shelf can be up to 20% of the blade feather mass for the first stage, up to 15% for the second, and up to 10% for subsequent stages. Centrifugal forces from the mass of the banded shelves impart additional tensile loads to the blade.

Between the feather and the tail, the blades have a shelf that improves the transition from the feather to the tail. The combination of these shelves limits the flow part of the turbine wheel.

Sometimes the blades are made **with elongated tails** (legs) (Fig. 2.43). This design provides reliable insulation of the disk from hot gases and also provides damping of blade vibrations. Blowing the elongated tails with cooling air significantly reduces the temperature of the disk rim and allows the disk to be made thinner and lighter.

Fig. 2.42. Blades with band shelves

Fig. 2.43. Blades with elongated tails

To reduce the level of vibration, **box-shaped dampers** are sometimes installed between the tail shelves (see Fig. 2.40, *c*). In some designs, for the same purpose, **banding tubes (or wire)** are installed in the tails (Fig. 2.40, *d*) or in the profile parts of the blades (Fig. 2.40, *b*). When banding tubes are installed in the profile parts, losses in the turbine stage increase and the turbine efficiency decreases.

To improve vibration damping, **blades with elongated tails can be installed two in one groove** (Fig. 2.40, *d* and 2.44) (this design solution is often used by GE Vernova). In this case, the friction forces on the contact surfaces of the tails that arise during vibrations dampen these vibrations and reduce the stresses in the blades.

For the manufacture of blades, chromium-nickel alloys with additives of alloying components are used. The blanks for blades are usually obtained by stamping, necessarily maintaining the longitudinal arrangement of the material fibers, or by precision casting. With conventional stamping, allowances are left for mechanical processing (milling, etc.). Precision casting allows you to obtain a blade profile of the required shape and final size with an accuracy of up to 0.1 mm. Mechanical processing of the blade in this case is reduced only to polishing.

Fig. 2.44. Installing the two blades in one groove

Directional crystallization and the single-crystal structure of the turbine blade, which are provided during casting, allow for an increase in the working temperature of the blades by 80-120 K compared to blades with an equiaxed structure. At the same time, the gas temperature in front of the turbine blade of the first stage on modern engines can increase to 2050 K [15].

The advantage of blades obtained by directional crystallization, compared to conventional casting, is that the alignment of grain boundaries along the axis of acting stresses (directional structure) or the elimination of grain boundaries (monostructure) increases the plasticity of the material at elevated temperatures, and also increases the long-term strength of alloys. Controlling the directional crystallization process allows you to set the necessary orientation of the crystal lattice of the alloy and, due to this, significantly increase the resistance to thermal fatigue, which is fundamentally important for parts operating at elevated temperatures.

To ensure high turbine efficiency, the blades are subjected to careful control during production, which checks the accuracy of the relative location of the cross-sections along the length of the blade and the feather relative to the tail, as well as the accuracy of the cross-section profiles themselves and the cleanliness of the blade processing.

To facilitate the rotor balancing process, the blades are divided by mass and, when assembled, are selected in such a way that the deviation in mass of individual blades in the set does not exceed 5-10 g. Moreover, blades with a difference in mass of no more than 1-2 g are installed in opposite grooves of the rim.

The blades are monitored for their natural frequency of oscillation. If the frequency is below a set value, some material is removed from the end of the feather, or the rigidity of the blade fastening to the rotor is increased.

Axial fastening of gas turbine blades **with fir-tree tails** is currently widely used (Figs. 2.45 and 2.46).

Fig. 2.45. Turbine blades with fir-tree tails

The blade is installed in the groove of the disk using teeth located on the wedge leg of the blade and, accordingly, on the side surface of the wedge groove in the disk. The blade is held from moving along the groove on one side by a protrusion 1 (see Fig. 2.46, *a*), made during the manufacture of the blade, and on the other – by a bending antenna 3, which is shown in the position before bending. The movement of the blade in the groove of the disk after bending the antenna is 0.1-0.3 mm. Since during engine repair, the antenna may break, the blades are locked with a bending plate, which keeps them from moving in both directions (Fig. 2.46, *b*). Other methods of axial fixation of the blades are also used.

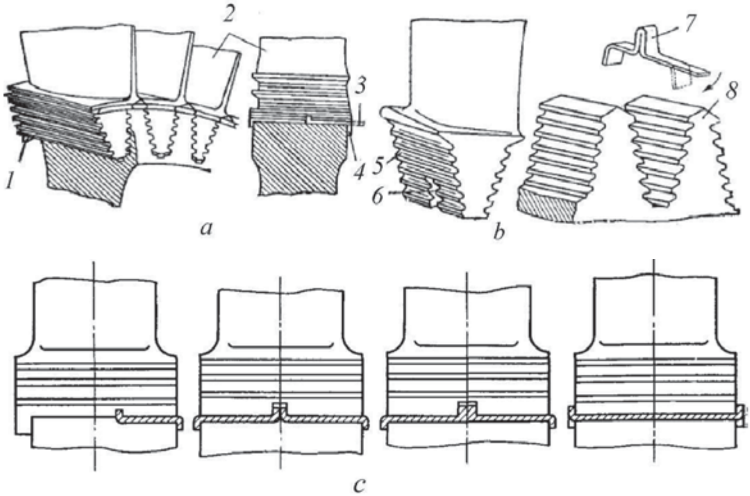


Fig. 2.46. Methods of attaching the turbine blades to the disk with the fir-tree tails and locking the blades from moving along the groove:

a – using the front locking protrusion and the antenna of the plate stopper; *b*, *c* – with the plate stopper; 1 – front locking protrusion; 2 – blade feather; 3 – antenna of the plate stopper in the unbent state; 4 – the same in the bent state; 5 – tail of the blade; 6 – groove for fixing the plate stopper; 7 – bending plate stopper; 8 – disk

The fir-tree fastening has become the best in practice for axial turbine blades. **The advantages of this fastening** are:

– the radial size of the disc rim is the smallest; the blades and disc with this fastening are the lightest;

– the small dimensions of the root part of the blade tail allow a significant number of blades to be placed on the disk;

– the loose fit of the blade does not prevent the thermal expansion of the rim, due to which the thermal stresses in the blade are eliminated and those in the disk rim are reduced;

– a loose fit allows the blade to self-align as the turbine rotates, so that bending stresses are minimal;

– due to the presence of friction forces in the fastening, the fit has good damping capacity;

– easy replacement of blades and cooling of the fastener by blowing air through the gaps in the lock is possible.

Along with these advantages, fir-tree fastening also has **disadvantages**:

– heat dissipation from the blade to the disk due to poor thermal contact between the tail and the groove is reduced;

– there is a significant concentration of stress in the teeth of the fastener, which can lead to the formation of fatigue cracks;

– to ensure uniform loading of the teeth, high precision of the fastener processing is required.

In recent decades, there has been a shift from a large number of teeth in a fir-tree fastening (seven to eight) to a smaller number (three to four). In the latter case, the teeth can be made larger and stress concentration in the fastening can be reduced.

Sometimes (usually in small turbines) the blades are made as one piece with the disk by precision casting or milling from forgings (**monowheel or "blisk"-type wheel**; abbreviation of "bladed disk") (Fig. 2.47). Modern blade machines allow for the necessary manufacturing accuracy and a high class of surface cleanliness. (It should be noted that "blisk"-type wheels are made not only for turbines, but also for compressors of gas turbine engines.)

Fig. 2.47. "Blisk"-type wheel

Cooling of the blades of gas turbines allows for maintaining their temperature within the limits, ensuring long-term mechanical strength. The use of cooling provides the possibility of increasing the temperature of the gases in front of the turbine, which increases the efficiency of the engine. In addition, the temperature of the cooled blades is equalized by volume in order to reduce thermal stresses and increase the engine life.

Various methods of cooling blades are used [19]. **Open-air cooling methods** have become widely used, in which air taken from the compressor part of the engine is used as a coolant. Air, when moving through the internal channels of the blade, cools it and then is discharged in one way or another into the flow part, where it mixes with the working fluid flowing there. The movement of air is carried out due to the pressure difference between the point of selection in the compressor and the point of discharge in the turbine, taking into account, of course, the aerodynamic resistance on its path. The first stages of gas turbines are most often cooled, the working fluid of which has the highest temperature.

It should be taken into account that the extraction of air from the compressor section, on the preliminary compression of which mechanical energy has already been spent, leads to a decrease in the power generated by the engine. Therefore, air extraction is attempted from stages that, on the one hand, guarantee the pressure drop necessary for movement, and on the other hand, are located as close as possible to the beginning of the compressor.

Currently, the main schemes for open-air cooling of turbine blades are:

– convective scheme;

– convective-film scheme;

– transpiration scheme with porous cooling.

In the convective scheme, the coolant (compressor air) moves through the internal channels of the blade, thereby cooling it. To increase the cooling efficiency, it is necessary to increase, firstly, the heat exchange surface area and, secondly, the heat transfer coefficient from the walls to the air. The latter is achieved by maximum turbulence of the coolant on its way.

The discharge into the flow part in a convective scheme is usually carried out in areas with relatively low pressure – in the area of the leading edge or through the upper end of the blade. Due to this, the convective scheme has a pressure reserve in the internal channels of the parts and can use significant aerodynamic resistance to intensify heat transfer on the internal surfaces of the blades.

In order to intensify heat exchange, the following design solutions are used: jet blowing, transverse finning, cylindrical pins, holes, swirling flows ("cyclonic cooling"), "vortex matrix", etc. The preferred type of heat exchange intensification is that which achieves the required level of heat removal with the lowest aerodynamic resistance.

In order to increase the heat exchange surface area, longitudinal ribs, multi-flow loop air flow schemes, and "vortex matrices" are used.

The design of a cooled turbine blade with a "vortex matrix" developed by Zorya-Mashproekt is shown in Fig. 2.48.

Fig. 2.49 shows a cast hollow turbine blade with internal ribs and an insert deflector (a perforated plate designed to evenly distribute cooling air over the inner surface of the blade), in which transverse air movement is implemented.

Fig. 2.50 shows a model of a cooled turbine blade of the Zorya-Mashproekt UGT 25000 engine, its casting core, as well as a device (box) for manufacturing this core.

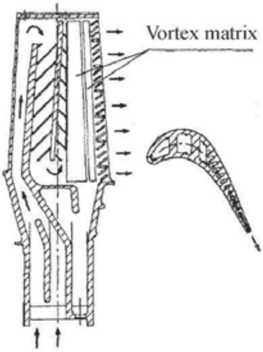


Fig. 2.48. Design of a cooled blade with a "vortex matrix" developed by the Zorya-Mashproekt

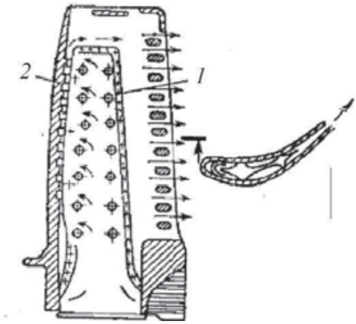


Fig. 2.49. Cooled blade with insert deflector (1), with air supply to the leading edge and transverse ribs (2) on the inner surface

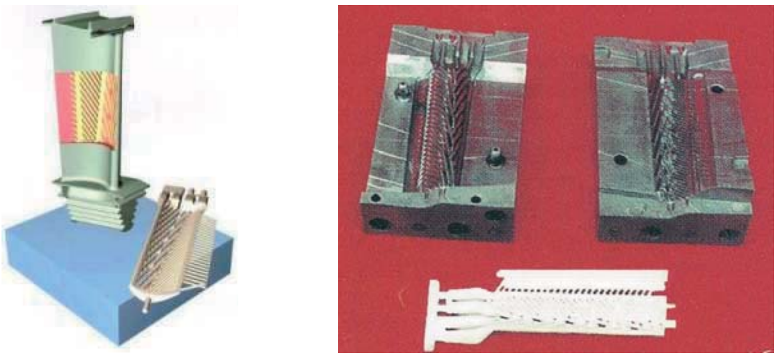


Fig. 2.50. Cooled turbine blade of the Zorya-Mashproekt UGT25000 engine, its casting core, and core box

In the convective-film scheme, the cooler (air) is used first in the internal convective cooling circuit, and then to organize a film (barrier) curtain around the outer surface of the blade (Fig. 2.51 and 2.52).

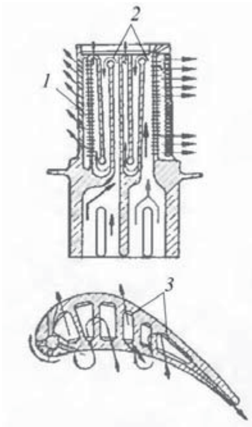


Fig. 2.51. Blade with convective-film cooling:

1 – hole for coolant release to the outer surface; 2 – channels with loop flow of coolant; 3 – ribs (heat transfer intensifiers)

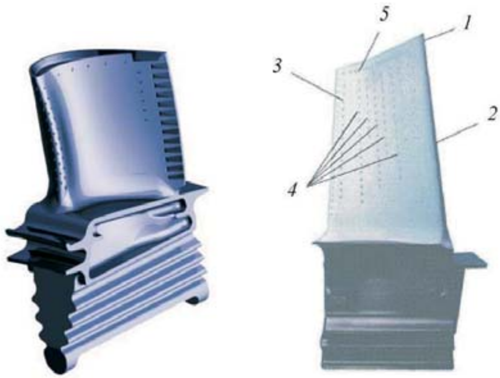


Fig. 2.52. Turbine blades with convective-film cooling:

1 – trailing edge; 2 – holes in the trailing edge; 3 – holes on the leading edge; 4 – holes on the trough; 5 – holes near the upper end of the blade

The film curtain is formed by blowing coolant from the inside of the blade outwards through rows of special holes. The protective effect is achieved due to the fact that the temperature of the curtain is significantly lower than the temperature of the main gas flow.

The most effective is film cooling when air is released onto the hottest surfaces – the leading edge, the concave surface of the blade, etc. Air release into these zones requires a fairly significant pressure reserve, since these zones are also, as a rule, zones with relatively high pressure.

Various shapes of coolant outlet channels are used - cylindrical, conical, slotted, and trapezoidal.

Factors that affect the efficiency of film cooling are:

– the shape of the channels (for technological reasons, holes with a diameter of 0.35-1.1 mm are most common);

– the distance between the axes of adjacent channels (for holes, this value is about three diameters);

– number of rows of holes (2-6 – for the backrest, 1-2 – for the shoulder blade trough).

It should be noted that the convective-film scheme currently provides the highest efficiency of cooling the blades at a given coolant flow rate. Despite its complexity and high manufacturing cost, it is widely used in the first stages of high-temperature gas turbines.

Fig. 2.53 shows the development of the cooling system for the high-pressure turbine blades of the Rolls-Royce Holdings plc from a single-pass convective scheme (Fig. 2.53, *a*) to a single-pass convective-film scheme (with a film on the leading edge and on a trough near the trailing edge) (Fig. 2.53, *b*) and then to a multi-pass convective scheme (Fig. 2.53, *c*) with intensive film cooling.

The transpiration (penetrating) cooling scheme is the most promising and should provide the greatest effect from creating a film curtain [22].

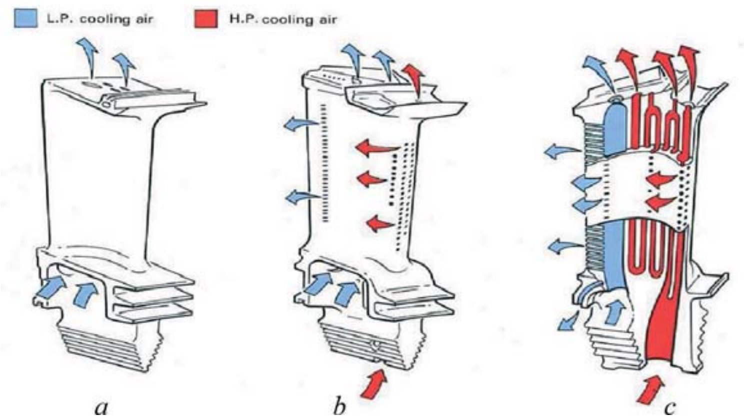


Fig. 2.53. Development of the blades cooling system of Rolls-Royce Holdings plc [10]

The blades in this case are made with porous walls. The air from the internal cavities of the blades, having carried out their convective cooling, penetrates through the pore system to the outer surface, creating a barrier film curtain. Unlike the convective-film scheme, such a curtain will be as uniform as possible. The effect of increasing the heat exchange surface will be as maximum as possible.

It should be noted that the implementation of this scheme in its classical sense is hardly possible due to manufacturing difficulties and the more than probable clogging of the holes in the walls of the blades. Therefore, simpler variants of quasi-porous cooling are currently being implemented, for example, the "lamella" scheme and the "blades with cooled walls" scheme.

In general, it should be noted that the creation of a successful design of cooled blades is associated with significant difficulties of both a constructive and technological nature.

Control tasks and questions

1. Describe the design of the blades of axial compressors.
2. What materials are used to manufacture axial compressor blades?
3. Explain the methods of connecting the blades of axial compressors to the rotor.
4. Describe the design of the guide vanes of axial compressors.
5. Explain the structure of the rotary guide vanes of compressors.
6. Describe the structure of the nozzle vanes of axial turbines.
7. Explain the design of axial turbine blades.
8. Analyze the methods of cooling turbine blades of gas turbine engines.

3. ROTORS OF AXIAL TURBOMACHINES

The rotor is a unit that combines all the rotating elements of a gas turbine engine. It is its main and most responsible part. The compressor rotor is used to compress air; the turbine rotor directly participates in obtaining mechanical rotational energy.

The design of the rotor, as well as the materials, calculation methods, manufacturing and assembly technologies used for its manufacture, must ensure its reliability under the influence of significant centrifugal, bending, and torsional loads, at high temperature and pressure differences, at nominal and partial operating modes, as well as during their rapid change.

Rotors of axial turbomachines (compressors and turbines), which are part of gas turbine engines, are characterized by a wide variety of design solutions. Typically, the rotor consists of a shaft, disks or a drum, blades, and some small parts fixed to the shaft or manufactured together with it: labyrinth sleeves, seals, drives of mounted mechanisms, couplings, oil-repelling rings, etc.

According to the manufacturing method, there are **solid and composite rotors**, and according to the design, there are **disc-type, drum-type, and disc-drum type (combined) rotors** [7].

3.1. Solid rotors

Solid rotors are manufactured by machining a single solid forging. They have a minimal number of parts attached to them (usually only small parts: seal and relief cavity bushings, oil shields, etc.), or none at all.

Advantages of solid rotors (compared to composite rotors):

– rational use of material;

– high reliability in all operating modes (the absence of disks connected by fasteners eliminates the possibility of imbalance or separation of rotor parts during operation);

– significant rigidity (this property ensures a high critical speed);

– less labor-intensive manufacturing (there are no complex machining operations with high accuracy of the seating of the parts to be connected, as well as rotor assembly and balancing operations).

Disadvantages of solid rotors (compared to composite rotors):

– difficulties in manufacturing large diameter rotors, which are explained by the difficulty of obtaining high-quality forgings of large sizes and their control, as well as the need for special powerful forging and pressing equipment (for this reason, the outer diameter of solid rotors, as a rule, does not exceed 1500 mm);

– higher material cost and worse consequences from production defects (it is necessary to choose expensive alloy steel as the forging material, which is only suitable for first-stage discs, while other elements could be made from cheaper steels; a defect in the processing or assembly of a solid rotor leads to immeasurably greater material losses and production delays than a defect in a separate disc);

– higher mass and thermal inertia.

Some researchers [7; 15] note that although solid rotors are less labor-intensive to manufacture, the production cycle of such rotors may be longer than that of composite rotors due to the impossibility of parallel machining of parts and their assembly.

Solid rotors are manufactured in two structural types: disc-type (Fig. 3.1) and drum-type (Fig. 3.2) rotors.

The main advantages of disc-type rotors (compared to drum-type rotors) are:

– high strength under the action of centrifugal forces;

– lower mass and thermal inertia (this is especially valuable during a sharp change in operating mode, as it causes lower thermal stresses in the rotor; this also has a positive effect on the magnitude of the relative thermal expansions of the rotor and housing);

– better conditions for rotor quality control during manufacturing.

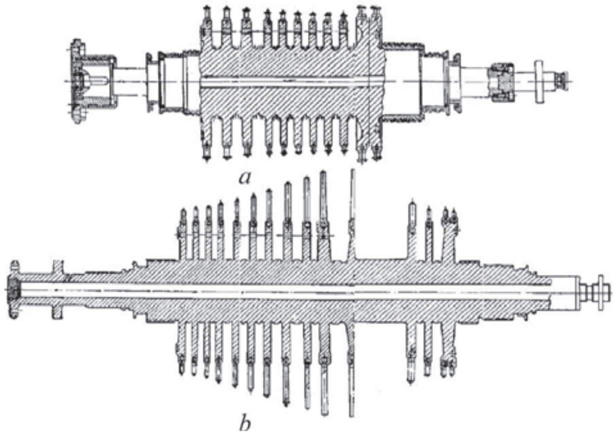


Fig. 3.1. Solid disk-type rotors of axial turbines [15]

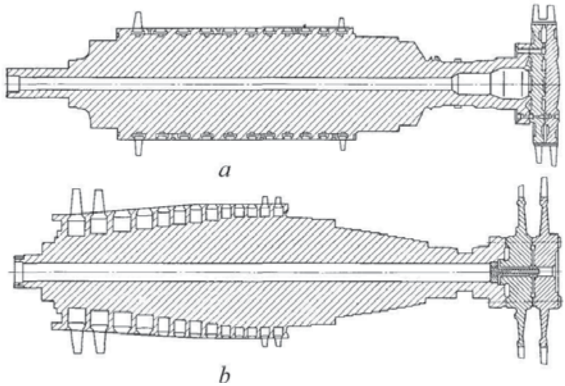


Fig. 3.2. Rotors of gas turbine engines (on the left side – solid drum-type rotors of axial compressors, on the right side – disk-type composite rotors of turbines) [7]:
a – rotor of the Power Machines (LMZ) GTN-9 engine; *b* – rotor of the English Electric EM-85 engine

The main advantages of drum-type rotors (compared to disc-type rotors) are:

– high transverse rigidity (for this reason, their critical speeds significantly exceed the operating ones);

– simpler design and less labor-intensive manufacturing.

It should be noted that when designing turbomachines, the disk-type rotor design is used in all cases unless other considerations require a different rotor design.

Solid rotors are often made with **a through central hole (boring)** (see Figs. 3.1 and 3.2). The presence of a central hole improves the final quality of the forging, as it has a positive effect on the uniformity of the distribution of mechanical characteristics across the cross section. For this reason, the hole is sought to be as large in diameter as possible.

The positive impact of the central hole on the reliability of the rotor is as follows:

– the less benign part of the metal located near the rotor axis, in which slag inclusions and other defects are most likely to be present, is removed;

– it is possible to inspect the inner surface of the forging using a special device;

– the volumetric stress state inherent in a solid rotor is reduced to a more favorable and more studied state.

However, it must be remembered that the presence of a central hole leads to an increase in the circumferential stresses acting in the rotor (in these rotors, they double).

For forgings with small outer diameters (up to approximately 400 mm), a central hole is not required, as they can be well forged and heat-treated without it.

Solid rotors, as a rule, are characterized by the fastening of the blades in grooves machined around the circle. In some cases (see Fig. 3.2, *b*), to reduce the mass of the rotor and reduce the stresses in its central part, deep grooves are provided in the grooves for fastening the blades.

Due to their advantages, solid rotors are widely used in steam turbines. In gas turbine engines, they are rarely used, as a rule, only in design schemes borrowed from the practice of steam turbine construction. (Some manufacturing companies that mass-produce steam turbines transfer their design and production experience to the gas turbine engines they manufacture.) However, many researchers note that the mass use of solid rotors in gas turbine units could significantly simplify and reduce the cost of their design [15].

3.2. Composite rotors

Composite rotors (prefabricated and welded) consist of separate, relatively small parts (shafts, discs, etc.) connected by fasteners (bolts, studs, pins) or by welding.

Advantages of composite rotors (compared to solid rotors):

– possibility of manufacturing rotors of significant diameter;

– lower cost (parts that are connected can be made of different materials: more loaded parts – from stronger and more expensive metals, less loaded parts – from cheaper ones);

– fewer consequences from production defects;

– lower (in most cases) mass and thermal inertia.

Disadvantages of composite rotors (compared to solid rotors):

– significant complexity of assembly and balancing;

– possibility of separation or imbalance of rotor parts during operation (for prefabricated rotors);

– presence of residual welding stresses (for welded rotors).

The main tasks that are solved when developing a composite rotor design:

– choosing a connection method (how to connect elements);

– choosing a centering method (how to center elements with each other);

– choosing a method of torque transmission (how to transmit torque from one element to another).

Composite rotors are distinguished by significant design diversity. Conditionally, according to their design, they can be classified into disk-type, disk-drum type, and drum-type rotors.

3.2.1. Disc-type composite rotors

A disk-type rotor is a set of individual disks, fixed in one way or another on a common shaft. The advantages and disadvantages of disk-type rotors have already been noted by us earlier.

In most discs, the following structural elements can be distinguished (Fig. 3.3): rim 1, hub 3, and middle part 2, which is called the disc body.

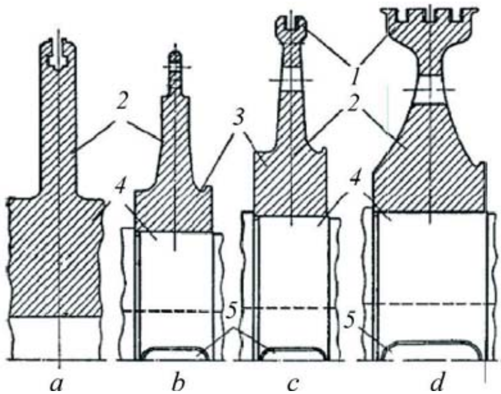


Fig. 3.3. Different types of discs for rotor with pressed-on discs [7]:
a – flat (straight) disc; *b* – flat-conical disc; *c* – conical disc; *d* – hyperbolic disc; 1 – disc rim; 2 – disc body; 3 – disc hub; 4 – shaft; 5 – key

The rim of the disk serves to fasten the blades. Its dimensions and configuration are determined by the adopted type of fastening, the external load from the blades, and the rotor speed. In the case of using a fir-tree or dovetail fastening, the axial size of the rim is small and, as a rule, does not exceed the thickness of the disk body.

The disk body usually has a complex configuration and includes elements for connecting the disks to the shaft, elements for fastening labyrinth seals, and parts of the cooling system. To reduce the mass of the disk, annular material samples can be made.

The disc profile is tried to be symmetrical. The side surfaces of the discs are usually polished to increase the fatigue resistance of the material. High requirements are placed on the quality of the blanks.

Gas turbine disc bodies are sometimes provided with **holes for the passage of cooling air**. However, because increased stresses may occur on the surface of the disc adjacent to the holes, holes should be used only when necessary, and the edges of the holes should be carefully rounded to the largest possible radius.

Discs with a central hole usually have **a hub** – a central part with a thickened profile. Its purpose is to reduce the circumferential stresses that occur on the surface of this hole. The hub is also often used in the rotor connection.

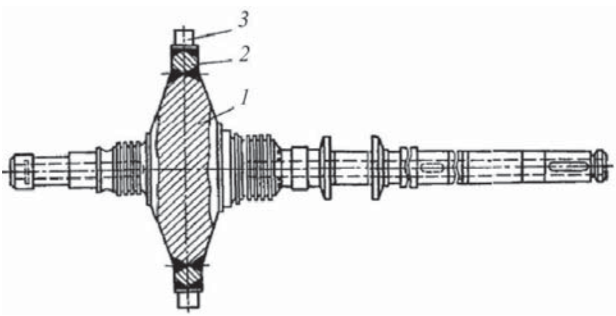


Fig. 3.4. Welded composite disc with welded blades:
1 – blades; 2 – austenitic steel rim; 3 – central part of the disk

In gas turbines, the so-called **welded composite discs** are used (previously practiced by GE Vernova), in which the rim is made of heat-resistant austenitic steel (or nickel-based alloy), and the central part is made of simpler pearlitic steel (Fig. 3.4). The heat-resistant rim steel is capable of operating at high temperatures, while pearlitic steel also has sufficient mechanical strength at temperatures below 820 K. The weld location is determined by calculating the limiting temperature for pearlitic materials (800-850 K) [7].

The shafts of composite disk-type rotors operate in relatively difficult conditions. They are subjected to static and dynamic loads of various magnitudes and natures. In addition, they experience the thermal effects of hot parts connected to them.

Like any other loaded element of a turbomachine, the shaft must be sufficiently strong, rigid, extremely light, and also simple to manufacture. The design of the shafts is determined by the type of engine, the adopted power scheme, and the number and location of the supports.

To ensure the required rigidity and simplicity with a small mass, the shafts are made hollow, with the largest possible outer diameter. With the same aim, attempts are made to reduce the distance between the supports and the length of the console, to make shafts with a smaller number of stress concentrators (slots, threaded sections, transverse bores, shoulders, grooves, transitions of cross sections, etc.) and at the same time to reduce the degree of stress concentration caused by them.

The shaft diameter is often limited by the allowable diameter of the bearings, the size of which, in turn, is limited by their speed limit.

Shafts can also be forged together with the discs.

Let's consider the main (by connection method) types of composite disk-type rotors.

Rotors with pressed-on discs

A rotor with pressed-on discs is a shaft of variable cross-section (usually hollow), forged from carbon or low-alloy steel, which carries discs made of high-alloy steel, which are fitted with tension (Fig. 3.5).

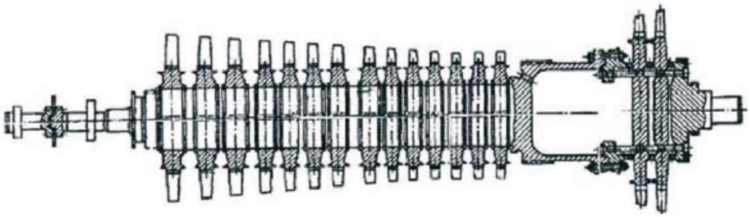


Fig. 3.5. Gas turbine engine rotor (compressor part – with pressed-on discs, turbine part – with peripheral tie bolts)

It should be remembered that under the influence of centrifugal forces acting on the disk and due to the temperature difference between the disk hub and the shaft, the fit in operating conditions weakens and a gap may appear, which will cause vibration of the rotor and its destruction. The approximate value of the tension is 0.001 times the shaft diameter. The difference between the maximum and minimum tensions is usually 0.05-0.08 mm.

Fitting the discs with tension does not eliminate the need to use additional elements that ensure the transmission of torque from the disc to the shaft, for example, keys (Fig. 3.6) – one or two for each disc.

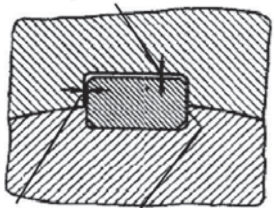


Fig. 3.6. Key for fastening the disk on the shaft

Fig. 3.7 shows the most common ways of mounting discs on the shaft.

Direct mounting of the disk on the shaft with tension is shown in Fig. 3.7, *a*. In this case, the mounting bore of the disk is made with a diameter slightly smaller than the diameter of the shaft. The disk is mounted on the shaft in a hot state, which creates the necessary tension after the disk has cooled. The disk is preheated in boiling water or an oil bath, after which it is pressed onto the cold shaft in a hot state. Longitudinal keys are placed on the shaft, which prevent the disk from turning during operation.

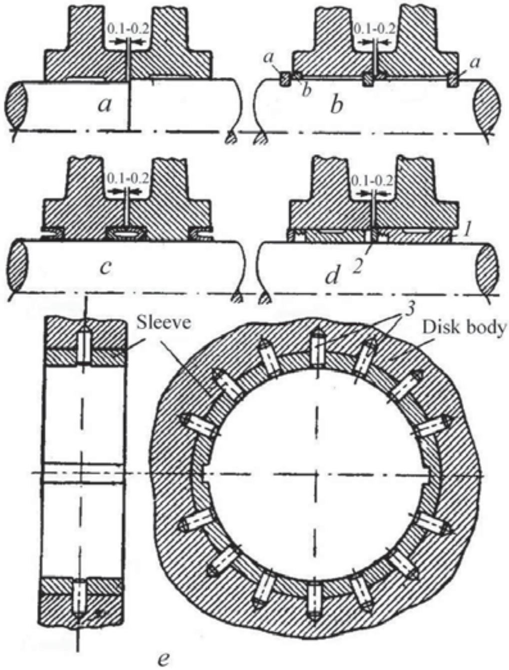


Fig. 3.7. Methods of mounting discs on the shaft

Fig. 3.7, *b* shows the method of mounting the disk on special rings. A split ring "a" is inserted into the annular recess on the shaft. A key is inserted into the ring slot and the longitudinal recess of the shaft. After prolonged heating in boiling water (2-3 hours), the disk is mounted on the shaft with one recess on the ring "a", and a ring "b" is inserted into the other recess, which has a stepped shape with a part protruding from under the disk by 0.1-0.2 mm. Thus, a gap of 0.1-0.2 mm is obtained between the disks, necessary for free axial expansion of the disk hubs during rotor heating.

Fig. 3.7, *c* shows the method of seating the discs on special spring rings of U-shaped cross-section. Due to their elastic properties, these rings are pressed on one side against the disc hub, and on the other side against the shaft.

Fig. 3.7, *d* shows the method of fitting the discs onto the split conical bushings 1. The disc is fitted onto the bushing using a hydraulic press. The length of the bushing is slightly shorter than the disc hub, which makes it possible to install a spacer ring 2 with a belt protruding from under the disc by approximately 0.1 mm.

Fig. 3.7, *e* shows the landing of the disk on the finger sleeve. A finger sleeve (ring) is pressed into the disk body. Radial holes 3 are drilled in the ring together with the disk, into which fingers (radial pins) are inserted. The disk is also mounted on the shaft in a hot state, which ensures tension. The finger sleeve is held from turning by two longitudinal keys. The method is quite reliable and has become widespread for rotors operating in the high-temperature range.

According to the body profile, the discs used in these rotors can be classified as follows.

The simplest (from the point of view of manufacturing) is a flat (straight) disk, the profile of which is a rectangle (see Fig. 3.3, *a*), which can be used with small loads and small (up to 120 m/s) circular speeds on the periphery.

A flat-conical disk (see Fig. 3.3, *b*) is also quite simple to manufacture, but provides a more uniform distribution of stresses in the body and therefore can be used at higher circular speeds.

A conical disk, the profile of which between the so-called "neck" (see Fig. 3.3, *c*) and the outer boundary of the hub is a trapezoid, is more technologically complex, but can be used at significant (up to 300-400 m/s) circular speeds.

The profile of the body of a hyperbolic disk (see Fig. 3.3, *d*) is outlined by a hyperbola. Processing of such disks requires special molding devices and is quite complex. The strength characteristics of hyperbolic disks are high, but due to the difficulties of manufacturing and high cost, they have not gained much popularity. They are used for multi-crown turbine stages.

The advantage of rotors with pressed-on discs is ease of manufacturing, assembly, and balancing.

Rotors of this type find some application in axial compressors of gas turbine engines. In gas turbines, which are characterized by significant temperature differences (and consequently significant thermal stresses), the fit of the disks onto the shaft with tension is undesirable and is used very rarely. Such rotors have found quite wide application in steam turbines of small and medium power, which operate on steam with a temperature of up to 650 K.

Rotors with tie bolts

The connection of individual disks to the tail journals using axial tie bolts – one central or several peripheral – has become widespread in axial compressors and gas turbines.

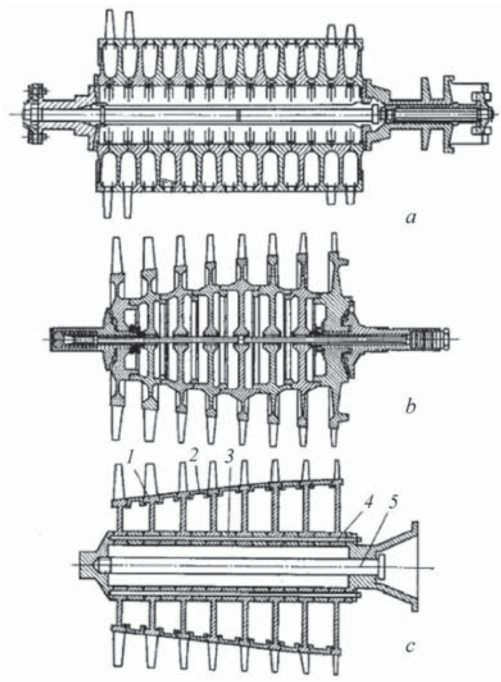


Fig. 3.8. Disc-type and disc-drum type composite rotors of axial compressors with a central tie bolt [15]

If there is **one central tie bolt** (Fig. 3.8-3.10), it is necessary to provide special elements for torque transmission. Thus, in the rotors in Figures 3.8, *a*, 3.10, *b* and 3.10, *c*, the torque is transmitted by axial pins in the disk hubs.

Fig. 3.9. Installing the center tie bolt on a GE Aerospace T58 engine

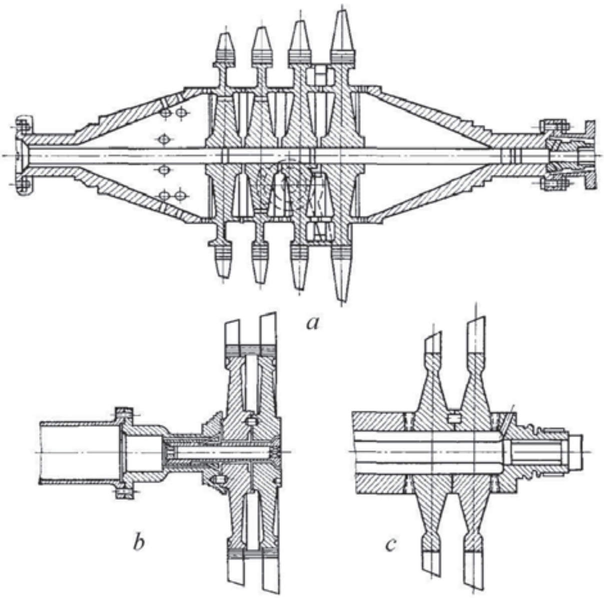


Fig. 3.10. Disc-type composite turbine rotors with a central tie bolt [15]

In the rotor in Fig. 3.8, *b*, the torque is transmitted by friction forces along the disk landing belts. The disks are also connected by radial locking bolts.

The central bolt 5 tightens the compressor rotor in Fig. 3.8, *c*. The torque is transmitted through the keys between the disks 1 and the inner spacer rings 3, as well as by friction between the disks and the outer rings 2. The peripheral bolts 4 are technological; they facilitate the assembly process.

End splines of various configurations are also widely used for torque transmission (Fig. 3.10, *a*). The most commonly used are end triangular splines (**Hirt-type connections**) (Figs. 3.11 and 3.12). End splines of curved (involute) shape are also widely used - their teeth have a barrel-shaped configuration of the "hourglass" type (Fig. 3.13) [15].

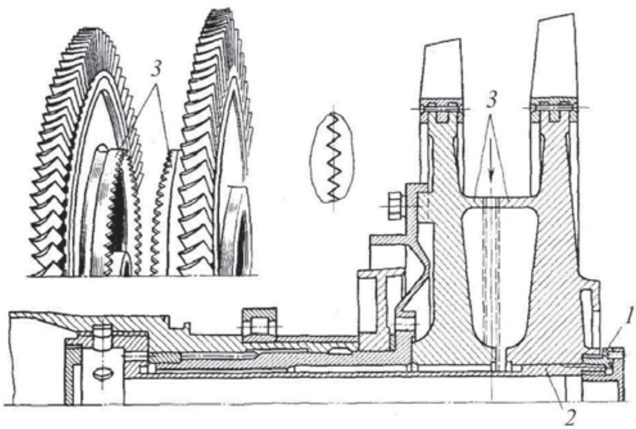


Fig. 3.11. Rotor with central tie bolt and torque transmission using triangular end splines:

1 – nut; 2 – tie bolt; 3 – end splines

The central tie bolt takes on significant loads due to its pre-tensioning and the possible temperature difference between the discs and the bolt. To avoid loosening of the discs during operation, the central bolt, which is made of high-strength alloy steel, is installed with significant pre-tensioning, and in some cases spring elements are inserted under its head and under the nut (see, for example, Fig. 3.8, *b*).

Fig. 3.12. Turbine disk with triangular end splines

Fig. 3.13. GE Aerospace turbine disk with end curved splines

To avoid resonant oscillations and deflection, the central tie bolt rests in one or more places on a corresponding seat in the disk. With a significant diameter of the internal bore of the rotor, one of the disks is made with a small hole specifically to support the central bolt.

The scheme in which the disks are connected **by peripheral bolts** is more often used (Figs. 3.14 and 3.15). This makes it possible to manufacture disks without a central hole to reduce the level of stress in their body. In addition, connecting the disks by peripheral bolts is the only possible type of assembly of disk rotors, through which the shaft of another turbocompressor or power turbine passes.

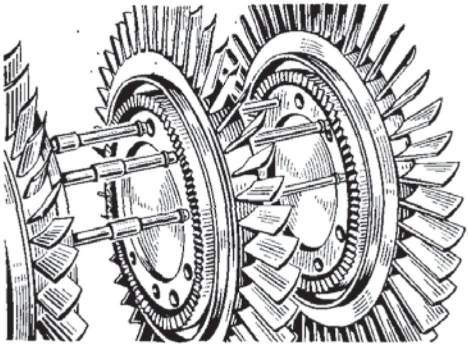


Fig. 3.14. Rotor with peripheral tie bolts and torque transmission using triangular end splines

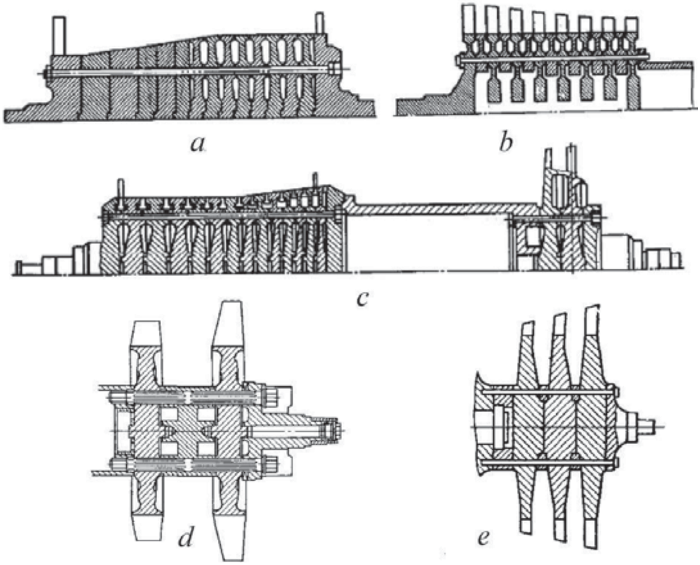


Fig. 3.15. Disc-type composite rotors with peripheral tie bolts [15]

Most often, discs in rotors of this design are made without central holes, with a constant thickness or conical shape. Since the loads from the blades are relatively insignificant, and the rigidity of such rotors is considerable, the discs can be manufactured without lightening recesses.

Mutual centering of the discs is performed both by small-diameter grindings in the central part and by belts in the peripheral shelves. Discs are sometimes centered using spacer sleeves, which also serve to transmit torque.

Peripheral bolts are not intended for torque transmission. Other parts are used for this purpose in the rotor. The most commonly used are end splines of various configurations (such as Hirt and others) (see Fig. 3.11-3.14).

Peripheral bolts withstand significant loads from tightening, as well as bending loads caused by centrifugal forces and different radial movements of adjacent disks. This imposes strict requirements on the thermal state of the rotor – the cooling system must ensure close temperatures of different disks at their junctions. Uneven tightening of peripheral bolts can lead to deflection of the rotor and the occurrence of vibrations. To prevent excessive deflection of peripheral bolts, the gap between the bolts and the holes in the disks is chosen to be minimal, or several support strips are provided along the length of the bolt. It should also be noted that the holes for the bolts weaken the disk body and create stress concentrations in these places.

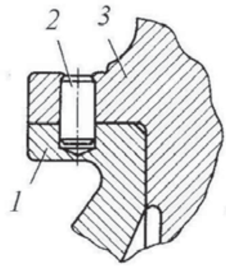


Fig. 3.16. Connection using radial pins:
1 – shaft; 2 – radial pin; 3 – disk

Rotors with radial pins

The non-separable method of connecting the disks to each other and to the shaft using smooth radial pins is considered advantageous in terms of weight, rigidity, ability to operate at high temperatures while maintaining centering, and relatively low technological complexity (Fig. 3.16-3.19).

The torque in such rotors is transmitted by friction on the surfaces that are connected with tension and by radial pins that work on shear. The dimensions of the pins are chosen such that even in the absence of friction, shear resistance is ensured [15]. (Friction on the connected surfaces may cease after a certain period of operation due to uneven heating of the connected parts and local plastic deformation of these surfaces.)

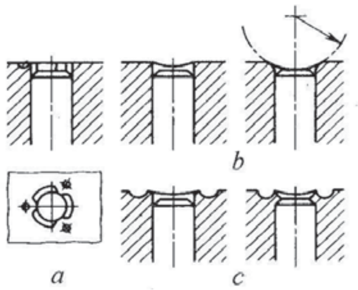


Fig. 3.17. Radial pin fixing options: *a* – by coreing; *b* – by pressing with a ball or spherical surface; *c* – by rolling a cylindrical strip of small thickness

In the diagram in Fig. 3.18, *a*, a force ring 3 is used for fastening, which is installed on the annular flanges of the disks under tension and fastened to them with pins. The pins are inserted with a tension of 0.2-0.4 mm into the holes machined together in the fastening elements of the connection unit. Such a connection is characterized by significant strength and rigidity (connection on a significant radius, tension on the landing surfaces, and radial pins) and is relatively simple to manufacture. It eliminates the weakening of the disk by the mounting holes.

It is possible to connect the disks to each other without the help of a power ring by means of developed annular shoulders.

A mandatory condition in the connection under consideration is the radial position of the pins. For example, according to the standards of the Nevsky Plant, the deviation of the pin axis from the radial position by more than 0.02 mm at the depth of the hole with a disk landing diameter of about 200 mm is not allowed [15].

To protect the pins from falling out under the action of centrifugal forces, various methods of fixing are provided. For example, pin 4 in Fig. 3.19 is fixed with an axial locking pin 2; pin 6 is fixed by stamping the material along the edges of the holes; pin 9, installed inside the belt, has a head. Often, for the purpose of locking, special rings, labyrinth sleeves, etc., are provided above the pins. In all cases, it is necessary to make a chamfer or groove on the pin. Their presence reduces the risk of cracks and tears in the material during its settlement. Various options for fixing by creating plastic deformations are shown in Fig. 3.17.

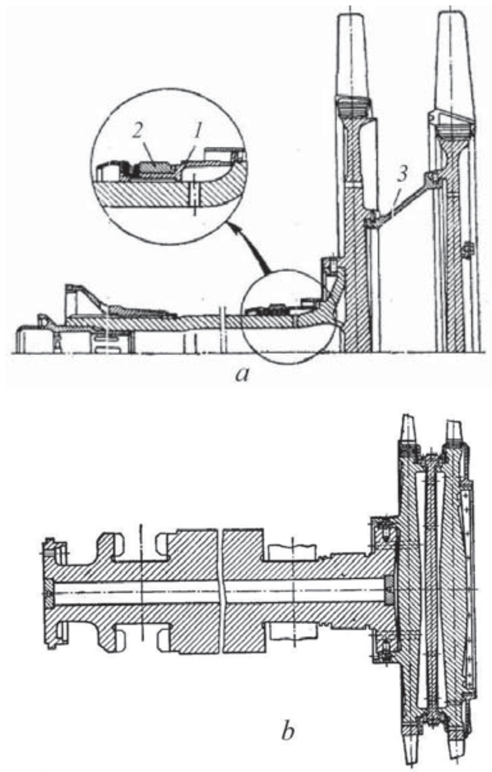


Fig. 3.18. Turbine rotors with radial pins [7]:

1 – intermediate sleeve (thermal choke); 2 – inner bearing cage; 3 – power ring

Fastening using radial pins is often used to attach cantilever disks (see Fig. 3.18, *b*).

The disadvantage of the connection is that when the disks are heated and pulled out, the tension on the joint surfaces decreases, and with prolonged operation, a gap forms, which leads to a decrease in the rigidity of the structure.

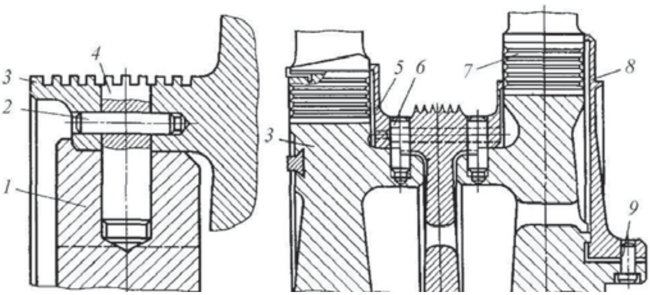


Fig. 3.19. Rotor connection using radial pins [15]:
1 – shaft; 2 – locking pin; 3 – disc; 4, 6, 9 – radial pins; 5 – intermediate disc; 7 – blade; 8 – cover disc

Rotors with flange connection

A flange connection, i.e., a connection using flanges or collars on the shaft and disk, is widely used in gas turbine engine rotors. In such a design, the transmission of torque and axial force from the disk to the shaft can be carried out in different ways.

Types of flange connections of rotors are shown in Figs. 3.20 and 3.21. Fastening can be carried out using screws (see Fig. 3.20, *a*), bolts (see Fig. 3.20, *c* and *d*) or studs (see Fig. 3.20, *b*). Centering is ensured by special belts, which create tension (see Fig. 3.20, *c*), bushings (see Fig. 3.20, *b*), or bolts (see Fig. 3.21).

A flange connection is a type of collapsible connection. It is relatively easy to manufacture and provides ease of assembly and disassembly, but it has several significant disadvantages, such as significant heat dissipation from the disk to the shaft, weakening of the disk by holes, and weakening of the fastening due to stretching of screws (bolts) operating at high temperatures.

The weakening of the disk body by the holes for screws or bolts is compensated by local thickening of the disk, which, however, leads to an increase in its mass. Often, to eliminate the weakening of the disk by the holes, a flange is made on it (see Fig. 3.20, *c* and 3.21), by means of which the disk is connected to the shaft.

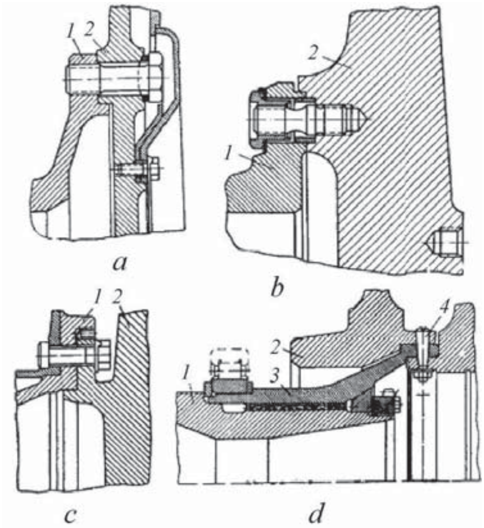


Fig. 3.20. Types of rotor flange connections [7]:

a – using screws; *b* – using pins and spacer sleeves; *c* – using radial splines and bolts; *d* – using an intermediate spline shank and radial conical bolts; 1 – shaft; 2 – disk; 3 – shank; 4 – conical bolt; 5 – nut

Torque transmission is carried out:

– due to friction on the contact surfaces of the disk with the shaft flange (see Fig. 3.20, *a*);

– due to the shearing action of the zone sleeves (see Fig. 3.20, *b*) or bolts (see Fig. 3.21);

– radial or end splines.

When transmitting torque by friction, to reduce the required tightening of screws (bolts), the connection of the disk with the shaft is placed at the largest possible radius, with a significant contact surface of the connected parts.

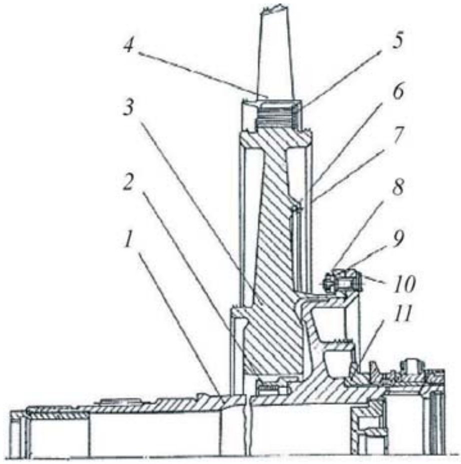


Fig. 3.21. Low-pressure turbine rotor of the Zorya-Mashproekt UGT15000 engine [15]:

1 – turbine shaft; 2 – seal sleeve; 3 – disk; 4 – blade; 5 – segment; 6 – retainer; 7 – balancing weight; 8 – nut; 9 – washer; 10 – bolt; 11 – sleeve

Fig. 3.20, *c* shows a flange connection of the disk with the shaft, in which the torque is transmitted by involute radial splines, and the axial force by bolts that work in tension. Reliable centering of the disk is ensured by the tension on the centering collar, which increases when the disk is heated and stretched. However, the seating stresses increase significantly.

Reliable centering of the disc in hot and cold conditions is ensured by torque transmission using end splines.

In the design shown in Fig. 3.20, *d*, the connection of the disk 2 with the shaft 1 is carried out using an intermediate element - a splined shank 3. The disk is connected to the shank using conical bolts 4. The transmission of torque and axial force is provided by friction forces and the work of conical bolts on the cut. The shank is connected to the shaft via splines, which transmit torque to the shaft. The axial fixation of the shank on the shaft is carried out by a nut 5. The splined connection of the shank with the shaft helps to reduce heat transfer from the disk to the shaft.

3.2.2. Disc-drum type composite rotors

The positive qualities of the drum-type (high rigidity) and disc-type (high strength) rotors are combined in the disc-drum type rotor (Figs. 3.22-3.27). It is a set of disks with cylindrical shoulders or spacers developed on the periphery, which form drum sections. These sections, connecting with each other or with the disks, transmit torque. Conical sleeves are attached to the front and rear disks.

The choice of material and shape of the discs is not limited by any special technological or design requirements. Therefore, the composite disc-drum type rotor can be made as strong and light as possible.

Unlike disk-type rotors, this rotor has high bending stiffness and, therefore, a significant critical speed.

Disk-drum type rotors are widely used in axial compressors of gas turbine engines (the multi-stage nature of compressor rotors leads to an increase in their length and the distance between supports, which negatively affects bending stiffness); they are also often used as part of gas turbines.

There are the following **ways to connect the parts of a disc-drum type rotor**:

– by pressing and radial pins;

– by tie bolts;

– by flange connection under bolt;

– by welding.

In the first case, the drum part of one disk is pressed onto the rim of the other (see Fig. 3.22, *a*), for which the first disk is heated to 100-150 °C. After pressing, radial holes are drilled and turned, into which the pins that transmit torque are installed under tension. The significant diameter of the drum provides high strength and rigidity of the rotor and allows you to place a large number of small-diameter pins. Even with significant rotor diameters, the thickness of the disk and drum walls is small. The mass of such a rotor does not exceed the mass of a drum-type rotor.

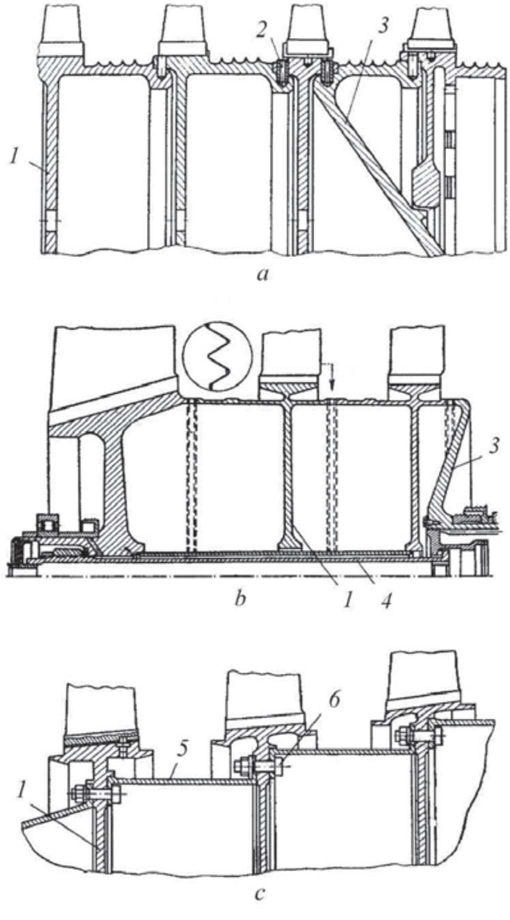


Fig. 3.22. Disc-drum type rotors [15]:

a – connection with radial pins; *b* – connection with a central tie bolt; *c* – flange connection of disks; 1 – disk; 2 – radial pin; 3 – trunnion; 4 – tie bolt; 5 – intermediate ring; 6 – zone bolt

The disadvantages of this connection method are:

– difficulties in manufacturing discs with developed shoulders;

– difficulty of disassembling the rotor after pressing the discs.

Rotors with parts connected **by means of tie bolts** do not have the latter drawback (see Fig. 3.22, *b*, as well as Fig. 3.8, 3.14 and 3.15). End splines are often used in such rotors to transmit torque and to center the parts relative to each other. (The design features of rotors with tie bolts were discussed earlier in chapter 3.2.1.)

Fig. 3.23 shows the rotor of the SGT5-2000E (V94) engine, typical of Siemens Energy [7].

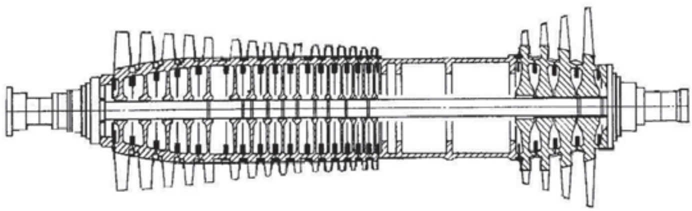


Fig. 3.23. Rotor of the Siemens Energy SGT5-2000E engine

The rotor is a two-bearing, disk-drum type, consisting of disks carrying the blades, two end parts, and an intermediate (between the compressor and the turbine) drum-type insert. The disks of the compressor and turbine rotors have a central hole, are mutually centered by triangular end splines, and are tightened by a central bolt using a special hydraulic device. The force created by the device is easily controlled and, therefore, can always be maintained at the same. This method of tightening the end nuts is convenient for repair work performed directly at the place of operation. The tightening bolt rests on the disks in several places to prevent vibration.

Fig. 3.24 shows the turbine rotor of the GE Vernova MS7001B engine [7], the connection of the parts of which is carried out using peripheral tie bolts.

Between the main disks, which carry the blades, intermediate disks are installed, with the help of which the nozzle devices are sealed and which form the drum part of the rotor.

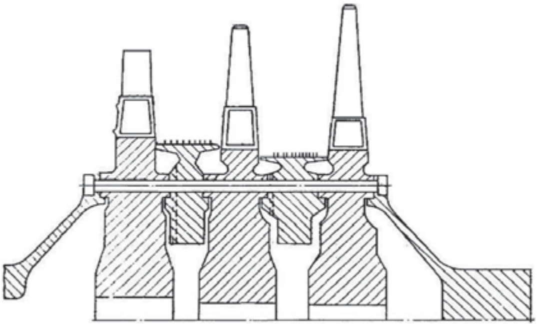


Fig. 3.24. Turbine rotor of the GE Vernova MS7001B engine

In rotors **with a flange bolted connection** (see Fig. 3.22, *c*), torque transmission is carried out by zone bolts or friction on the mating surfaces. Centering is carried out again by means of zone bolts or annular collars.

As the diameters of the discs increase, the diameters of the intermediate rings also increase, which leads to a staggered arrangement of the bolts. Such a rotor has a low mass, is easy to manufacture, and convenient to assemble.

In disk-drum type rotors, **trunnions (conical or curly)** with short shafts for installing bearings are installed at the ends (see Fig. 3.22, *a* and *b*). Steel trunnions are manufactured together with the shaft; in trunnions made of aluminum or titanium alloys, the shafts are pressed in and, in addition, they are connected to the trunnions by means of radial pins. The latter provides centering and torque transmission when the tension is weakened as a result of heating the wall or its elastic deformation under the action of centrifugal forces.

The trunnions are usually attached to the disks in the same way as the disks are connected to each other, i.e., by radial pins, tie bolts, or flanges. Increasing the taper of the trunnions (the angle at the base) increases the bending stiffness of the rotor and reduces the likelihood of joint divergence at the disk joints.

There are two types of trunnions: straight taper trunnions (see Fig. 3.22, *a*) and reverse taper ones (see Fig. 3.22, *b*). As noted in [14], with a taper of 45 to 75°, trunnions of the first type have approximately twice the stiffness of trunnions of the second type. The advantage of reverse taper trunnions is the increased tightening of the central tie bolt under the action of centrifugal forces in rotors of the corresponding type. In addition, the use of reverse taper trunnions allows you to reduce the length of the rotor and thereby reduce the size and weight of the gas turbine unit as a whole.

The rotor discs, trunnions, shafts and drums are machined with a high degree of precision and are statically balanced. The assembled rotor (with attached blades) is dynamically balanced.

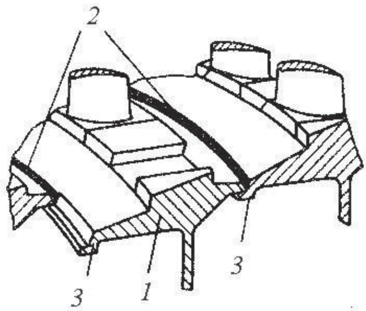


Fig. 3.25. Connecting rotor sections by welding:
1 – rotor section; 2 – joint; 3 – centering belt

The high level of modern welding technology allows **the use of welding methods in the manufacture of turbomachine rotors** (Fig. 3.25). Welding in gas turbine engineering is one of the leading processes that allows for significantly increasing the metal utilization rate, reducing metal consumption, labor intensity, and increasing the reliability and performance of manufactured products [2].

Welded disc-drum-type rotors consist of several discs, usually of equal resistance, connected by annular welds around the periphery of expanded rims, the outer surfaces of which form a drum on which the blades are mounted (Fig. 3.26).

Fig. 3.27 shows a typical design of the rotor of Alstom gas turbine engines. The rotor is welded and consists of 5-12 compressor disks, a tail unit, an intermediate drum part of the rotor, and two or three turbine disks. The rotor is welded from stamped disks forged from ferritic steel (12% Cr, also containing Ni, Mo, V), in two stages: the internal joint is welded in an argon environment, and the rest of the joints are under a layer of flux. The welded rotor provides high vibration stability, since it has no detachable joints. Due to the high rigidity of the structure, the critical number of revolutions of the rotor is significantly higher than the working one.

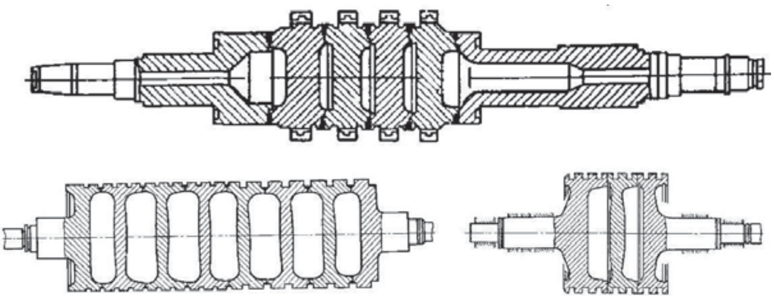


Fig. 3.26. Welded disc-drum type rotors

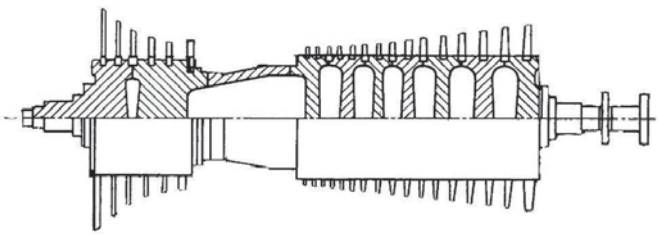


Fig. 3.27. Typical rotor design of Alstom (Brown-Bovery) engines

It should be said that, despite significant advantages, the manufacture of welded rotors has a number of difficulties of a metallurgical, technological, and design nature. For the manufacture of parts of such rotors, steel is required that combines sufficiently high elastic and plastic characteristics with high-quality weldability. The design of the joint and welding technology must ensure the production of a high-quality welded joint, despite the significant (from the welding point of view) thickness of the parts being joined. It is necessary to use sophisticated means of quality control of the welded joint.

In complex engine schemes, **the connection of rotor parts can be carried out simultaneously in several ways**. As an example, consider the rotor of the low-pressure compressor of the Zorya-Mashproekt UGT15000 engine (Fig. 3.28).

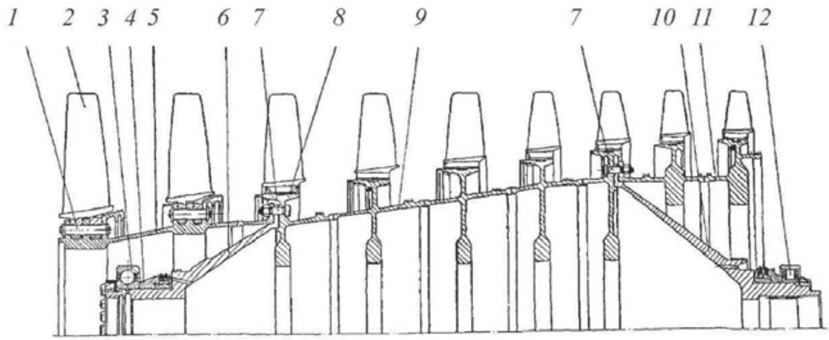


Fig. 3.28. Rotor of the UGT15000 low-pressure compressor [15]:
1 – finger; 2 – blades; 3 – front support; 4 – oil deflector; 5 – drum of 1-2 stages; 6 – front tunnion; 7 – bolt; 8 – lock; 9 – drum of 3-7 stages; 10 – rear trunnion; 11 – drum of 8-9 stages; 12 – rear support

The rotor of the 9-stage compressor is a two-bearing, disk-drum type design, consisting of a front trunnion 6, a rear trunnion 10, blades 2, and disks connected by electron beam welding into three integral units: drum 5 of the first-second stages, drum 9 of the third-seventh stages, drum 11 of the eighth-ninth stages. The drums are connected and to the trunnions by means of flanges and bolts 7. Disks of the 1st, 2nd, 8th, and 9th stages are cantilevered on the rotor.

3.2.3. Drum-type composite rotors

The drum-type composite rotor is a thin-walled cylinder or truncated cone made of steel or aluminum alloy, on which the blades are fixed. The two covers that cover the drum from the ends have trunnions on which the rotor rests on the bearings (Figs. 3.29 and 3.30).

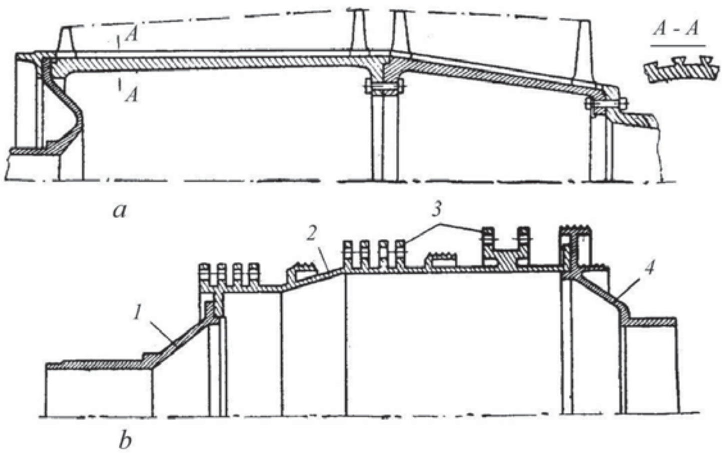


Fig. 3.29. Composite drum-type rotors:

a – rotor with longitudinal grooves, *b* – rotor of a low-pressure compressor of a two-spool gas turbine engine; 1 – front trunnion; 2 – drum; 3 – attachment points of the blades; 4 – rear trunnion

The individual parts of such rotors are connected by means of flanges (bolts or studs) or radial pins, and devices are used to accurately center the parts and ensure freedom of thermal movement of the connected parts. It is also possible to manufacture a composite drum-type rotor from machined steel forgings by butt-welding them together.

On the outer surface of the drum, annular or longitudinal (see Fig. 3.29, *a*) grooves are cut for fastening the blades. **With longitudinal grooves**, the number of blades for all stages is the same and equal to the number of such grooves. With the same mass, a drum with longitudinal grooves is less durable than one with annular ones. However, manufacturing a drum with longitudinal grooves is cheaper, and fastening the blades is much simpler. The distance between the blades is maintained using spacers that are installed in the grooves.

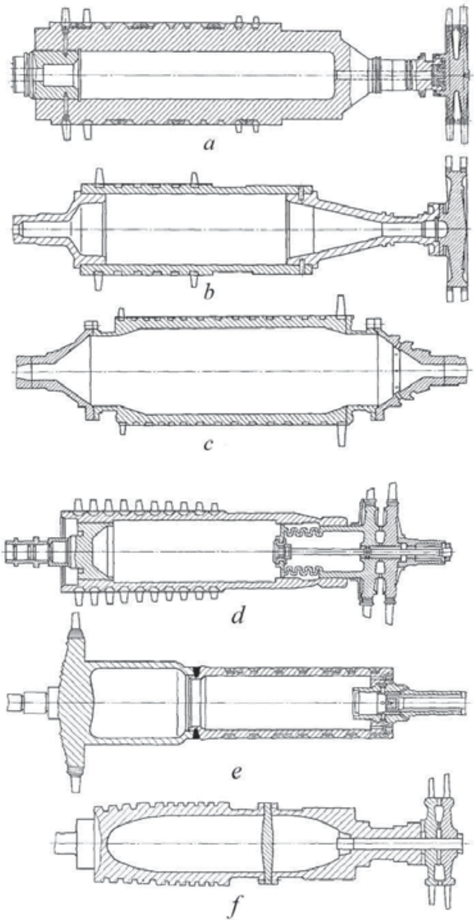


Fig. 3.30. Drum-type rotors of axial compressors [7]

The annular grooves allow for a different number of blades to be installed for each stage, but attaching the blades and replacing them during repairs is quite complicated.

The main advantages of composite drum-type rotors are simplicity of design and manufacture, low mass, and significant transverse rigidity, which provides a sufficiently high critical speed.

The disadvantage of the drum-type rotor is its low strength under the action of centrifugal forces. It is allowed to use it only in low-speed turbomachines with a peripheral speed of up to 200 m/s.

Composite drum-type rotors find some application in axial compressors of gas turbine engines. However, due to the low rotational speeds, the pressure of the stages of such compressors is not high, which causes an increase in the total number of stages, an increase in the length and mass of the rotor. In addition, low-speed compressors are also disadvantageous for the turbines connected to them, since they cause an increase in their diameter.

Composite drum-type rotors are currently used mainly in low-speed steam turbines with small initial steam parameters or in the absence of special requirements for the maneuverability of the unit.

The considered designs of gas turbine engine rotors do not exhaust all possible variants of their execution. In each specific case, the rotor design must be coordinated with the general scheme of the engine, the number and location of supports, the type of power housing, as well as the operating loads.

3.3. Connection of turbomachine rotors

The rotors of turbomachines are connected to a gearbox or to a torque consumer (for example, to a generator) using couplings. The couplings must transmit torque, and in some designs (with a common thrust bearing) also axial force. If each of the connected rotors has its own thrust bearing, then it is necessary that the coupling allows independent axial displacement of the shafts. In this case, it is desirable that the vibration of one rotor is transmitted to the other through the coupling and that the design of the coupling allows for some eccentricity of the connected shafts, as well as deviation of their axes from the common axis of the rotor.

The following **main types of couplings** can be distinguished: rigid (flange), flexible, and elastic couplings.

In Fig. 3.31, two different versions of **a rigid (flange) coupling** are shown up and down from the center line [15]: coupling halves 1 and 2 are connected by bolts installed with a gap (option I) or without a gap (option II).

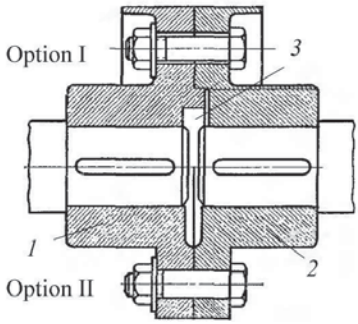


Fig. 3.31. Rigid (flange) coupling

In the first case, the torque is transmitted due to the friction forces that arise at the joint of the coupling halves from the tightening of the bolts; in the second case – directly by the bolts that work in shear.

Bolts installed without a gap can simultaneously perform the function of centering the shafts. When installing bolts with a gap, centering is performed by an annular protrusion (shoulder) 3.

Installing bolts without a gap allows for smaller couplings and is therefore more common. The holes for the connecting bolts in this case are machined by reaming and the bolts are fitted tightly into the holes.

The advantage of flange couplings is the simplicity of the design and relatively small dimensions. The main disadvantage is the transfer of vibration of one rotor to the bearings of another. In addition, the coupling itself can serve as a source of vibration due to its insufficiently accurate centering.

Flange couplings are widely used for various types of turbomachinery, most often for connecting the shaft of a small- or medium-power turbine with the shaft of a generator. This makes it possible to install the turbine rotor and the generator rotor on three bearings. Flange couplings are also used to connect the rotors of two- and three-cased high-power steam turbines in order to use one thrust and three support bearings on two rotors, which makes it possible to reduce the total length of the turbomachine.

Flange couplings are sometimes used to connect relatively short two-bearing rotors of a compressor and a gas turbine. The connection of the compressor and turbine parts of the rotor is carried out in this case using flanges and zone bolts.

In turbomachines, where misalignment of the connected shafts and significant mutual axial displacements are possible, **movable couplings** are used, the most common of which is **the gear (spline) coupling**.

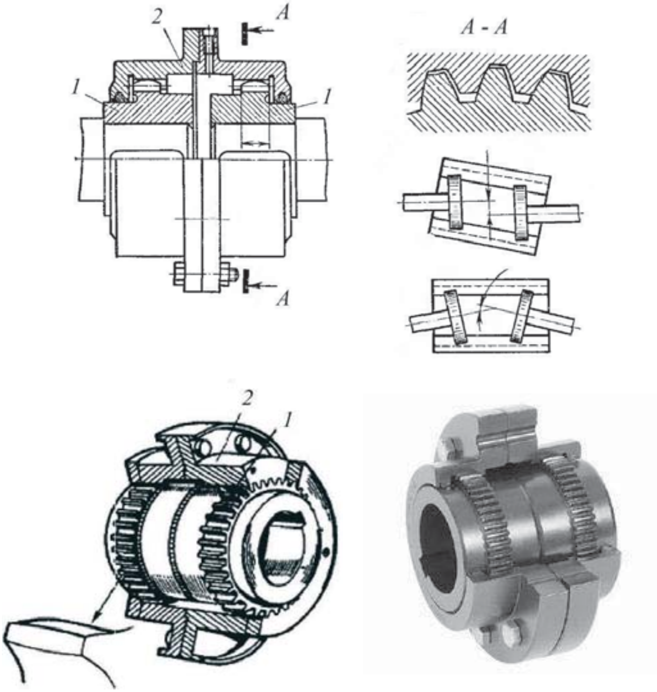


Fig. 3.32. Gear (spline) coupling:
1 – wheels with radial splines; 2 – connecting gear drum

The simplest version of the gear coupling is shown in Fig. 3.32 [7]. The coupling consists of two sprockets 2 with involute profile splines fixed at the ends of the connected shafts and a connecting drum (gear ring) 1, into the recesses of which the sprocket splines enter. The total lateral clearance between the splines should be no more than 0.3-0.4 mm; the contact area of each spline should be no less than 60% of its working surface.

To protect the gear coupling from wear, it is advisable to supply lubrication to it.

Gear (spline) couplings in gas turbine units are often used to connect the compressor and turbine shafts, and also, in some cases, to connect the turbine to the gearbox shaft.

The elements of the gear connection of the compressor and turbine rotors of a gas turbine engine are shown in Fig. 3.33.

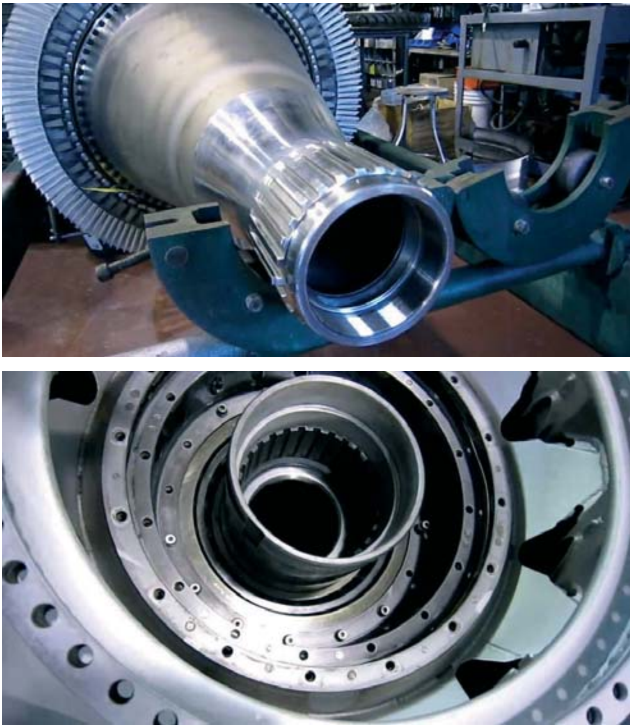


Fig. 3.33. Elements of the gear connection of the engine rotors

The gear (spline) coupling makes it possible to significantly simplify the production technology (there is no need for precise centering and joint processing of the landing surfaces in the bearing housings), provides the possibility of assembling individual units (separately, the compressor, combustion chamber, and turbine) and separate balancing of the compressor and turbine rotors.

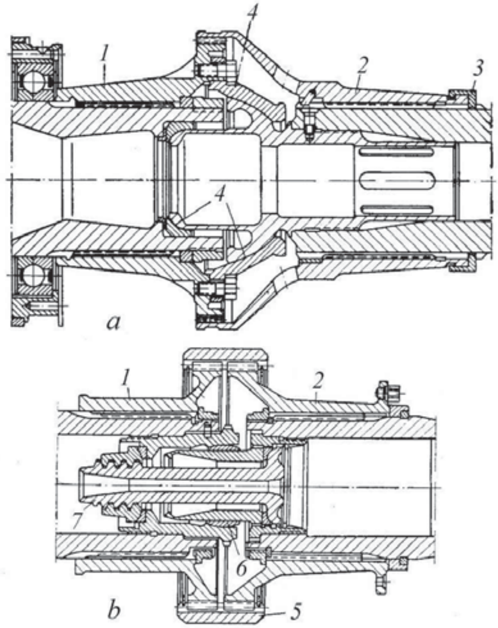


Fig. 3.34. Gear (spline) couplings:

a – with spherical tip; *b* – with tie bolt; 1 – driven splined sleeve; 2 – leading splined sleeve; 3 – retaining ring; 4 – hemispherical bearing surfaces; 5 – splined sleeve; 6 – spherical ring; 7 – connecting bolt

In their design, gear couplings can differ significantly from each other. For example, the couplings shown in Fig. 3.34, *a* and *b*, consist of leading and driven splined sleeves, fixed respectively on the turbine and compressor shafts [7]. The torque is transmitted by means of short splines on the sleeve crowns, and the axial forces are perceived by hemispherical support surfaces 4 (Fig. 3.34, *a*) or by the connecting bolt 7 (Fig. 3.34, *b*). Because the axial forces from the compressor and turbine rotors are directed in opposite directions, only the difference of these forces is transmitted to the thrust bearing.

Such couplings are reliable in operation, but a bit complicated. In addition, their use complicates installation, since special hatches in the engine housing are required to assemble the coupling.

A simple connection of the turbine and compressor shafts is shown in Fig. 3.35 [7]. Here, the external splines of the compressor shaft mesh with the internal splines of the turbine shaft and serve to transmit torque. The shafts are connected in the axial direction by means of a coupling bolt 3, which is screwed into the compressor shaft and abuts against the turbine shaft through a thrust bushing.

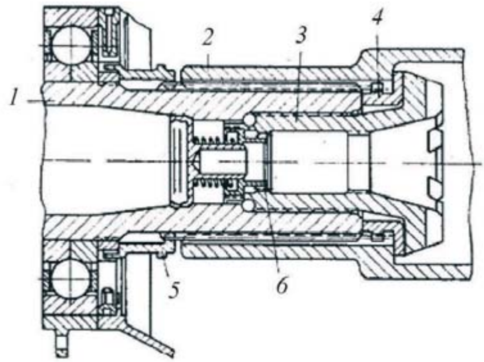


Fig. 3.35. Connection of turbine and compressor shafts:
1 – compressor shaft; 2 – turbine shaft; 3 – tie bolt, 4 – thrust bushing; 5 – distance bushing; 6 – tie bolt lock

The axial clearance of 0.3-0.5 mm, which is assumed between the end of the turbine shaft and the distance sleeve, as well as the gaps in the splines, allow for some skewing of the shafts. To screw in the tie bolt, free access to the end of the compressor shaft is required.

Many other splined methods of connecting the turbine and compressor shafts are also used.

In elastic couplings[1] torque is transmitted from one shaft to another through an elastic (resilient) element. Thanks to this, vibrations that can be transmitted from one rotor to another are quickly damped in the coupling, and the transmission of bending moment from one shaft to another is also excluded, even in the absence of perfect alignment of the axis lines of the connected shafts.

Fig. 3.36 shows a spring-elastic coupling used by the Power Machines Company (LMZ) [7]. The flanges that are put on the shafts have toothed threads. A wave spring 1 is inserted between the teeth of the flanges. For ease of installation, the spring is cut into separate sections. The casing 2 is rigidly connected to one flange of the coupling. Gaps are provided between the holes of the second flange and the fastening bolts, as well as between the casing and the flange, which ensure axial movement of the rotors. The torque is transmitted using a wave spring.

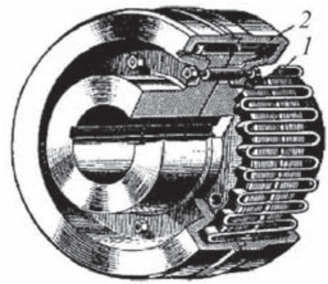


Fig. 3.36. Spring elastic coupling:
1 – wave spring; 2 – casing

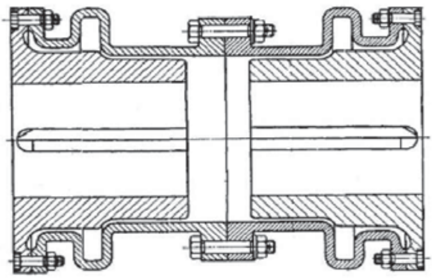


Fig. 3.37. Elastic coupling of the Power Machines (LMZ)

Oil for lubricating the friction surfaces comes from holes drilled in the hubs of the coupling halves near the bearings and drains through holes on the end surfaces of the housing, which are positioned so that the spring is flooded with oil.

[1] Elastic couplings are also often referred to in the literature as **flexible** ones.

The spring is inserted into the grooves of the half couplings with a gap of 0.5-0.8 mm.

The spring coupling allows independent longitudinal displacement of the rotors, as well as a slight eccentricity of the connecting shafts. Some complexity of the design and increased manufacturing cost are fully compensated by its advantages. This coupling is used mainly for connecting the turbine shaft to the gearbox shaft and for connecting the shafts of multi-casing steam turbines.

Fig. 3.37 shows an elastic coupling used by the Power Machines Company (LMZ) to connect turbine rotors to generator rotors or to connect medium- and low-pressure cylinders of steam turbines (such a coupling is sometimes called **semi-rigid** or **semi-flexible**) [7].

Flanges are fitted with tension on the ends of the shafts, which have a slight taper. The flanges are held from turning by longitudinal keys. A wavy semi-flexible part of the coupling is inserted between the flanges. The latter is rigid in torsion and elastic in bending. The presence of the wavy part of the coupling softens the transmission of vibration from one rotor to another. The semi-flexible part of the coupling is rigidly connected to the flanges by bolts.

Disc-type elastic couplings are widely used. These couplings transmit significant torques and can compensate for the displacement of the rotor axes. A type of disc-type coupling is used in its gas turbine units by Zorya-Mashproekt to transmit torque from the power turbine shaft to the consumer

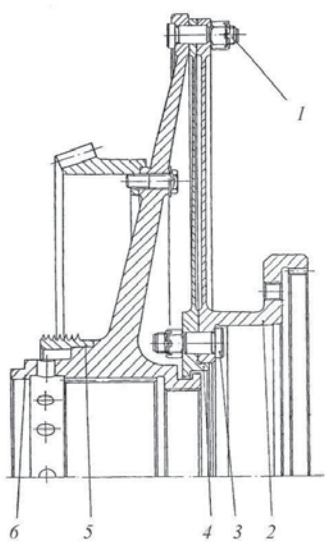


Fig. 3.38. Disk-type coupling of the Zorya-Mashproekt UGT15000 engine [15]:

1, 3 – clamping bolts; 2, 4 – coupling cheeks (disks); 5 – labyrinth; 6 – coupling housing shaft (Fig. 3.38).

The coupling consists of a housing 6, two elastic disk cheeks 2 and 4, connected together by clamping bolts 1 (on the outer diameter) and clamping bolts 3 (on the inner diameter). The coupling cheek 2 has a flange with slots, to which the output shaft is attached.

The torque from the power turbine shaft to the coupling housing 6 is transmitted by means of radial splines. A labyrinth 5 is installed on the housing 6, which, together with the rear cover of the turbine support ring, forms a seal for the oil cavity of the support ring. The housing also houses an element for the power turbine speed limit sensor.

The connection of the power turbine with the gearbox through an elastic coupling ensures normal operation of the gas turbine unit with minor misalignments of the engine and gearbox shafts.

Control tasks and questions

1. Name the main types of rotors of axial turbomachines by manufacturing method and design.
2. Describe the structure, advantages and disadvantages of solid rotors.
3. Describe the structure, advantages and disadvantages of composite rotors.
4. Analyze the design of disc-type composite rotors.
5. Explain the design of disc-drum type composite rotors.
6. Analyze the design of drum-type composite rotors.
7. Explain the main types of rotor connections in turbomachines.
8. Describe the structure of a gear (spline) coupling.

4. SUPPORTS OF TURBOMACHINES

By the support of a turbomachine (compressor or turbine), we mean the bearing device on which the turbomachine rotor rests and which receives and transmits to the housing the mass of the rotor and the axial loads acting on it.

The design of the support with any type of bearing must provide:

– supply and discharge of the required amount of lubricant;

– possibility of thermal expansion of the rotor;

– ease of assembly and disassembly;

– sealing the device against lubricant leaks;

– in some cases, axial fixation of the rotor.

In addition, the supports must be sufficiently strong and rigid. The latter, however, does not apply to special damper and elastic-damper supports.

The usual sequence for designing turbomachinery supports is as follows: a sketch and power diagram of the engine are drawn up, which allows you to select the number and type of supports; the loads acting on each support are determined; the type, size and accuracy class of the bearing are selected; the thermal regime of the bearings is assessed, which determines the type and consumption of lubricant; the types of oil and air seals are selected; the possibility of supplying lubricant to the support and removing it from it is checked.

The specific design of gas turbine engine bearings is associated with significant heat inflows from nearby heated rotor and housing elements. This requires the organization of a reliable bearing cooling system and special design measures that allow maintaining the alignment of the bearing housings and turbomachines.

The number of supports is selected depending on the length and stiffness of the engine rotor. The rotor of a turbomachine can be supported by two, three, four, or more supports.

Two supports are chosen for the rotor of a free power turbine (Fig. 4.1, *a*), as well as for a turbocompressor, in which the compressor rotor and turbine rotor are connected into a single structure (Fig. 4.1, *b*, *c*).

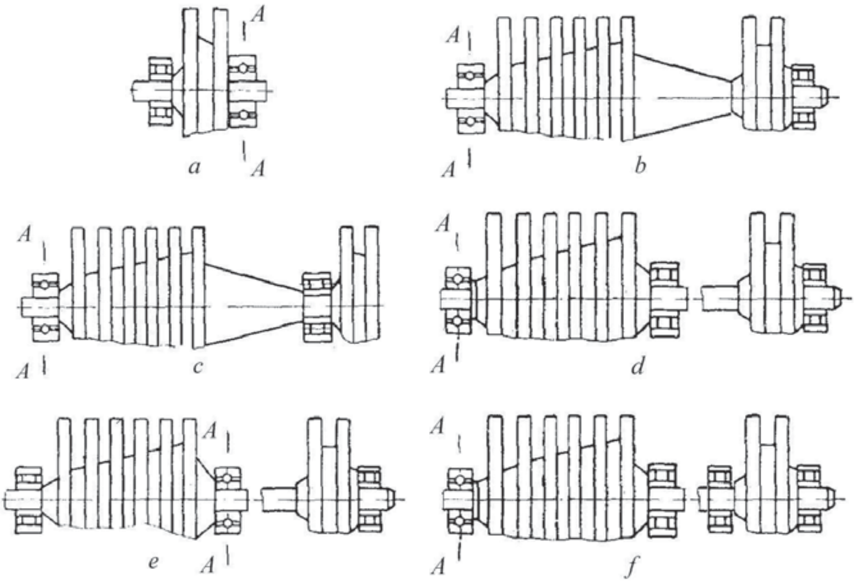


Fig. 4.1. Schemes of turbomachine rotors:

a – two-support rotor of a free power turbine; *b* – two-support rotor of a turbocompressor; *c* – two-support rotor of a turbocompressor with a cantilever arrangement of the turbine; *d*, *e* – three-support rotors of a turbocompressor; *f* – four-support rotor of a turbocompressor; AA – plane of axial fixation

For a free power turbine, an attempt is made to place a bearing that accepts axial loads (both from gas-dynamic forces and from thermal expansion of the rotor) on the consumer side, thereby preventing the transmission of these loads to it (to the consumer). In Fig. 4.1, plane AA is the plane of axial fixation of the rotor.

For a turbocompressor rotor, the axial fixation plane is usually located in front of the compressor. Heat inflows into the bearing from adjacent parts in this part of the engine are minimal, and therefore the thrust bearing will operate in better temperature conditions.

To reduce the total length of the turbocompressor, its rear support can be placed in front of the turbine (the so-called **cantilever arrangement of the turbine rotor**) (Fig. 4.1, *c*). However, the rigidity of the rotor is reduced. The examples are the rotors of high-pressure turbocompressors of the UGT3000, UGT6000, UGT10000, UGT15000, and UGT25000 engines manufactured by Zorya-Mashproekt.

The main advantage of the two-support design is its simplicity and lightness; however, the use of two supports is possible only with a relatively small distance between the supports and significant rigidity of both the rotor and the housing.

With a significant distance from the front support to the turbine or with a long (multi-stage) turbine, it is advisable to make **the three-support turbocompressor rotor** (Fig. 4.1, *d*, *e*). The axial fixation plane (thrust bearing) can be located either in the front (Fig. 4.1, *d*) or in the rear (Fig. 4.1, *e*) support of the compressor. The scheme in Fig. 4.1, *d* is typical, for example, for low-pressure turbocompressors of UGT3000, UGT6000, UGT10000, UGT15000 and UGT25000 engines manufactured by Zorya-Mashproekt; the scheme in Fig. 4.1, *e* is for the high-pressure turbocompressor of the UGT16000 engine of the same company.

Four-support turbocompressor rotors (Fig. 4.1, *f*) are used for long shaft lengths when obtaining sufficient rigidity of the rotor and housings entails a significant increase in their mass. When using a four-support rotor, assembly and balancing of the turbine and compressor are simplified, but the mass of the engine increases somewhat. An example of such a rotor design is the rotor of the low-pressure turbocompressor of the UGT16000 engine.

According to their functional purpose, bearings that are part of the supports of gas turbine engines are divided into radial, thrust, and radial-thrust bearings.

Radial bearings serve to absorb the mass of the rotor and possible additional radial forces that arise during operation (due to the ship's roll, rotor imbalance, etc.). Another purpose is to radially center the rotor relative to the turbine or compressor housing.

Thrust bearings are designed to absorb axial forces acting on the rotor due to the gas-dynamic effect of the flow on the blades, the pressure drop on the disks, and the end surfaces of the rotor shaft. In addition, thrust bearings are used for axial centering of the rotor relative to the turbine housing.

Radial-thrust bearings are designed to accommodate combined loads, i.e., simultaneously acting radial and axial loads.

By design, turbomachinery bearings can be rolling or plain bearings.

4.1. Supports with rolling bearings

Rolling bearings are widely used in turbines and compressors of aircraft-type gas turbine engines. **Their main advantages** (compared to plain bearings) include:

– low friction coefficient (and therefore, low mechanical losses and low oil consumption for cooling the bearing, which significantly simplifies the design of the lubrication system);

– the ability to operate reliably at high speeds (in modern gas turbine engines, these bearings operate at speeds of 5,000-18,000 rpm; in auxiliary engines, the speed reaches 35,000-120,000 rpm) [15]);

– wide selection of standard bearing types;

– insignificant length;

– ease of installation and maintenance.

Disadvantages of rolling bearings (compared to plain bearings):

– low durability (this parameter significantly depends on the load and rotor speed and ranges from 8,000 to 30,000 hours);

– larger radial size;

– higher mass.

In gas turbine engines, **ball bearings** (as radial and radial-thrust bearings) and **roller bearings** (as radial bearings) are used.

Fig. 4.2, *a-f* and 4.3 show **different types of ball bearings**.

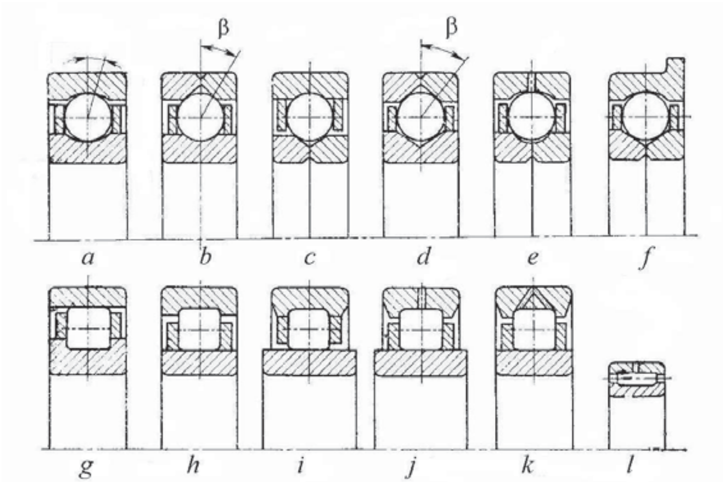


Fig. 4.2. Types of rolling bearings:

a-f – ball bearings; *g-l* – roller bearings

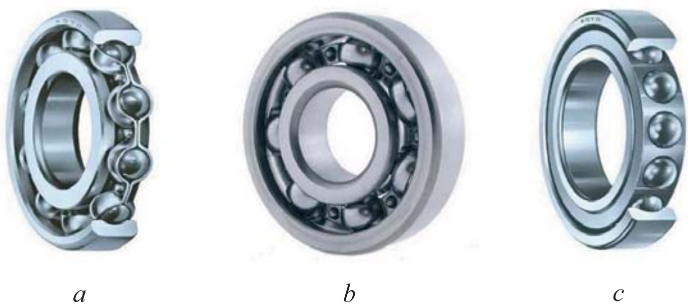


Fig. 4.3. Ball bearings:

a, *b* – with stamped riveted separator; *c* – with solid separator

A single-row two-point (two points of contact of the ball with the rings) **ball bearing** (Fig. 4.2, *a*) can accept radial and insignificant axial loads. The radius of the raceway in it is slightly larger than the radius of the ball. Under the action of the axial force, a contact angle β is formed in the bearing, the value of which determines the permissible load (the larger the angle, the greater the axial load that can be sustained).

The bearing shown in Fig. 4.2, *b*, has an outer split ring; the bearing shown in Fig. 4.2, *c*, has an inner split ring. Bearings with split rings have a larger number of balls, a higher depth of the grooves in the rings, and an increased contact angle β, so they can accommodate higher radial and especially axial loads. They are called **three-point bearings**, meaning that each ball has contact with the raceways of the rings at three points.

The bearing shown in Fig. 4.2, *d* has an outer split ring and a deeper, profiled two-radius groove of the inner ring, which makes it possible to increase the contact angle β without increasing the axial clearance of the bearing. Each ball in such a bearing has contact with the rings at four points, which is why they are called **four-point bearings**. Such bearings can accept even greater radial and axial loads.

Four-point bearings, to enable their assembly, can be made with one split ring: an external (see Fig. 4.2, *d*) or internal (see Fig. 4.2, *e*) ring. Split rings make it possible to use a stronger non-separable separator. However, along with rolling, due to different speeds at the places of contact of the balls and raceways, the balls experience sliding, which increases friction, causes heating, and requires an increased supply of oil to the bearing for cooling.

Three-point and four-point bearings are used for significant axial loads in compressor and turbine bearings. Three-point bearings are also used in devices where small axial clearances are required, such as in bevel gears.

If the axial load is so great that it cannot be supported by a single four-point bearing, a support consisting of two or three bearings is used, as will be shown below.

Fig. 4.2, *f* shows a bearing with a flange on the outer ring, which is designed for mounting to the housing (the flange is clamped between the housing and the flange screwed to it).

Fig. 4.2, *g*-*l* and 4.4 show **different types of roller bearings**. They differ in that the sides that guide the rollers are made on the inner (see Fig. 4.2, *g*) or outer (see Fig. 4.2, *h*) rings. The inner rings of bearings intended for significant axial thermal displacements of the shaft and housing are made wider (see Fig. 4.2, *i*, *j*) than the outer ones.

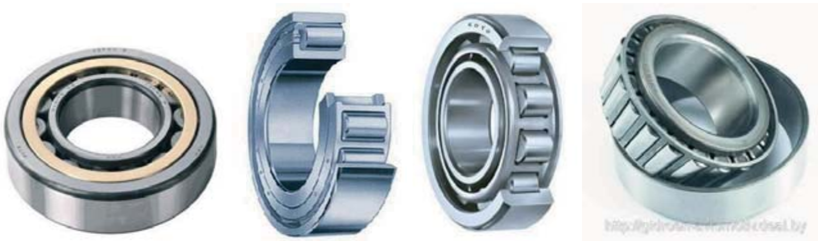


Fig. 4.4. Roller bearings

Roller bearings, due to their larger contact surface, can withstand significantly higher radial loads than ball bearings (on average by 70-90%). However, they are not designed to absorb axial loads at all and do not allow the shaft to be misaligned. When the shaft is misaligned, the rollers start to work with their edges, and the bearing quickly collapses.

Needle roller bearings (see Fig. 4.2, *l* and 4.5) have much smaller mass and radial dimensions than other types of rolling bearings, but their load-bearing capacity and speed are limited. With increasing rotational speed, the permissible load of needle bearings decreases, so they are used at low loads and rotational speeds, for example, in the supports of rotary blades of axial compressors, in transmissions to units, etc.

To reduce radial dimensions, roller bearings can be made without an outer or inner ring. In this case, the rollers roll on specially treated surfaces of the parts by carburizing or nitriding.

Fig. 4.5. Needle roller bearing

The bearing industry of the former Soviet Union produces rolling bearings in five size series: ultralight, extra light, light, medium and heavy. With the same values of the inner diameter of the bearing, the lighter the series, the higher its permissible speed and lower load capacity (i.e., radial loads that can be sustained), and vice versa.

The bearings produced can be classified **by the class of manufacturing accuracy**. There are accuracy classes 0 (normal class), 6 (slightly-high), 5 (high), 4 (extra-high), and 2 (ultra-high class). The performance of the bearing depends to a large extent on the manufacturing accuracy, but at the same time, its cost also increases [15]:

Class of accuracy	0	6	5	4	2
Relative cost	1	1,3	2	4	10

As noted in [15], when using bearings of accuracy classes 6 and 5 with massive non-ferrous metal separators, their maximum speed can be increased by approximately 1.5 times, and the use of bearings of accuracy classes 4 or 2 with textolite separators allows almost doubling the maximum speed compared to bearings of accuracy class 0. To achieve such indicators, increasing the accuracy class, however, must necessarily be accompanied by compliance with higher requirements for the quality of installation and operation of the bearing device as a whole.

Rolling bearings are standard designs, the main dimensions of which, depending on the operating parameters, are given in catalogs and reference books. Therefore, when designing supports based on them, one usually does not deal with the calculation of bearings, but with their correct selection based on the calculated load conditions.

However, it should be noted that catalogs and reference books contain data on bearings mainly of general application, which may not provide the required durability of gas turbine engine supports. Therefore, when designing a new turbomachine, specialized companies often design a new bearing specifically for it, taking into account the operating conditions of a particular bearing device.

Bearing separators can be riveted (separable) (Fig. 4.6) or solid (Figs. 4.7 and 4.8). They are manufactured by stamping or machining.

Fig. 4.6. Disassembled ball bearing with riveted separator

The separator distributes the rolling elements evenly around the circumference, which eliminates friction between them and evenly distributes the load on them; the use of massive separators made of non-ferrous metals allows them to withstand significant rotational speeds. The use of separators makes it possible to reduce the size of balls and rollers. Separator-free bearings, despite their increased load capacity, are not used in gas turbine engine bearings due to the rapid occurrence of the so-called annular wear effect, which is caused by the collision of rolling elements during their rotation.

Fig. 4.7. Solid bronze ball bearing separator

Fig. 4.8. Roller bearing with a solid bronze separator

In compressor and turbine bearings, single-row ball and roller bearings of light series of high accuracy classes with well-balanced massive bronze or duralumin separators are mainly used.

The separators are centered on the outer or inner ring.

When centering the separator on the outer ring, the lubrication of the inner ring and centering surfaces is improved, the separator is balanced during operation, and heat is removed from the separator through the cooler outer ring. The disadvantages of centering on the outer ring include: an increase in the mass of the separator, the need to accurately select the gap between the separator and the outer ring to avoid its seizure when heated due to the difference in the coefficients of linear expansion of materials, and deformation of parts under the action of centrifugal forces.

For good bearing operation, the condition of the contact surfaces of the separator and rings is of great importance, depending on the quality of processing and the process of their running-in during the first engine run-in.

The minimum coefficient of friction in rolling bearings is observed with a thin film of low-viscosity oil. Lubrication in rolling bearings is necessary to protect the working surfaces from corrosion and wear during dry friction and, mainly, to remove friction heat.

Increasing the diameter of the balls or rollers reduces the coefficient of friction but increases the size and weight of the bearings. Increasing the number of balls or rollers increases the coefficient of friction.

The friction coefficient of ball bearings is in the range of 0.002-0.004 and can reach 0.008 when starting the engine at low temperatures due to the increased viscosity of the oil.

In roller bearings, due to friction at the ends of the rollers, the friction coefficient is greater than in ball bearings, and is 0.003-0.006.

In needle bearings, the friction coefficient is even higher, and is 0.008-0.020. This is explained by the slipping of the needles and the friction between them. In the unloaded part of the bearing, the needles do not roll at all, but slide; in the loaded part, slippage (already during rolling) occurs due to misalignment of the needles due to their significant relative length.

The conducted studies have shown that in roller bearings, which are supports for the rotors of gas turbine engines and operate in conditions of supplying them with quite significant amounts of oil, power losses consist not only of friction losses. As a result of the mixing of oil by rolling elements in the bearing channel, hydrodynamic losses occur, which account for up to 80% of the total power consumption for the bearing drive. In this regard, losses in gas turbine roller bearings cannot be estimated in the generally accepted way for conventional rolling bearings - through the given friction coefficient, because it does not reflect the phenomena occurring in these bearings in general [15].

The durability of rolling bearings depends on their design, the magnitude and direction of the loads acting on them, the speed, the operating conditions, and especially the temperature of the bearing parts.

The temperature of the bearing is affected by the heat input from the connected parts, the load, the speed, the amount of oil supplied, the conditions of its supply and circulation in the bearing, the design of the bearing elements. The oil jets must pass into the gap between the separator and the inner ring (Fig. 4.9) so that the oil is supplied to the inner ring, the raceway of which is insufficiently lubricated as a result of the action of centrifugal forces.

To reduce the temperature of a rolling bearing, efforts are made to improve the circulation of oil through it by making holes or slots in the outer ring, special milling on the separator, manufacturing outer rings without sides, etc.

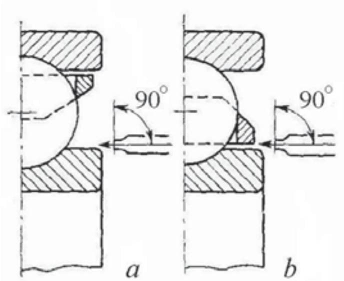


Fig. 4.9. Oil supply from the nozzle: *a* – when centering the separator on the outer ring; *b* – when centering the separator on the inner ring

It is necessary to strive for the heated oil to flow freely and not to be retained in the bearing. Fig. 4.10 shows a graph of the temperature change of three roller bearings depending on the speed. The lowest temperature is the bearing with a smooth outer ring and a separator centered on the outer ring. The highest temperature is the bearing with a separator centered on the inner ring, which is explained by the small gap between them, as a result of which a small amount of oil gets to the rollers and rings. The bearing with flanges on the outer ring and a separator centered on the outer ring occupies an intermediate position in terms of temperature [15].

At a certain temperature, which is called **the limit temperature**, the operation of the bearing at the permissible rotation frequency becomes unreliable. This is explained by the fact that the lubrication of parts deteriorates at elevated temperatures; in addition, it should be borne in mind that at temperatures above 175 °C, tempering occurs and the hardness of bearing steel SHX15 (massively used for the manufacture of bearing rings) decreases, which is reflected in the performance of the bearing. With improved heat treatment of bearing steel SHX15, the limit temperature of the bearing can be shifted up to approximately 250 °C.

When there is an axial load on the bearings and to fix the shaft in the axial direction, the bearing rings must be held on the shaft and in the housing from axial movement using various types of securing devices.

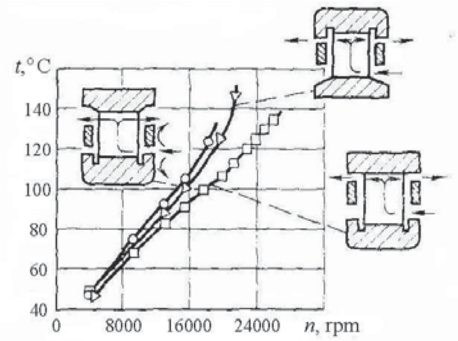


Fig. 4.10. Temperature of the outer rings of three roller bearings depending on the speed

The method of fastening the bearing rings on the shaft and in the housing is selected depending on the magnitude and direction of the load, the speed, the type of bearing, the requirements for assembly and disassembly of the device, and the production capabilities of the manufacturer. The higher the axial loads and the higher the speed of rotation of the rotor, the more reliable the axial fastening of the bearing rings should be.

If the bearing is not subjected to axial load and it is only necessary to prevent accidental displacement of the bearing, the axial fastening of the ring on the shaft is carried out by means of a fit without the use of additional fastening devices.

The most common **methods of fastening inner bearing rings to the shaft** are as follows:

– by a nut and a lock washer, the inner tongue of which enters the groove on the shaft, and one of the corner teeth bends into the slot of the nut (Figs. 4.11, *a* and 4.12);

– by a split elastic thrust ring, usually of rectangular, and sometimes circular cross-section, which is inserted into the annular groove of the shaft (Figs. 4.11, *b* and 4.13, *a*);

– by a slotted nut, which is counterbalanced by a tightening screw (Fig. 4.11, *c*).

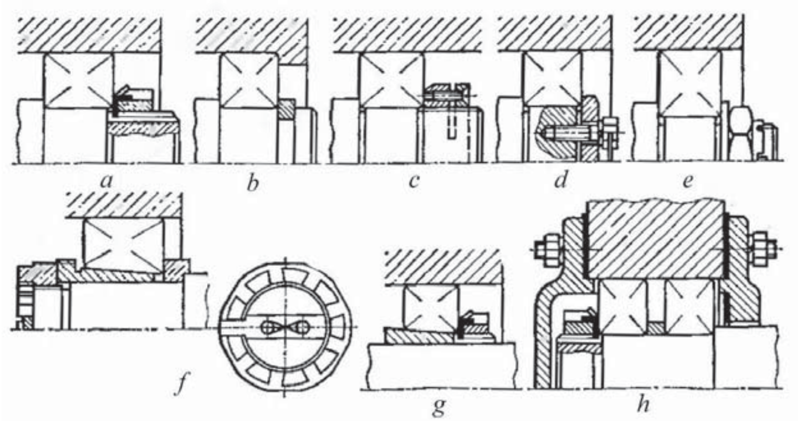


Fig. 4.11. Fastening the inner ring of bearings to the shaft [15]:
a – nut and lock washer; *b* – elastic thrust ring; *c* – nut with a slot; *d* – flat end washer; *e* – flat and box washers; *f* – fixing sleeve; *g* – clamping sleeve; *h* – nut, lock washer and sleeve

In the case of mounting the bearing on a solid shaft, the bearing is secured with a flat end washer, which is attached to the shaft end with screws that are held against turning by lock washers and wire (Fig. 4.11, *d*). It is possible to secure with a flat washer and a crown washer with a cotter pin (Fig. 4.11, *e*). For ease of installation and removal, the bearings are secured to the shaft using a retaining (Fig. 4.11, *f*) or clamping (Fig. 4.11, *g*) bushing, nut, and lock washer. Two bearings are secured with a nut, lock washer and bushing between the inner rings (Fig. 4.11, *h*).

The most common **methods of fastening the outer rings of bearings in the housing** are as follows:

– by stamped or machined shaped cover and bolts (Fig. 4.14, *a*);

– by a split thrust elastic ring of rectangular or circular cross-section, which is inserted into the groove of the housing opening (Figs. 4.14, *b* and 4.13, *b*);

– by a retaining ring installed in the groove of the outer ring (Fig. 4.14, *c*);

– with a thrust flange on the outer ring (Fig. 4.14, *d*);

– by nut with external thread (Fig. 4.14, *e*);

– with a massive cover and bolts (Fig. 4.14, *f*).

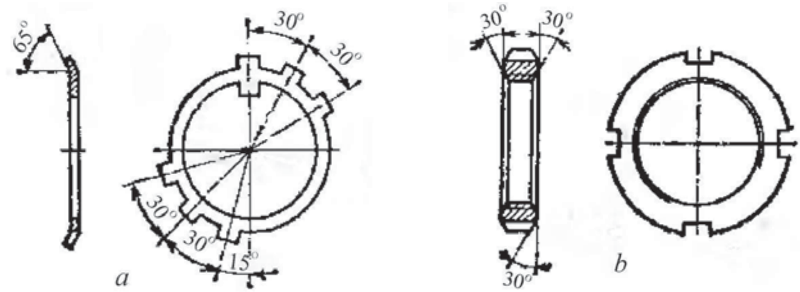


Fig. 4.12. General view of lock washers (*a*) and nuts (*b*) intended for fastening rolling bearings on shafts [15]

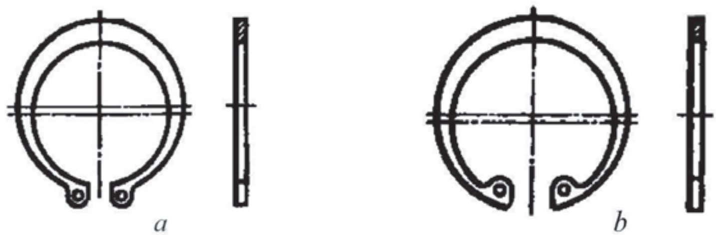


Fig. 4.13. Split elastic ring for axial fixation of inner (a) and outer (b) rings of bearings

A characteristic feature of rolling bearings used in gas turbine engines is the large clearances between the rolling elements and the rings. These clearances are necessary to prevent the parts from seizing when the bearings heat up and expand.

At high speeds and in the absence of axial force on a radial-thrust ball bearing (or with a small amount of this force), centrifugal and gyroscopic forces can cause the balls to slip relative to the raceways, which can damage the bearing's working surfaces. To prevent this, a constant axial load is applied to such a bearing, i.e., a preload is performed.

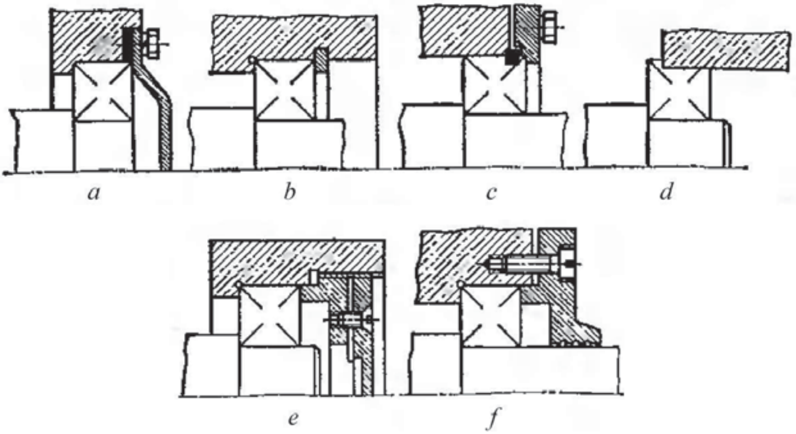


Fig. 4.14. Fastening the outer ring of bearings in the housing [15]:
a – shaped cover and bolts; *b* – split elastic thrust ring; *c* – retaining ring; *d* – thrust flange on the outer ring; *e* – nut with external thread; *f* – solid cover and bolts

For the same purpose (to avoid slipping) in gas turbine engines, **special loading devices are also used for roller bearings**. Fig. 4.15 shows a similar device for the roller bearing of the front support for the high-pressure compressor of the Zorya-Mashproekt D050 marine engine.

The loading device consists of a sleeve 2, eight packages of disc-shaped spring washers 3 (see also Fig. 4.16), between which are placed intermediate washers 4, spheres 6, segments 9, front support sleeve 7 and adjusting ring 5.

Before assembling the loading device, the settlement of the set of packages of disc springs with intermediate washers under a load of 5-6 kN is determined. This pressing force is provided in the assembled form by means of the adjusting ring 5. The pressing force from the washers 3 is transmitted through the sphere 6 and the segment 9 to the outer ring of the bearing 8, deforming it, and this eliminates the relative sliding of the rolling elements and bearing rings.

Let's consider some designs of supports for gas turbine engines with rolling bearings.

The axially fixed support is shown in Fig. 4.17. Here, a three-point radial-thrust ball bearing with an internal split ring is used (the design of the device will change slightly if the split ring is external). The adjusting ring 10 is used to adjust the axial clearance in the flow-through elements of turbomachines, and the oil-repelling ring 11 is used to seal the bearing cavity. These rings and the labyrinth seal 6 are tightened with a nut 9. The nut 14 fixes the outer ring together with the labyrinth sleeve in the bearing cup 13 and is locked with a special block 15 and screw 16.

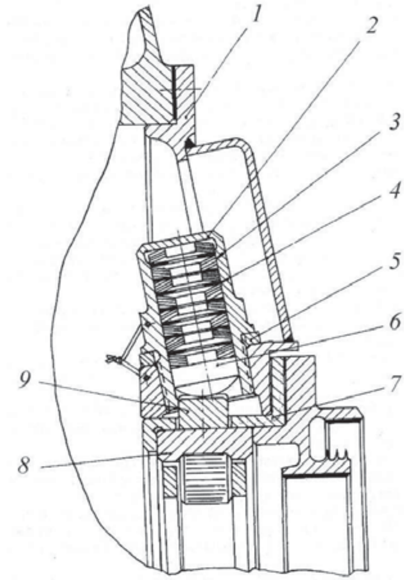


Fig. 4.15. Loading device [15]:
1 – wall of the front support of the compressor; 2 – sleeve; 3 – spring washer; 4 – washer; 5 – adjusting ring; 6 – sphere; 7 – sleeve of the front support; 8 – bearing; 9 – segment

When installing roller radial bearings, it is necessary to ensure that the outer and inner rings are fixed from axial movement. Fig. 4.18 shows the front support of the compressor of a gas turbine engine with a bearing 8, which has flanges on the outer ring. This ring is placed in a steel intermediate cup 9, which is pressed into an aluminum housing 4. Installing the bearing directly in a housing made of aluminum or magnesium alloys is not used, since during engine operation, due to the difference in thermal expansion of the materials of the ring and housing, as well as due to rotor vibrations, the fit of the bearing in the housing may be noticeably disturbed. In this design, the outer ring of the bearing is fixed from axial movement by the flanges of the housing on the left and the flanges of the oil seal sleeve flange 2 on the right.

The inner ring of the roller bearing 8 is tightened with a clamping nut 7. The main design feature of the assembly is the possibility of removing the housing 4 of the support from the rotor without unscrewing the nut 7. To do this, the diameter D_3 of the seal is chosen to be larger than the diameter of the raceway of the inner ring of the roller bearing, and the outer diameter of the clamping nut $D_1 < D_2$. This ratio of dimensions is often used in the design of gas turbine engine supports with roller bearings.

Fig. 4.16. Elastic disc washers

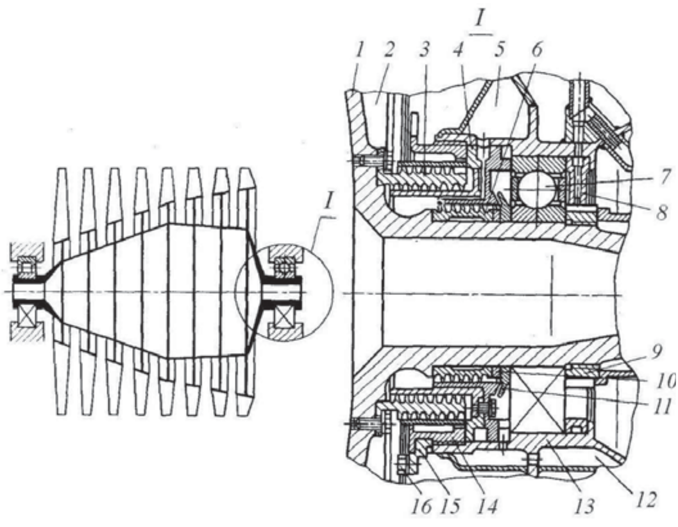


Fig. 4.17. Support with axial fixation [15]:

1 – compressor shaft; 2 – cavity with compressed hot air; 3 and 6 – labyrinth seals; 4 – labyrinth sleeve; 5 – intermediate cavity; 7 – bearing; 8 – nozzle ring; 9 and 14 – nuts; 10 – adjusting ring; 11 – oil deflector ring; 12 – housing oil sump; 13 – bearing cup; 15 – block; 16 – locking screw

Effective **thermal protection of the bearing device** is essential for the compressor rear support and especially for gas turbine bearings.

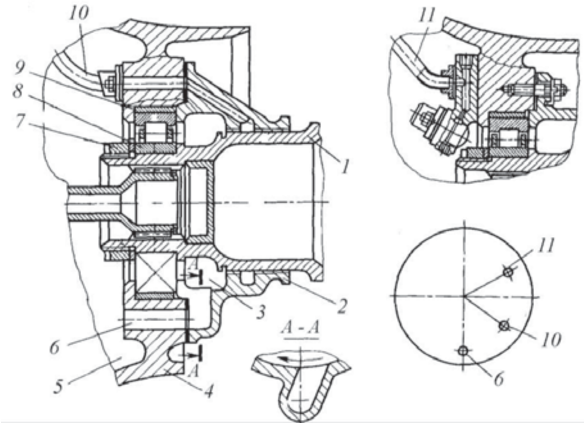


Fig. 4.18. Support with bearing [15]:

1 – shaft trunnion; 2 – oil seal; 3 – oil cavity; 4 – housing; 5 – cavity; 6 – oil drain hole; 7 – clamping nut; 8 – roller bearing; 9 – intermediate cup; 10 – air supply; 11 – oil supply

Thermal protection of the bearing is carried out in various design ways. Fig. 4.17 shows the application of one of them. In this case, the bearing 7 is installed in the rear support of the compressor, and the cavity 2 is filled with compressed heated air. So that the air, having passed through the labyrinth seal, does not enter the bearing cavity, it is discharged into the intermediate cavity 5 and then into the atmosphere through the blowing system. This guarantees a slight excess air pressure in front of the labyrinth seal 6 and, consequently, minimal leaks of hot air into the bearing cavity. In several designs, cold air of low pressure is supplied through a cavity similar to cavity 5 or through a special line. In this case, leakage through the labyrinth seal 3 is reduced, and relatively cold air enters the bearing cavity.

Another method of thermal protection of bearings can be seen in Fig. 4.19. To improve the cooling of the support in housing 1, longitudinal grooves are made (section A-A). In addition, air is supplied through the rotor cavity and hole 3, as well as hole 2 in the labyrinth seal, under pressure created by one of the intermediate stages of the compressor. The air pressure is selected so that it spreads to both sides of hole 3, preventing the entry of more heated air from the air cavity 4 into the support cavity.

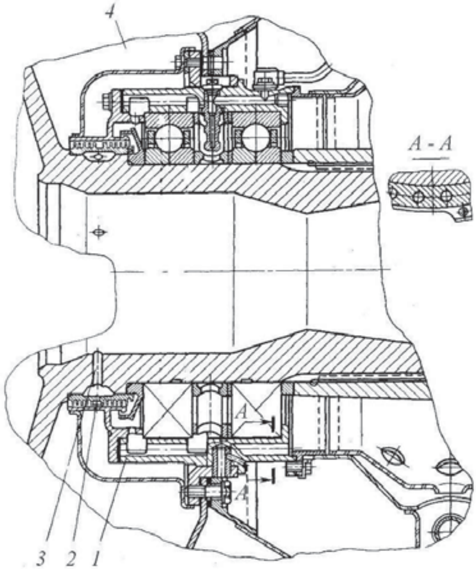


Fig. 4.19. Support with reduced heat input into the bearing cavity [15]

Carefully organized cooling of the front support of the turbine rotor is shown in Fig. 4.20. Air is blown through the air cavity 11, which is taken after the compressor or after a special fan. This air cools the walls of the bearing housing 12 and contributes to the cooling of the bearing from the outside. This bearing is also cooled quite intensively from the inside: from the same cavity 11 through the gap 10 between the turbine shaft 14 and the stepped sleeve 1, part of the air flows under the action of the pumping effect of the radial grooves 13 made on the end surface of the stepped sleeve 1.

The stepped sleeve 1 rests on the shaft along a cylindrical protrusion 9. To reduce pressure losses when air enters the gap 10, 36 grooves are made on the protrusion 9, located at an angle of 45° to the shaft axis (left-hand spiral).

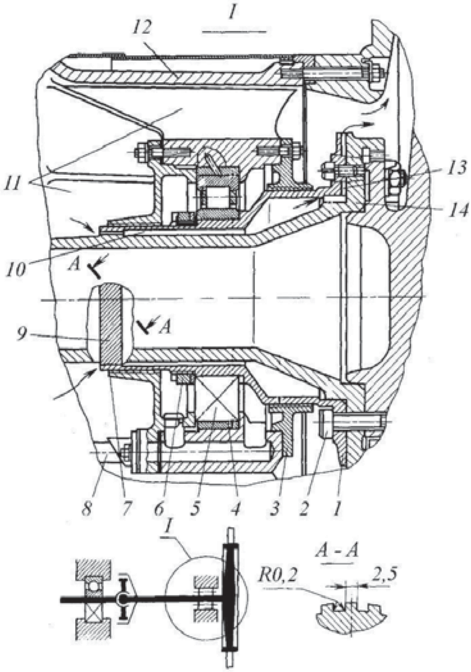


Fig. 4.20. Turbine front support [15]:

1 – stepped sleeve; 2 – bolt; 3 and 7 – seal housings; 4 – steel sleeve; 5 – bearing; 6 – nut; 8 – oil drain line; 9 – shaft protrusion with grooves; 10 – gap; 11 – air cavity; 12 – bearing housing; 13 – radial groove; 14 – turbine shaft

Heating of the bearing and oil in the support (Fig. 4.21) is prevented by the blowing of air into the cavity 1 from the space behind the compressor. The pressure of this air is higher than that of the gas in the gap 2 between the turbine disk 5 and the housing 1, and only slightly higher than in the cavity 4 of the support. Therefore, the air, which has a fairly high temperature, flows from the air cavity 3 into the cavity 4 of the support only in a small amount. Such a design of the unit (with "closing" the hot gas from penetrating into the cavity of the support) is quite typical for bearing devices of turbomachines for various purposes.

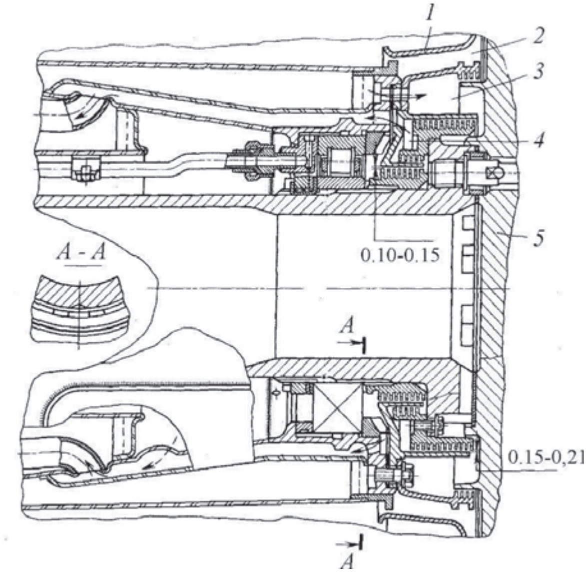


Fig. 4.21. Turbine support with reduced heat input to the inner ring of the bearing [15]

Reducing the heat flux entering the bearing from adjacent, hotter parts can be achieved by reducing the wall thickness at the transition points from the housing flanges to the cup, as well as by installing protective screens, which are sheet steel partitions concentric with the turbine shaft, between which air is blown, as well as screens that have heat-shielding coatings (asbestos, fiberglass).

Reducing the heat flow from the shaft to the bearing is often achieved by installing an additional sleeve between them, which acts as a thermal choke, or by making the shaft seating surface on which the inner ring of a ball or roller bearing is mounted discontinuous. Such a surface creates increased resistance to heat transfer from the heated shaft to the bearing.

With significant axial loads on the rotor (this is often found in the supports of free power turbines), supports consisting of **two or three parallel operating ball bearings** are sometimes used (Fig. 4.22).

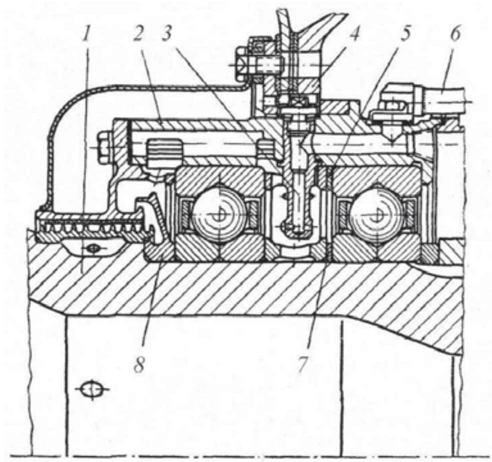


Fig. 4.22. Support with two parallel operating radial-thrust bearings [15]:
1 – trunnion; 2 – cup; 3 – nozzle; 4 – support body; 5 and 7 – calibrated rings; 6 – oil supply tube; 8 – adjusting ring

The use of two (or three) ball bearings to absorb axial load is possible under the following conditions:

– the diameters of the tracks of the outer and inner rings must differ by no more than 5 microns;

– the runout of the ring ends should not exceed 5 microns;

– the contact angles in a set of bearings must not differ by more than 2°;

– the fit of the rings on the shaft and in the housing must be the same [15].

To achieve approximately uniform load distribution between the bearings, the length of the spacer rings 5 and 7 between their inner and outer rings is selected so that all working clearances under the action of axial load are selected simultaneously and the balls of both bearings simultaneously come into contact with the rings.

Oil is supplied to the bearing under **an excess pressure of 0.35-0.55 MPa**, which is necessary to ensure a given flow rate through the calibrated nozzle holes.

There are two main requirements for installing nozzles in bearing devices: reliable fastening to the housing and ensuring the required angle of exit of the jet onto the bearing. The axis of the jet must be directed directly onto the raceways – into the gap between the ring and the separator. If oil gets onto the separator, it will be thrown away from it by inertial force, and cooling (as well as lubrication) of the bearing will not be provided.

When designing bearing devices, it is necessary to provide through-sections for free drainage of the oil-air emulsion into the pumping line. If the bearing is installed in a sealed cavity, then free drainage is carried out from both sides (see Fig. 4.20). Oil is drained by the pumping sections of the oil pump. Given the significant volume of oil, several such sections are used. Their number is set, for example, by the number of supports. Together with the injection section, they form a single oil unit.

* * *

Flexible rotor supports have some design features compared to rigid rotors.

If the rated speed of a turbomachine rotor exceeds the first critical speed, then this rotor is called **flexible** and, therefore, there is a need to reduce the amplitude of the bending vibrations of the rotor when gaining power and passing through the critical speed. For this purpose, elastic, damper, and elastic-damper supports are sometimes used.

In these supports, the mechanical energy of vibrations is converted into heat due to friction. The amplitude of shaft vibrations decreases, and the transition through the critical speed does not lead to the destruction of the supports.

The simplest to manufacture and assemble are **elastic supports**. Fig. 4.23 shows an example of such a support, where the elastic element is one or more rings 2 with protrusions. The rings are fixed between the outer ring of the bearing 1 and the housing 3 and are deformed during shaft vibrations. Such supports have a low damping capacity, since the vibration energy is dissipated only due to friction between the displaced metal layers in the rings 2. Such supports are also not able to transmit axial forces.

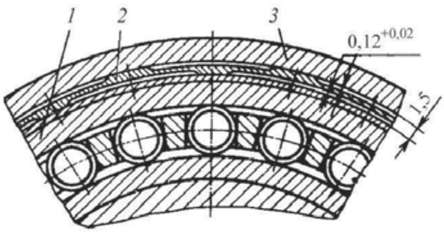


Fig. 4.23. Elastic support:
1 – outer ring of the bearing; 2 – elastic ring; 3 – bearing housing

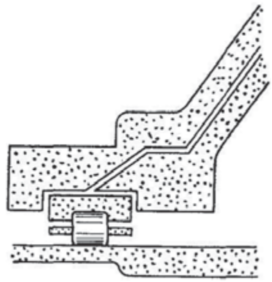


Fig. 4.24. Scheme of the damper support of the Conway aircraft engine [15]

In **damper supports**, the bearing is separated from the housing by a thin (up to 0.2 mm) layer of oil, and under load, this layer can be squeezed into the end gap. Fig. 4.24 shows the scheme of the damper support of the Conway engine. The oil is supplied under pressure and forms a film in the gap between the outer ring of the bearing and the housing. The unbalanced force of the rotor acting on the resistance squeezes out the oil film, while the amplitude of the rotor oscillations decreases. The intensity of vibration can thus be reduced by 60%, which increases the service life of the bearings. A similar support in various designs is widely used in gas turbine engines.

The support shown in Fig. 4.25 has good damping capacity. Vibration damping is carried out by squeezing out oil films located between steel strips 5. Oil is continuously supplied under pressure to the package of thin steel strips, the layout of which is shown in the same figure. The number of layers of strips can reach 20; the radial movement of the bearing relative to the housing is 0.15-0.3 mm. The stiffness of the support is determined by the hydraulic resistance in the gaps between the strips and depends on the speed and viscosity of the oil.

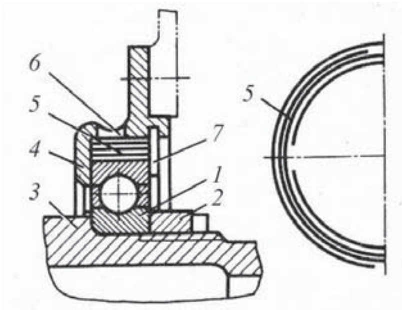


Fig. 4.25. Damper support:

1 – ball bearing; 2 – nut, 3 – compressor trunnion; 4 – bearing housing; 5 – package of thin steel strips; 6 – hole in the housing for supplying oil under pressure to the package; 7 – ring

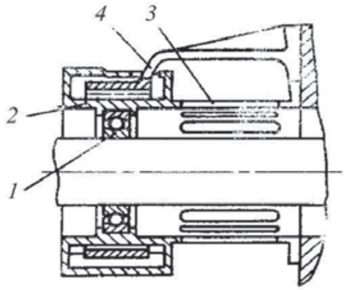


Fig. 4.26. Elastic-damper support [15]

Sometimes the elastic support and the damping oil device are combined, thereby creating an **elastic-damping support**. An example of such a support is the design in Fig. 4.26. The ball bearing 1 is placed in the housing 2, which is mounted on an elastic cylinder 3. Housing 2 is separated from the fixed housing 4 by a layer of oil. The rotor vibrations are damped due to the friction of the adjacent layers of the elastic element between themselves and from the extrusion of the oil film, which is located between the housings 2 and 4.

Elastic and damping supports allow the bearings to move relative to the housing within 0.1-0.2 mm. Obviously, this should not lead to the blades catching on the housing, so damping elements are usually installed on the front and rear supports of the three-support rotor of a gas turbine engine, where the gaps between the blades and the housing are the largest. In the middle bearing, the use of damping elements is impractical, since short blades require small radial gaps.

Since the critical rotor speed decreases with decreasing support stiffness, it becomes possible to control the critical speed so that it does not approach the operating speed during rotor rotation. Changing the support stiffness is achieved, for example, by controlling the oil pressure.

4.2. Supports with plain bearings

Plain bearings are used mainly in medium- and high-power gas turbine engines with a long service life (10,000-100,000 hours), as well as in steam turbines. Their use is typical for shafts of significant diameters (over 350-400 mm), for which high-speed rolling bearings are not produced by the industry.

The main advantages of plain bearings (compared to rolling bearings) include:

– significant service life (thanks to hydrodynamic lubrication and low wear, the service life of plane bearings is 30,000-50,000 hours or more);

– significant radial and axial loads can be sustained;

– less sensitivity to wear and shock load;

– lower noise level in operation;

– smaller radial size and weight.

Plain bearings are non-standardized; their design is largely determined by the traditions of manufacturing companies.

The structural scheme of **a typical journal (support) bearing** is shown in Fig. 4.27.

The shaft neck 1 is placed in the bearing liner 2 with a small gap, into which oil is supplied by a pump from the oil tank through channel 9. It passes between the neck and the Babbitt filling 10 of the liner, forming an oil film on which the shaft rotates under normal conditions. This eliminates contact between the metal surfaces of the shaft and the liner.

The used oil flows through the gaps from the ends of the liner into the bearing housing 7, from where it is directed by gravity into the oil tank.

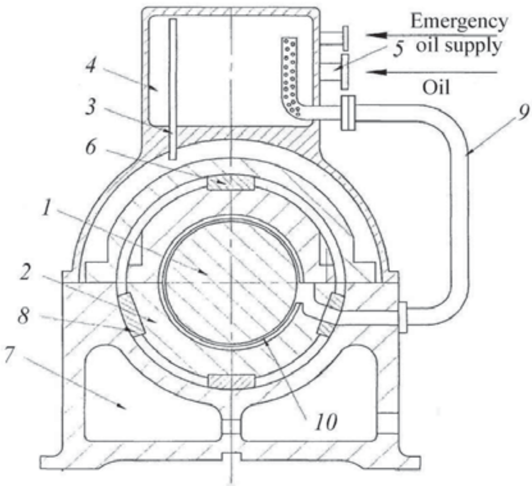


Fig. 4.27. Scheme of a journal bearing:

1 – shaft neck; 2 – bearing liner; 3 – overflow pipe; 4 – emergency tank; 5 – oil supply pipe; 6 – upper thrust pad; 7 – crankcase; 8 – support pads; 9 – oil supply pipeline to the bearing liner; 10 – babbitt filling of the liner

The position of the shaft neck, and therefore the entire rotor of the turbomachine in its housing, is determined by the positions of the liners. To install the liner in the desired position, three pads 8 are used on the lower half of the liner. The upper pad 6 serves to tightly clamp the liner in the bearing housing. Adjusting shims are installed between the liner and the pads for precise centering of the liner bore.

Emergency tanks 4 can be installed on the turbine bearing covers, which are continuously filled with oil through the oil line 5. Excess oil from them flows through the overflow pipe 3 into the bearing housing. In the event of an emergency stop of oil supply from the oil pumps, the protection system disconnects the current generator from the network (the scheme is considered using the example of a turbine that drives the current generator) and turns on the emergency lubrication pump, which is driven mechanically by the turbine rotor and supplies additional oil from the drain manifold to the emergency bearing tanks.

Lubrication of the shaft neck during the period of deceleration of the shaft rotation (during coasting) is carried out from the emergency tank through oil line 5 through a throttle washer with a specially selected hole size, which ensures a reduction in oil consumption with deceleration of the rotor rotation.

Let us consider **the principle of operation of a journal bearing** (Fig. 4.28).

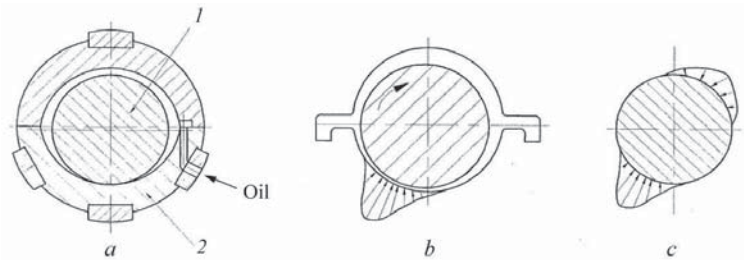


Fig. 4.28. Scheme of operation of the journal bearing

The stationary shaft 1 lies on the lower half of the liner 2, contacting it with its lower surface.

If you organize a flow of oil through the bearing and start rotating the rotor, the oil will begin to stick to the surface of the shaft neck and be captured by it. Since the oil has viscosity, it will be captured layer by layer under the shaft neck, and as a result, at a certain speed of rotation, a stable oil film will appear between it and the liner. In this case, the lifting force that occurs in the bearing is formed due to the pressure that occurs in the oil layer. A journal bearing that works on this principle is called a **hydrodynamic**. If you measure the pressure along the circumference with a "floating" neck, the pressure distribution will look like the one shown in Fig. 4.28, *b*.

The oil pressure in the narrow wedge-shaped gap (wedge) between the neck and the liner, starting from the drain channel, will increase, reaching a maximum in the radial cross-section, which is slightly below the minimum (since the center of the neck is shifted in the direction of rotation (see Fig. 4.28, *b*)). If the pressure distribution is replaced by an equivalent force, then its vertical component will counteract vertical forces, in particular the weight of the rotor, and its horizontal components will counteract all horizontal forces.

The friction force between the layers of oil depends on their relative speed, so a stable oil film is formed only at a sufficiently high speed, when the oil layer adhering to the surface of the shaft neck begins to capture the adjacent layer. At low speeds, the oil film either does not form at all or periodically disappears. Accordingly, there is constant or periodic contact of the shaft neck with the inner surface of the liner. In this mode of operation, semi-dry (semi-liquid) friction occurs, in which, if special measures are not taken, a significant amount of heat will be released, causing wear of the liner surface and, most importantly, the shaft neck surface. It is to reduce friction at low speeds that **the inner surface of the liner is filled with an antifriction alloy – Babbitt**.

After the turbomachine stops, to prevent thermal bending, the rotor is slowly rotated by the turning gear. In this case, a semi-dry friction mode occurs between the shaft neck and the surface of the liner. For powerful turbomachines with heavy rotors, this will inevitably lead to rapid wear of the Babbitt casting and a change in the shape of the liner bore. In addition, significant power of the turning gear will be required. Therefore, powerful turbomachines with heavy rotors are equipped with **hydrostatic rotor lift** (hydrolift): several holes are made in the lower half of each liner, into which oil is supplied at a pressure of 4-8 MPa. Under its influence, the shaft rises by 20-60 microns, ensuring ease of inclusion of the turning gear and eliminating liner wear. When working on hydrolift at low speeds, the journal bearing works as a hydrostatic bearing. When the speed increases and a stable hydrodynamic oil wedge is formed, the hydraulic lift is turned off.

Two important principles of designing journal bearings can be noted: the inadmissibility of cutting grooves in the lower loaded part of the liner (they reduce the bearing capacity, as they tear the oil film and reduce the excess pressure in it to zero) and the organization of such an oil supply, in which it first enters the upper unloaded part of the liner, and then, having passed half the circumference, creates an oil wedge in the lower part.

Existing **designs of journal bearing liners** can be divided into single-wedge, double-wedge, and multi-wedge (segmented).

A single-wedge bearing has **a cylindrical bore** (Fig. 4.28, *b*). When the shaft neck rotates, one supporting wedge is formed under it. The principle of operation of a single-wedge bearing is discussed above. Its design is the simplest, and therefore it is widely used for low-power turbomachines.

With the increase in the power of the turbomachine and the increase in the circulating disturbing forces, the operation of a single-wedge bearing does not ensure the stability of the shaft rotation on the oil film, and intense vibration occurs. One of the measures to combat it is the use of a double-wedge bearing, which has **an elliptical (lemon) bore** (see Fig. 4.28, *a*).

In a double-wedge bearing, an oil wedge is formed not only in the lower but also in the upper half of the liner (Fig. 4.28, *c*). As a result, a force appears that acts on the upper part of the shaft neck and prevents the appearance of intense vibration.

The required ovality of the liner is calculated very accurately and is practically carried out as follows (Fig. 4.29). A gasket is installed between the halves of the liner in the connector, the thickness of which is selected in such a way that after performing cylindrical boring, removing the gasket, and connecting the halves of the liner, the required ovality is obtained. The manufacturing accuracy of the gasket is +0.02 mm in thickness, and the liner is bored with a tolerance of +0.05 mm.

Several **requirements** are imposed on the design of thrust bearings.

1. The bearing must operate reliably, preventing excessive heating of the oil and wear of the liner. When the liner wears out, the vibration characteristics of the entire rotor change, and intense vibration may occur. The oil in the bearing is

heated by friction forces between the layers of oil in the film and by heat coming along the shaft from hot parts of the turbomachine.

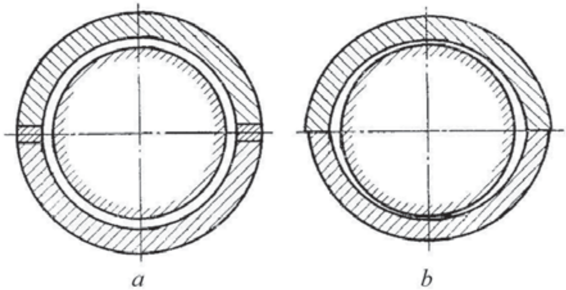


Fig. 4.29. Scheme of a liner with a lemon bore:
a – liner with gasket before boring; *b* – liner assembled without gasket

The amount of heat transferred along the shaft usually does not exceed 10-20% of the heat released in the oil layer. When heated to 115 °C, the inner surface of the liner, filled with Babbitt, softens and its resistance to wear and deformation deteriorates sharply. At 150 °C, the integrity of the oil film is broken; at 350 °C, the Babbitt filling melts with a serious accident of the entire engine.

Therefore, to maintain the temperature level of the bearing, all operating instructions strictly stipulate the oil temperature at the inlet (usually 35-45 °C), the normal temperature at the outlet (about 65 °C) and the limit oil temperature at the outlet (about 75 °C), at which an immediate stop of the turbomachine is required. The temperature of the Babbitt filling, controlled by thermocouples, should not exceed 100 °C.

2. The bearing must be vibration-resistant and have sufficient damping capacity. Vibration resistance is understood as the ability not to react to random disturbances that are always present in a turbomachine.

All variable forces acting on the shaft line and causing it to vibrate are ultimately damped in the oil layer of the bearings. Therefore, the higher their damping capacity, the less vibration occurs.

3. The design of the journal bearing should ensure a slight change in the radial clearances in the turbomachine in all operating modes (in idle mode and under any loads). For this, it is necessary that the shaft neck float on the oil wedge is insignificant. This will allow the turbomachine to have small radial clearances and small losses from leakage through them.

4. Friction losses in the bearing should be negligible. With shaft neck sizes reaching 600 mm and more, friction power losses can reach 200 kW per bearing.

The listed requirements are contradictory in the sense that satisfying one of them in full makes it is not possible to fulfill the others. For example, to ensure high vibration resistance and economy, it is necessary to have small gaps between the shaft neck and the liner and a small float of the shaft neck. At the same time, the oil heating increases, and the bearing operation becomes sensitive to distortions of the shaft neck relative to the liner; there is a possibility of semi-dry friction and jamming, that is, reliability decreases.

The design features of supports with journal plain bearings are mainly due to their location relative to the heated housings and rotor sections; however, the basic design of the bearings (discussed earlier) is the same for all components of a gas turbine engine.

Journal plain bearings are made as fixed or split. Fixed bearings have a small diameter and are very convenient for axial assembly of a turbomachine. Split bearings are more common in gas turbine engines, which facilitate assembly and quality control.

Fig. 4.30 shows a typical design of a journal bearing liner used by Power Machines (LMZ) [15]. This steel liner consists of two halves, 13 and 14, and is installed in the bearing housing on four pads 8, three of which are attached to the lower 14 half of the liner with screws, and one to the upper 13 half of the liner. Steel washers 7 under the pads are used to center the liner relative to the housings.

In addition to the clearance Δ, the normal operation of the bearing is affected by the tension between the cover and the liner. In bearings of powerful gas turbine engines, similar to those shown in Fig. 4.30, the cover is installed with a tension of 0.15-0.20 mm.

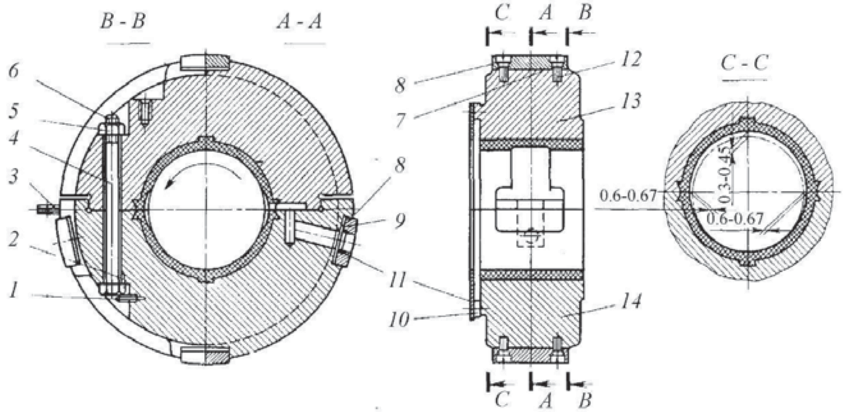


Fig. 4.30. Liner of the thrust bearing of the Power Machines (LMZ):
1, 2 – set screws, 3 – lock washer; 4 – connecting bolt; 5 – nut; 6 – cotter pin; 7 – gaskets with a hole thickness of 1; 0.5; 0.15 and 0.1 mm; 8 – mounting pad; 9 – oil diaphragm; 10 – screw; 11 – oil deflector ring; 12 – screw; 13 – upper half of the liner; 14 – lower half of the liner

The halves of the bearing liner are connected during assembly by four bolts, two of which are made with a zone to allow for precise installation of the upper and lower halves of the liner relative to each other. The liner is held against rotation in a circle and against displacement in the axial direction by a lock washer 3.

Oil is supplied to the bearing through a metering diaphragm 9, the diameter of which is selected according to the oil flow rate found from the calculation. The oil pressure is 0.14-0.18 MPa. The oil is supplied to the unloaded part of the bearing so that when the shaft rotates, it is captured in the direction of rotation in the wedge-shaped gap. In this case, the oil first flows along a wide groove made in the upper liner, cooling it. An oil deflector ring 11 is placed on the side of the final labyrinth seal of the turbine or compressor housing.

In cases where significant shaft deformations are possible or the installation is not carried out accurately, it is recommended to use **self-aligning bearings**. An example is the journal bearing of the Turboatom company, used for 50 MW engines (Fig. 4.31) [15].

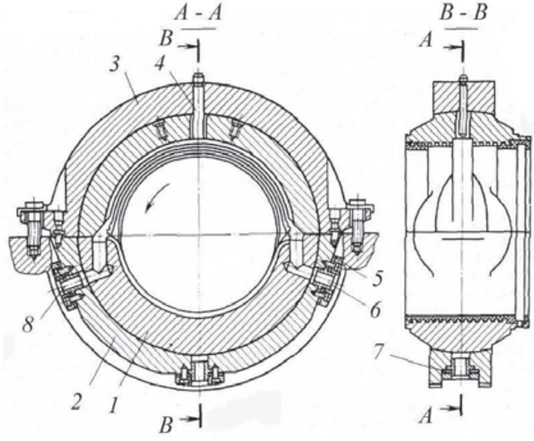


Fig. 4.31. Journal bearing of a 50 MW gas turbine engine:
1 – liner; 2 – support half-ring; 3 – cover, 4 – pin, 5 – support pad; 6 and 8 - bushings; 7 – gasket

The bearing liner 1 has a design similar to that discussed above, with one significant difference: it is not installed directly in the bearing housing, but in an intermediate cage (consisting of a cover 3 and a support half-ring 2), which is then mounted in the bearing housing. The contact of the cage with the liner is made on a spherical surface. Therefore, the liner has the possibility of small turns, which reduces the misalignment of the shaft neck relative to the liner and reduces their mutual wear. The liner is centered in the housing by three pads 5 and gaskets 7.

The housings and covers of journal bearings of gas turbine engines can be cast from gray cast iron or steel, cast-welded, or welded. For bearing liners, materials with good antifriction properties, sufficient strength, good fit to the shaft neck, and high wear resistance are used. In order to save non-ferrous metals, liners are usually made of cast iron or steel, and the inner surface of the liners is filled with a thin layer of Babbitt. In general, the recommended thickness of the filling layer is 1% of the inner diameter of the liner; for bearings of powerful gas turbine engines, this is approximately 1.5-2 mm. To better secure the Babbitt layer, grooves are milled or drilled in the liners, which have a dovetail shape in cross section.

For bearings of gas turbine engines, high-tin babbits are usually used. For bearings of high-speed small-sized engines, lead bronze or silver can be used as an antifriction layer [15].

* * *

The thrust bearing serves to perceive the resulting axial force applied to the rotor and transmit it to the housing parts. At the same time, it fixes the axial position of the rotor in the turbomachine and provides the necessary axial clearances in its flow part and seals.

Mitchell thrust hydrodynamic bearings[1] with movable self-aligning pads have found wide application in modern gas turbine engines [15].

The schematic diagram of this type of thrust bearing is shown in Fig. 4.32. A thrust disk (collar) 4 is made on the turbine shaft 1, which, through an oil layer, rests (depending on the direction of the axial force) on pads 3 or 5, which rotate about the swing ribs 8. Lubricating oil is supplied from the pump to the manifold 6, from which it is supplied to the thrust pads 3 through the holes in the mounting ring 2. An oil film is formed between the thrust pad and the thrust collar, which prevents their contact.

Oil supply to the pads 5 can be carried out either in the same way (from another collector), or by bypassing the oil into the chamber of these pads. The shaft at the point of exit from the bearing housing is sealed, and the oil, which is

[1] Australian engineer George Mitchell (1870–1959) patented this design in 1905.

supplied to the liner, fills the inner cavity of the liner and exits into the bearing housing through the holes in the upper half of the liner.

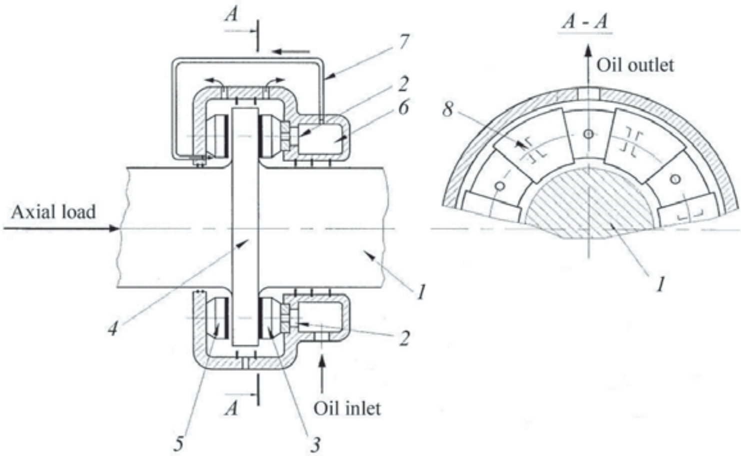


Fig. 4.32. Scheme of a thrust bearing

Let's consider **the principle of operation of the thrust bearing** in Fig. 4.33.

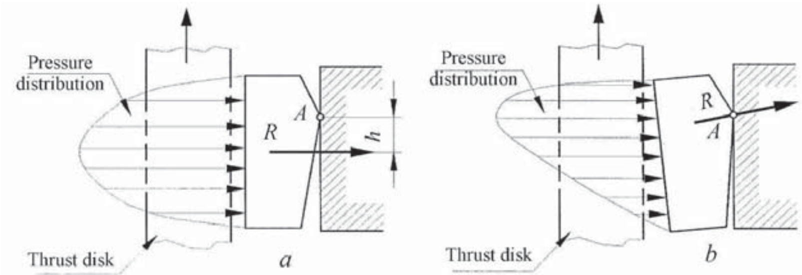


Fig. 4.33. Position of the thrust pad:

a – unstable position; b – stable position

Before the start of rotation, the bearing liner is filled with oil. With the start of rotation, the oil that adheres to the bearing ridge is captured layer by layer under the pad and, since the free axial displacement of the ridge is prevented by the axial force applied to the rotor from the pad (stop), a certain pressure distribution occurs on the surface of the pad. For the simplicity of further considerations, it can be replaced by an equivalent force *R*, which is applied to some point of the pad and rotates it in the same way as the distributed pressure.

If at some point the pressure distribution on the surface of the pad is as shown in Fig. 4.33, *a*, then its state will be unstable, since the resultant *R* will work as a lever with the shoulder *h* around the rib *A* of the pad support. Therefore, the pad will begin to rotate. The pressure distribution will change. Due to the fact that the gap at the input part of the pad will increase, the pressure here will decrease; on the output part of the pad, on the contrary, it will increase. The force *R* itself will not change, since the axial force applied to the rotor has not changed. The pad will rotate until the force vector *R* passes through the support rib *A* and the shoulder *h* becomes zero (Fig. 4.33, *b*). This will be the stable position of the pad, in which a narrowing channel is formed between the surfaces of the ridge and the pad – a wedge through which fresh oil passes, which comes from hole 2 (see Fig. 4.32). The opening of the wedge automatically changes with the change in the axial force acting on the rotor: the higher this axial force, the more the wedge opens and the higher the resultant *R* on each pad will be.

The considered scheme of operation of the thrust bearing assumes that the axial force acting on the rotor is always applied in one direction. However, in real conditions in modern powerful turbomachines, the direction of the axial force can change when switching from mode to mode. Therefore, all thrust bearings are made with two rows of thrust pads located on both sides of the rotor thrust collar (Fig. 4.34).

When a bearing with two rows of thrust pads is operating, oil wedges appear on the pads of both rows. In this case, the main (working) pads are loaded not only by the axial force Rz applied to the rotor, but also by additional forces applied to the thrust collar from the side of the additional (setting) pads. The amount of possible shaft movement between the rows of thrust pads is called the axial run-out in the thrust bearing. The larger the run-out, the smaller the loading from the side of the setting pads. However, the run-out cannot be made excessively large, as this can lead to interference in the flow part and the appearance of significant shock loads on the pads when the sign of the axial force changes.

A small axial run-up is also dangerous, and not only because additional force is generated from the idle row of pads. With a small run-up and the necessary rotation of the pads according to the load, the gap between them and the ridge decreases. At the same time, the oil consumption under the pad decreases, and it heats up a lot. Following the oil, the pad heats up, and the surface becomes flat. As a result of overheating, the oil film loses its load-bearing capacity and disappears. The ridge comes into contact with the pads, and due to the release of a significant amount of heat, either instant melting of the Babbitt on the thrust pads occurs, or their rapid wear occurs. An axial shift of the rotor occurs, and if it is greater than the axial clearances provided in the turbomachine, the blades catch on the housing, which leads to a serious accident.

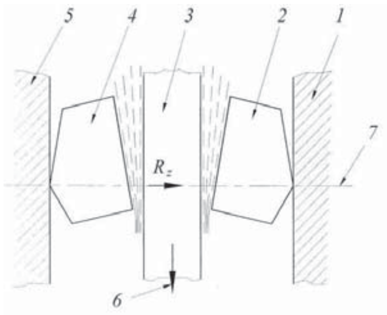


Fig. 4.34. Thrust pad arrangement scheme:

1, 5 – liner body; 2 – working pad; 4 – setting pad; 6 – direction of the circular velocity vector of the point of the collar that contacts the pad; 7 – turbine axis

Therefore, the axial runout in the thrust bearing is set so that the thickness of the oil film at the exit from the pads is not less than 40-60 microns, the average pressure on the pad does not exceed 2-4 MPa, and the temperature of the Babbitt casting is not more than 100 °C. Usually, when assembling the thrust bearing, an axial runout of 0.5-0.6 mm is set. The operation of the thrust bearing must be absolutely reliable, since its failure entails, if the appropriate protection does not work, the most serious accident.

Thrust pads are the main parts of the thrust bearing, which determine the reliability and quality of its operation. The bearing capacity of the thrust bearing depends on the specific pressure, the shape of the pads, the method of their support, their total number, and other factors. The radial rib to which fresh oil is supplied is called the input 3, and the opposite one is called the output 4 (Fig. 4.35, *a*). The central angle formed by the input and output ribs is 28-35°.

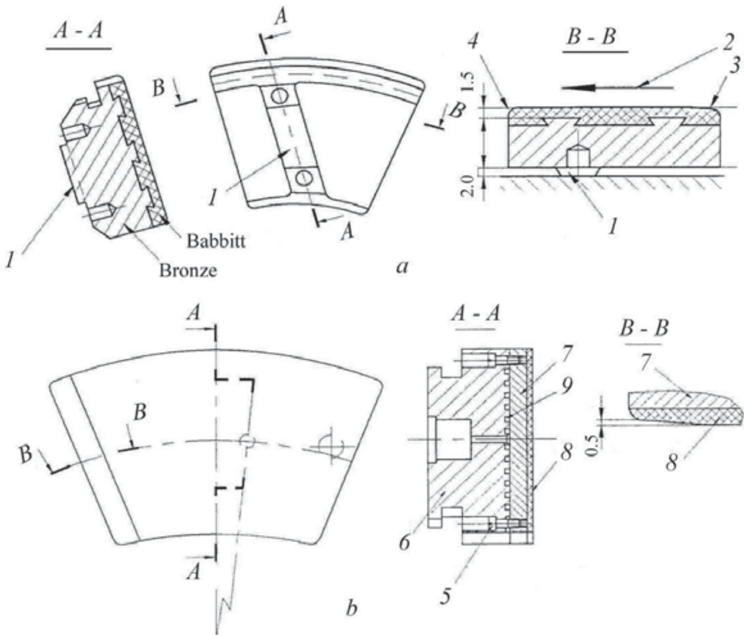


Fig. 4.35. Thrust pads:

a – conventional pad; *b* – laminar pad; 1 – turning edge; 2 – direction of rotation of the thrust disk; 3 – input edge; 4 – output edge; 5 – screw; 6 – base; 7 – copper gasket; 8 – babbitt filling; 9 – oil channels

The surface of the pad facing the thrust ridge is filled with Babbitt. For a better fit of the Babbitt filling in the pad, dovetail-type grooves are made. For a complete fit of the surface of the babbitt filling to the rotor ridge, they are periodically sanded, and therefore, the thickness of the babbitt filling gradually decreases. When it decreases to an unacceptable value, the pads are refilled with a new layer of Babbitt.

The role of Babbitt in thrust and journal bearings is different. In journal liners, Babbitt plays the role of antifriction material, necessary for operation in semi-dry friction mode. In most thrust bearings, the pads are located in an oil bath, so the semi-dry friction mode practically does not occur. However, with a sudden increase in the axial force to an unacceptable value, when the thrust collar comes into contact with the pad surface, babbitt is almost instantly melted, and the rotor is sharply moved by the amount of melted babbitt. This axial displacement of the rotor is used as a signal for protection, which instantly turns off the engine.

Each thrust pad is suspended on one or more pins and contacts the support ring only along the swing edge 1, around which it rotates freely when the load changes. The swing edge is made parallel to the output edge of the thrust pad. This ensures the uniformity of the gap along the output edge when the pad is rotated, a more uniform distribution of pressure over the pad surface, and lower wear. The swing edge is not located in the middle of the pad, but closer to the output edge. This, on the one hand, eliminates the possibility of the pad tipping over with the oil wedge closing, and on the other hand, when the pad is rotated, the gap at the output does not decrease so much that the output part of the pad wears out.

During normal operation of the thrust bearing, the main heat release occurs in the oil layer. Part of this heat is carried away by the oil flow, and part is transferred to the pad. The layers of the pad adjacent to the Babbitt filling heat up and expand more than the layers at the rocking ribs. As a result, the initially flat surface bends. The higher the temperature unevenness along the thickness of the pad, the more it deforms and the lower its bearing capacity. To reduce temperature unevenness, the pads are made of materials that have high thermal conductivity and a low coefficient of expansion. Copper-based materials are used as such materials - bronze and brass.

The bearing capacity of **a laminar pad** is almost twice as high as that of a conventional pad (Fig. 4.35, *b*).

A thin copper gasket 3 is attached to the steel base 2 with special screws 1, onto the surface of which a layer of Babbitt is soldered. Channels 5 are made in the base, through which oil passes, cooling the copper gasket. Thus, a small temperature difference is created along the thickness of the gasket, and it deforms little. This allows not only to increase the bearing capacity of the pad, but also to install it not on the edge, but on a point support (more precisely, a "penny"). Such a pad is held in the housing by means of ring pins.

With the same total surface area of the pads, which provide a certain average specific pressure, it is possible to perform either a large number of small or a small number of large pads. With a small number of them, the dimensions of the bearing increase, and with a large number, cooling becomes difficult: with a dense arrangement of the pads, the hot oil from under them does not have time to flow into the housing and is mixed with the cold oil supplied under the next pad. Calculations and operating experience have shown that the optimal number of thrust pads in a bearing is from six to eight (in practice, from 6 to 12 pads are used).

Thrust bearing pads are made of tin bronze; the working surface is filled with Babbitt 1-2 mm thick.

One effective way to ensure uniform load distribution across the thrust pads during bearing operation is to use **the Kingsbury lever equalization system**[1] [15]. A sample of a thrust bearing of this type is shown in Fig. 4.36.

The bearing is made symmetrical, with one thrust disk 12 located between the thrust pads 5 and 6. The bearing liner 8 is installed in the bearing housing 10 and is held from rotation by lock washers 20. The bushings 7 and 16 are installed in the bores of the liner, which consists of two halves, inside which closed circular lever equalizing devices are placed.

[1] Named after the American engineer Albert Kingsbury (1863–1943), who first proposed such a design.

The leveling device consists of the blocks 14 and 15. The blocks 14 have cylindrical protrusions 25, the guides of which abut the holder. Pin 13 is used to radially fix the blocks. The shoulders of the blocks 14 are supported by the shoulders of the blocks 15, held in the liner by pins 17. The pins of the blocks enter the axial grooves that do not prevent their axial movement along the shaft axis. The thrust pads 5 and 6 are held in the holder by figured protrusions that enter the bores in the holder on one side and are held by a split ring 1 on the other side. The stops 23 and 24 are pressed into the thrust pads and the upper blocks 15.

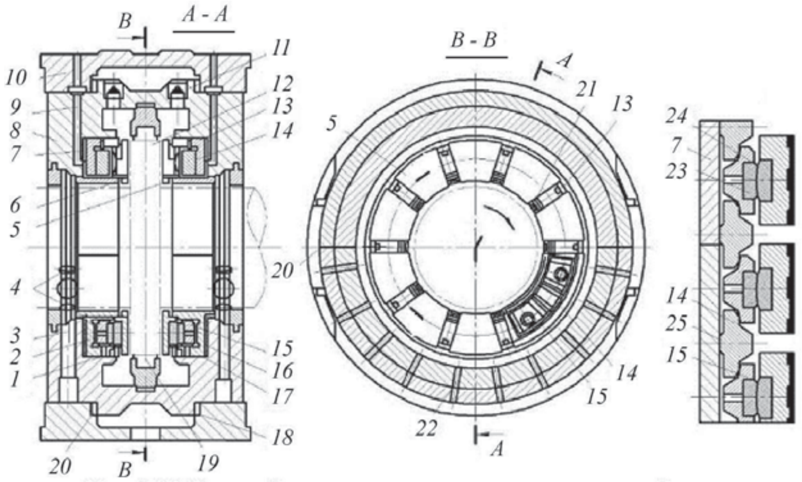


Fig. 4.36. Thrust bearing with equalizing devices [15]

The axial force acting on a certain thrust pad is transmitted from the stop 24 to the stop 23 of the block 15, then through the shoulders to the block 14, and from there to the holder 7.

All the blocks create a closed circular system. Such a system automatically maintains the same forces on the working pads. For example, if the force on one of the pads increases, it moves in the axial direction. Its movement through the blocks is transmitted to the other pads, which begin to move towards the thrust disk, taking on part of the load and unloading the overloaded pad.

Such a system can reduce the unevenness of the load on the pads several times, but the equalizing device will be effective only if it is carefully executed. All contact surfaces must be hardened (to reduce deformation and abrasion rate), carefully ground and fitted, and have a small contact area to reduce friction forces.

Oil is supplied to the thrust pads from the bore 3 through three holes 4 in the lower half of the liner directly into the cavity of the thrust pads. Oil flows out of the bearing through holes in the upper half of the liner, into which throttle diaphragms 11 are inserted to pressurize the oil in the bearing chamber. The side surface of the thrust disk is sealed, and the chamber 19 is drained through holes 22.

Setting rings 2 are used to adjust the axial runout. The position of the liner in the bearing housing is changed using ring 18. Channels 9 and drilling 21 are used to supply oil from emergency tanks.

Thrust bearings are usually combined with journal bearings. A typical design of a combined bearing liner for a shaft with a diameter of 170 mm is shown in Fig. 4.37 [15]. The steel liner 3 of the bearing is installed in a cage 1 made of two halves. The liner has a spherical surface, due to which its self-centering in the cage is achieved. The locking pin 8 keeps the liner from turning. The tension of 0.02-0.04 mm (and up to 0.04-0.08 mm for bearings with a diameter of 320-360 mm) between the spherical surfaces of the liner and the cage prevents axial displacement of the liner. The selection of adjusting rings 9 provides the necessary axial clearances in the flow part of the turbomachine. The thrust part of the liner is made as one unit with the support part. It is located cantilevered in relation to the bearing housing; The rotation of the liner is prevented by spring 2.

The thrust bearing disc is placed on the shaft with a tension of 0.05-0.07 mm. On one side, the disc rests against a shoulder machined on the shaft. Possible movement of the disc in the other direction is prevented by a split ring installed in a groove on the shaft. This ring, in turn, is compressed by a solid ring that surrounds it with a tension of approximately 1 mm.

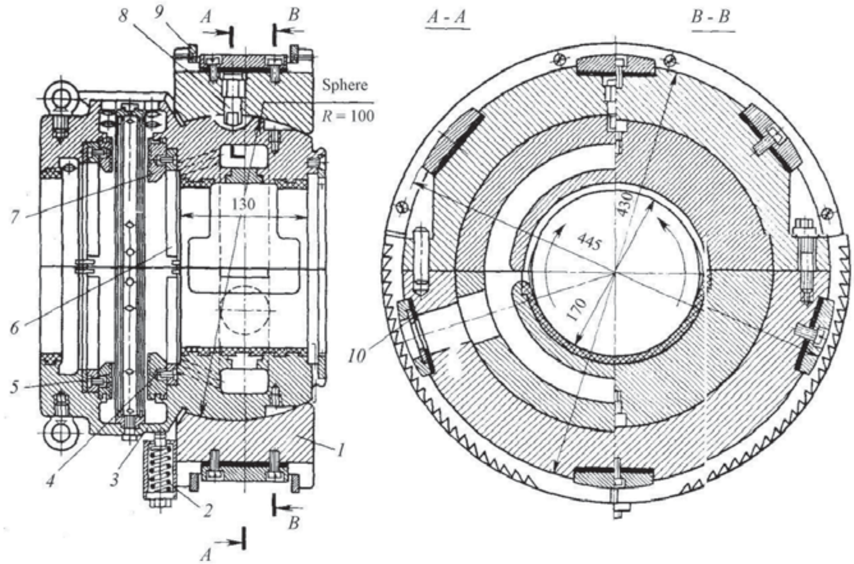


Fig. 4.37. Thrust bearing liner of the Power Machines (LMZ)

This bearing uses 10 thrust pads (see also Fig. 4.35, *a*). They rest on half rings 6 and are held by fingers 4 (two fingers per pad). These fingers freely enter the pad sockets. With this installation, the pad can swing around a rib parallel to the output edge. To provide an oil wedge, this rib divides an arc on the average diameter of the pad in a ratio of "three to two", with the longer part being on the side of the oil inlet to the pad.

The shaft is subjected to an axial force directed from left to right, and it is perceived by the working thrust pads 7. The setting pads 5 prevent the shaft from moving in the direction opposite to the axial force during operation. Such movements can occur during transportation of the turbomachine, as well as in variable and partial operating modes.

All the oil for the thrust bearing is supplied through the metering diaphragm 10 into the annular cavity, and from there through inclined (relative to the bearing axis) channels to the thrust pads. The oil is supplied to the setting pads through a separate pipe (not shown). The oil is drained from the bearing only through the upper part. This ensures that the working and setting pad chambers are filled with oil.

The combination of a thrust bearing with a radial-thrust ball bearing for the front rotor support of the low-pressure compressor of the Zorya-Mashproekt UGT25000 engine is shown in Fig. 4.38.

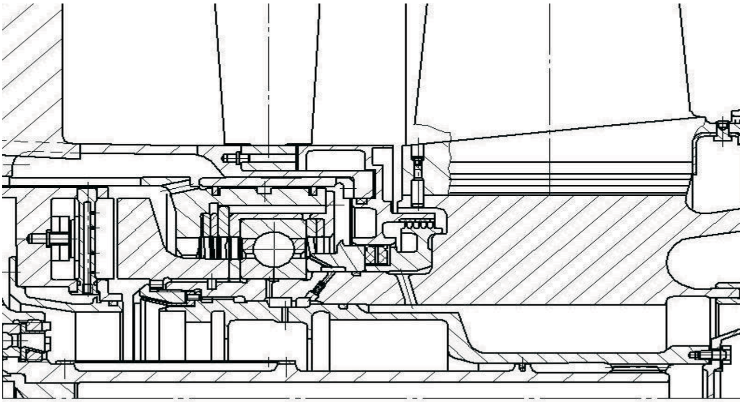


Fig. 4.38. Combined front support of low-pressure compressor of the Zorya-Mashproekt UGT25000 engine [20]

The ball bearing in this case is designed to absorb radial loads from the compressor rotor and, in part, axial loads that arise during the operation of the low-pressure turbocompressor and are directed against the movement of air.

To dampen vibrations that occur during rotor rotation and limit the amount of radial movement, the ball bearing is installed in an oil damper.

The magnitude of the axial load that a ball bearing can handle can vary depending on the operating mode of the gas turbine engine. When the axial load increases beyond a certain value, a thrust bearing comes into play, which is capable of handling a much larger axial force.

The thrust bearing consists of a housing in which eight self-aligning pads are evenly spaced around the circumference. Each pad with a supporting sphere rests on its own package of two flat springs. Oil supply nozzles are placed between the pads to create an oil wedge and cool the thrust bearing elements.

The principle of operation of the combined support is as follows: the outer ring of the ball bearing rests against a package of two pre-compressed flat spring rings; when the axial load (directed against the air movement) increases, the rotor of the low-pressure turbocompressor begins to move, transferring this load to the pads and thrust disk of the thrust bearing.

The oil supply to the ball bearing and the thrust bearing is carried out through different pipes. To ensure a more uniform supply of oil to the pads, it is supplied through two diametrically located pipes.

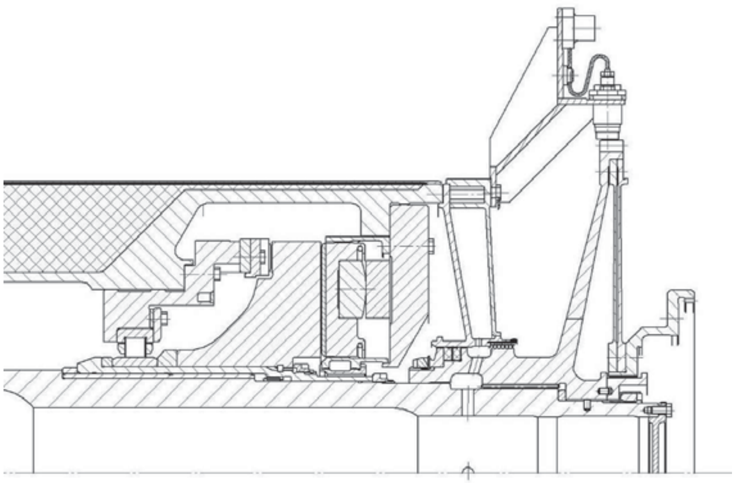


Fig. 4.39. Combined support of power turbine of the Zorya-Mashproekt UGT25000 engine [20]

Lubrication and cooling of the ball bearing is carried out by supplying oil from the end of the bearing, directed towards the compressor inlet. The oil is supplied to the gap between the inner ring and the separator at three points in equal portions.

The same engine uses a combination of a double-sided thrust bearing and a roller bearing for the rear support of the power turbine (Fig. 4.39).

Control tasks and questions

1. Name the main requirements for the structures of rotor supports.
2. Analyze the sequence of designing rotor supports.
3. Discover the main advantages and disadvantages of rolling bearing supports.
4. Explain the choice of the number of supports for the rotors of gas turbine engines.
5. Explain the methods of fastening rolling bearing rings in rotor supports.
6. Discover the design features of flexible rotor supports.
7. Discover the main advantages and disadvantages of plain bearing supports.
8. Explain the operation of a journal bearing.
9. Analyze the working principle of a thrust bearing.
10. Explain the operation of the Kingsbury lever equalization system.

5. SEALS OF GAS TURBINE ENGINES

When designing gas turbine engines, radial and axial clearances are installed between the rotors of turbomachines (compressors and turbines) and their stationary parts (stator). Unproductive flow of the working fluid through them leads to a decrease in the power of turbomachines, as well as a significant decrease in their efficiency. To prevent this, the radial and axial clearances between the rotor and stator in gas turbine engines are sealed (i.e., special devices are placed in them that minimize the leakage of the working fluid).

5.1. Sealing of peripheral radial clearances

The most responsible are the radial clearances between the tips of the blades and the housing of the turbomachine. The energy losses of the working fluid and, ultimately, the efficiency of the engine depend to a very large extent on the size of these clearances. The energy losses of the working fluid are caused by the circular flow of gas (or air) through the radial gap from the concave surface of the blade to the convex one, and in the axial direction - from the area of increased pressure in front of the blades to the cavity of reduced pressure behind them (Fig. 5.1) [11].

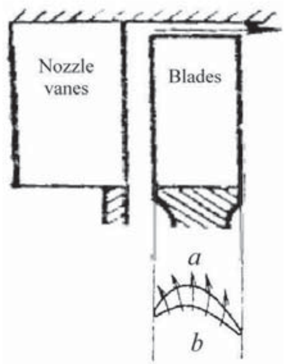


Fig. 5.1. Scheme of gas flows through the radial clearance of the turbine:
a – in the axial direction; *b* – in the circular direction

Reducing these radial clearances leads to an increase in the efficiency of the turbomachine. For example, according to [15], a decrease in the relative radial clearance (the ratio of the radial clearance to the blade length) by 1% leads to an increase in the compressor efficiency by 3%, which causes a decrease in fuel consumption in a gas turbine engine by up to 10%.

The value of the mounting radial clearance is determined taking into account the following factors: elastic and plastic deformations of the blades; thermal expansion of the housing, disk and blades; possible amplitude of oscillations of the housing and rotor; radial clearance in the bearings; technical tolerances for the manufacture of parts, as well as taking into account possible changes in the value of the mounting radial clearance during operation in various modes, including starting and stopping the engine.

Since it is impossible to accurately account for the magnitude and direction of the action of these factors, in practice, the mounting clearance is selected experimentally under the most unfavorable operating conditions. For example, for the turbine part of the engine, if you set the mounting clearance δ_M (Fig. 5.2, *a*), then at the start-up stage, due to the rapid heating of the relatively thin housing, the clearance will increase to δ_Z (Fig. 5.2, *b*). In a stable operating mode, as a result of heating and elastic deformation of the blades and disk, the radial clearance will decrease to δ_P (Fig. 5.2, *c*). When the engine stops, especially in an emergency, at the first moment, a large mass of cold air flows through the turbine, the housing cools faster than the blades and disks, and the radial clearance quickly decreases to δ_O (Fig. 5.2, *d*), and then the housing "sits" on the blades and the rotor jams. Therefore, the mode that determines the minimum value of the mounting clearance is the engine stop mode.

According to [15], the value of the mounting radial clearance in turbines is approximately 0.3-0.4% of the housing diameter.

Thus, to ensure reliable operation of turbines with the minimum possible radial clearances, it is necessary to reduce the shrinkage of the housing after the engine stops, for which the housing is made of steels with small coefficients of linear expansion, and ensure its effective cooling.

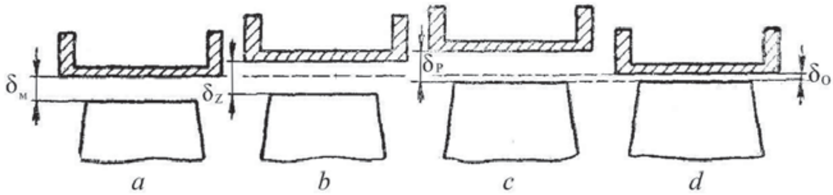


Fig. 5.2. Radial clearances between the housing and the tips of the turbine blades at different engine operating modes:

a – mounting clearance; *b* – clearance when starting the engine; *c* – clearance in operating mode; *d* – clearance when stopping the engine

For the same purpose, in the housing (or in the outer band ring) of turbines, metal-ceramic or honeycomb inserts are installed above the blades, which are easily triggered when the tips of the blades touch them, thereby reducing the load on the blades and preventing their breakage. As noted in [15], the use of metal-ceramic or honeycomb inserts leads to an increase in turbine efficiency by 2-3%.

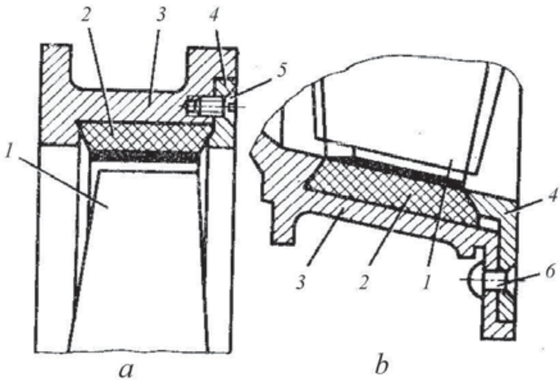


Fig. 5.3. Fastening of metal-ceramic inserts in the turbine housing [15]:

a – screw fastening; *b* – rivet fastening; 1 – blade, 2 – insert; 3 – housing, 4 – ring; 5 – screw; 6 – rivet

Metal-ceramic inserts (Fig. 5.3) are made of a conglomerate mixture of graphite, nickel and iron. The first (outer) layer is the basis of the insert and ensures its strength. It contains a small amount of graphite (6-8%). The second layer is working, easily activated, and contains more graphite than the first (12-16%). It should be noted that during operation, as a result of heat changes, vibrations, and other factors, cases of chipping of metal-ceramic inserts are observed.

More stable are **the honeycomb inserts** 3 (Fig. 5.4), which form a ring 2 above the blades 1. The inserts have thin walls, so the contact surface with the blades is approximately 10 times smaller than with a conventional labyrinth seal. Leakages through honeycomb seals depend less on the values of the gaps A and B than those that occur in gaps without seals (Fig. 5.5).

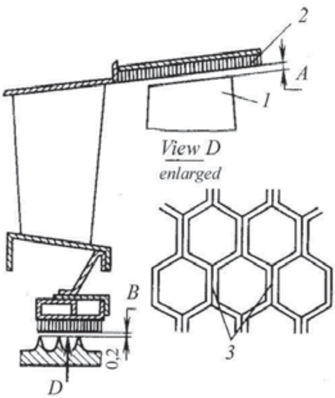


Fig. 5.4. Honeycomb seal in the turbine

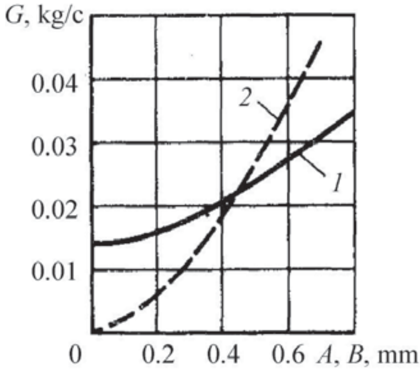


Fig. 5.5. Change in gas flow through gaps A and B (see Fig. 5.4):

1 – with honeycomb insert; 2 – with a regular smooth ring

In turbines, an effective means of reducing losses due to the flow of the working fluid through the radial gaps is **banded shelves on the blades** (Fig. 5.6). When installing blades with banded shelves, the turbine efficiency increases by approximately 1-2%. Creating **labyrinth combs** on the banded shelves or in the housing above the banded shelves leads to an increase in the turbine efficiency by approximately 1% [15]. The best effect is achieved in the first stages, where the blades are short and the relative losses due to flow are relatively large.

When designing turbines, special attention is paid to the correct location of the banding shelves relative to the nozzle device, which eliminates their protrusion into the flow and disruption of the smoothness of the flow part of the turbine.

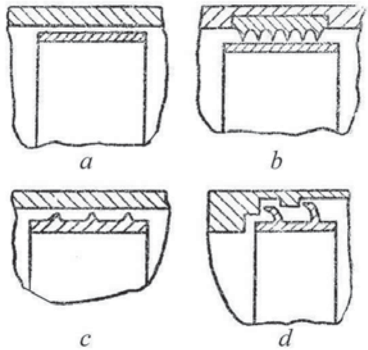


Fig. 5.6. Methods for reducing gas flow through the radial gap [15]:
a – installation of the banded shelves on the blades; *b* – installation of the banded shelves on the blades and the labyrinth in the housing; *c*, *d* – installation of the banded shelves with labyrinth combs

In many modern gas turbine engines, special **radial clearance adjustment systems** are used to reduce the peripheral radial clearance, in most of which the adjustment is carried out by changing the radial thermal expansion of the turbine housing. Fig. 5.7 shows the design of the high and low pressure turbine housings of the UGT25000 Zorya-Mashproekt engine with such a system [15].

In the UGT25000 engine, the radial clearance in the high-pressure turbine is determined by the thermal expansion of the regulating ring 1, in which the segment inserts 2 are fixed. The thermal expansion of the regulating ring in the radial direction, independent of other housing parts, is ensured by a flexible element 3, made as a single part with the regulating ring. The flexible element also ensures the centering of the regulating ring in the power housing of the high-pressure turbine 4.

To compensate for the thermal expansion of the regulating ring in the axial direction and minimize the leakage of cooling air, a seal 5 is used, formed by the annular protrusion of the regulating ring and elastic segment plates. Jet cooling of the regulating ring is organized using a flexible screen ring 6 with grooves and numerous "shower" cooling holes 8.

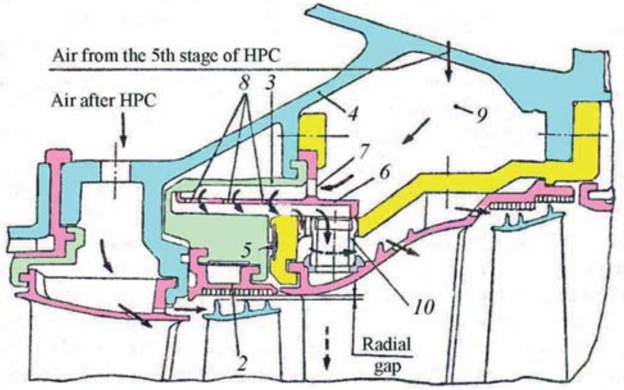


Fig. 5.7. Design of the high- and low-pressure turbine housings of the UGT25000 engine

Cooling air, which is taken from the wheel of the 5th stage of the high-pressure compressor, is supplied to the cavity of the high-pressure and low-pressure turbine housings 9 through three external pipes (not shown in the figure), on which air cut-off valves are installed. These valves are designed to stop the supply of cooling air (cut-off) and are controlled by the air pressure after the high-pressure compressor.

When the valves are open, the cooling air, which is supplied to the cavity 9 through the grooves 7 and holes 8, is supplied to cool the regulating ring. Then, through the fitting 10, this air is used to cool the nozzle vanes of the low-pressure turbine, and also through several holes – to cool its housing, the shelves of its nozzle vanes and the band shelves of the blades.

The algorithm of the control system involves starting, entering the idle mode, and rising to a mode of about 5% of the nominal with closed shut-off valves. Then the gradual opening of the valves begins, and at a mode of 10% of the nominal, the valves are fully open. When the power decreases, the valves close in the reverse order.

At the emergency or crash stop, due to the sudden drop in pressure downstream of the high-pressure compressor, the shut-off valves close almost instantly and prevent cooling of the control ring after the stop. Subsequent starts occur with the valves closed and, therefore, with a warmed ring.

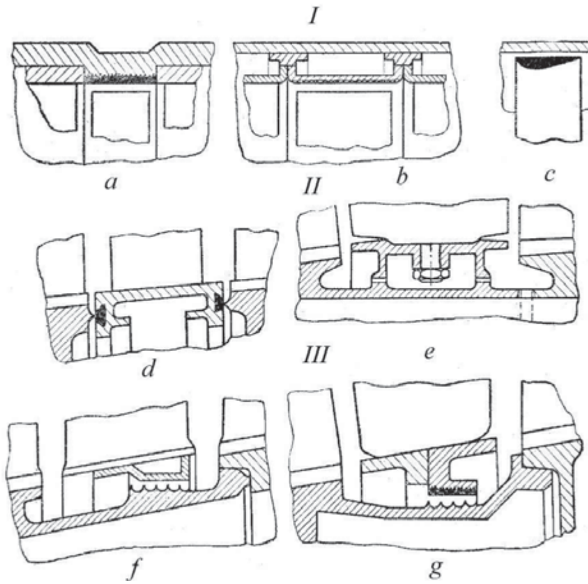


Fig. 5.8. Sealing devices in the flow section of the compressor [15]:

I – seal between the housing and the blades; II – seal between the blades and the guide vanes; III – seal between the guide devices and the rotor; *a* – coating of the belt above the blades with a layer of talc paste 2-3 mm thick; *b* – coating of the tract ring with a layer of alumino-asbographite 0.7 mm thick; *c* – thinning of the blade tip; *d* – end seal with a layer of talc paste; *e* – seal provided by small radial gaps; *f*, *g* – labyrinth seals

Figure 5.8 shows various **sealing devices in the flow section of the compressor**.

To reduce the mounting radial clearances, it is widely used to coat the stationary surfaces of the compressor (in places of possible contact) with special soft pastes based on graphite, talc, asbestos, aluminum powder, and other components. If the tips of the blades touch the coating, it is activated, ensuring self-installation of the minimum clearance.

In some compressors, in order to reduce radial clearances, the tips of the blades are thinned (Fig. 5.8, *c*), which ensures their refinement when touching the housing. In this case, a radial clearance of 0.1-0.25 mm is achieved.

5.2. Sealing of air and gas cavities

For various purposes, several air and gas cavities with different pressures are organized in the internal space of gas turbine engines. To eliminate or reduce the flow of air or gas from cavities with higher pressure to cavities with lower pressure, special sealing devices are installed at the boundaries of these cavities.

All seals of air-gas cavities of gas turbine engines can be conditionally divided into non-contact and contact. Let's consider the main ones.

In compressors and turbines of gas turbine engines, the following types of air-to-air ("gas-to-gas") seals are used between the rotor and stator:

– labyrinth seals;

– brush seals;

– gap seals;

– dry gas-dynamic seals;

– dry gas-static seals.

Let us consider each of the listed seals in turn.

Non-contact **labyrinth seals** have been widely used in all gas turbine engines. Their action is based on repeated throttling of gas flowing through channels with sharply varying cross-sections. The seal consists of sequentially arranged slots and vortex chambers. When gas passes through the slot, its pressure decreases and its speed increases (Fig. 5.9). In the chamber behind the slot, the speed decreases sharply (due to vortex formation) at a constant pressure. Thus, as a result of repeated throttling, significant aerodynamic resistance is created in the gas path, which reduces leaks.

The labyrinth seal cannot completely seal the gap, but it can reduce gas leakage through it to some acceptable values.

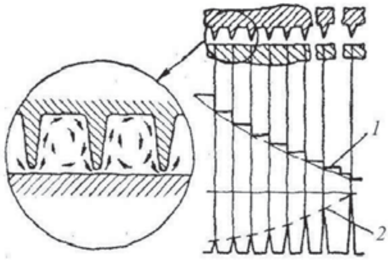


Fig. 5.9. Change in pressure (*1*) and gas velocity (*2*) in a labyrinth seal

The efficiency of the labyrinth seal depends on the number of combs, their configuration, the pressure drop between the sealed cavities, and the size of the gap. The edges of the combs must be sharp, since even a slight rounding reduces the sealing efficiency.

Figure 5.10 shows some designs of labyrinth seals [15]. Leakage through the seal is directly proportional to the flow coefficient *k*, which is determined experimentally. Other things being equal, the more sharply the direction of the gas changes in the labyrinth chamber, the more the gas is throttled and the more efficient the seal. For example, the labyrinth shown in Figure 5.10, *d* is almost twice as efficient as the most common labyrinth shown in Figure 5.10, *a*, but it is also more complex, since it requires considerable precision in manufacturing and a collapsible design of the labyrinth body.

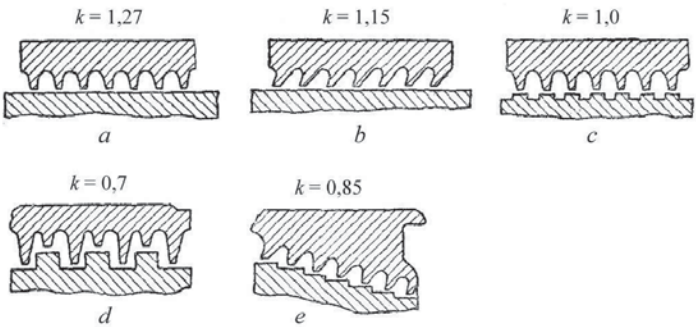


Fig. 5.10. Labyrinth seal designs

As noted in [15], minimizing leakage through a labyrinth seal can be achieved structurally by:

– increasing the size of the vortex chambers between the slots;

– tilting the labyrinth teeth against the direction of flow;

– stepped arrangement of combs in the seal.

Leakage through the labyrinth seal is also directly proportional to the annular area of the gap between the combs and the sleeve. To reduce it, it is necessary, firstly, to place the seal on the smallest possible diameter, bringing it closer to the hub of the disk, and secondly, to reduce the radial gap between the combs and the sleeve.

To implement the latter, cermet or honeycomb inserts (Fig. 5.11) can be installed in the labyrinth sleeve above the combs (Fig. 5.12), and a layer (up to 2 mm) of soft but heat-resistant material (graphite-talc or graphite-aluminum) can be applied in one way or another. As a rule, the sealing combs are located on the rotor, and the attached (cut-off) inserts or coatings are on the stator of the turbomachine.

Fig. 5.11. Honeycomb inserts of labyrinth seals

To reduce wear on the combs, strengthening coatings are applied to them - by plasma spraying or electron-spark alloying. During operation, the combs cut into the coatings and form grooves in them (see Figs. 5.12 and 5.13).

Fig. 5.12. Labyrinth combs

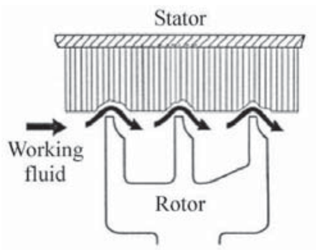


Fig. 5.13. Inserting the labyrinth combs into the honeycomb insert

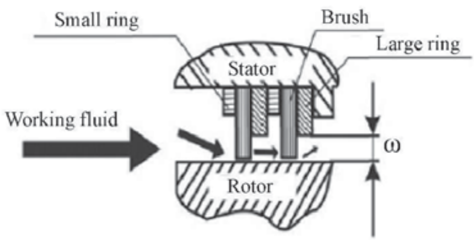


Fig. 5.14. Brush seal

Sometimes, to prevent the destruction of labyrinth combs when they touch the sleeve, an elastic suspension of this sleeve is provided.

With significant pressure drops, when a significant number of vortex chambers are required to reduce leaks, labyrinths can be made in two or three tiers to reduce the length.

Labyrinth seals are simple in design, easy to install, and have no restrictions on the temperature and pressure of the sealed medium, as well as on the relative sliding speed. When they are brought in, sometimes strength problems arise, including those of vibration origin, but they are constructively overcome. The disadvantages of labyrinth seals include the largest (compared to other types of seals) leaks.

To reduce air (gas) flow between cavities in gas turbine engines, contact **brush seals** are used.

Such seals have been introduced into the design of gas turbine engines since the early 1990s. They are bundles of wire made of a very hard alloy (for example, based on cobalt), which continuously touch (at an angle of approximately 45o) the rotating part of the seal. The diameter of the bristles is 0.025-0.1 mm, and up to 300 bristles are located per 1 mm of length in the circumferential direction [15]. The scheme of a double-row brush seal is shown in Fig. 5.14.

Each row consists of brushes clamped and secured between small and large flat rings (plates). There is a gap ω between the large plates and the rotor, as in a conventional labyrinth seal, and in the event of a brush breakage, the seal becomes a conventional labyrinth seal.

Fig. 5.15 shows one row of brush seals in a free (without rotor) state. The diameter of the rotor 4 is conventionally shown by a dotted line. When assembling the brush, overlap 5 is selected, and the brushes provide contact with the rotor using their elasticity. When the radial and axial clearances change during operation, the brushes easily adapt to these changes, maintaining continuous contact with the rotor.

The free length of the bristles 9, the protrusion of the bristles 6, the diameter of the bristles, and their density are the main characteristics of the brush seal:

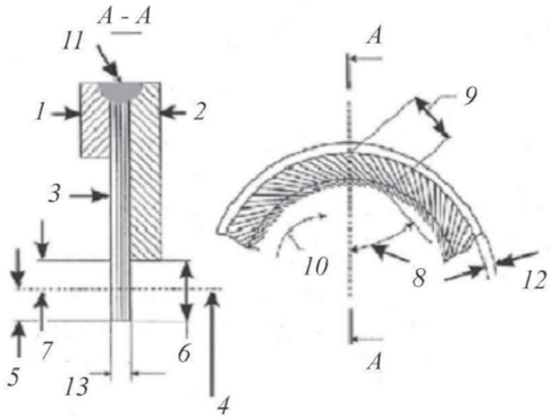


Fig. 5.15. Scheme of a brush seal:

1 – front plate; 2 – rear plate; 3 – wire bristle package; 4 – rotor diameter; 5 – radial overlap of bristles; 6 – radial protrusion of bristles; 7 – gap on the rear plate; 8 – angle of bristles in the free state; 9 – free length of bristles; 10 – direction of rotor rotation; 11 – weld; 12 – radial height of front plate 1; 13 – thickness of bristle package

– the larger the diameter of the bristles, the higher the pressure drop the seal can withstand, but the more leakage occurs;

– doubling the bristle density reduces leakage by approximately 30%, while the sustained pressure drop more than doubles;

– when the free length of the bristles is reduced, the pressure drop that can be maintained increases and the leakage decreases. However, this leads to an increase in the stiffness of the brushes, an increase in heat generation due to friction, and a decrease in the service life of the brushes.

The flexibility of the bristle bundle (Figs. 5.16 and 5.17) is the main advantage of the brush seal and allows it to provide a minimum gap and minimal leakage in all operating modes. Labyrinth seals, for example, receive maximum cutting in transient operating modes; therefore, in the nominal (and most important for gas turbine engines) mode, these seals operate with increased gap and leakage.)

As noted in [15], the efficiency of brush seals is 20-40% higher than that of labyrinth seals.

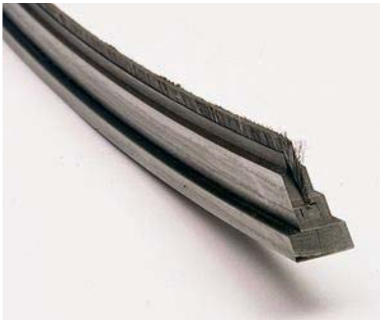



Fig. 5.16. Brush seal elements

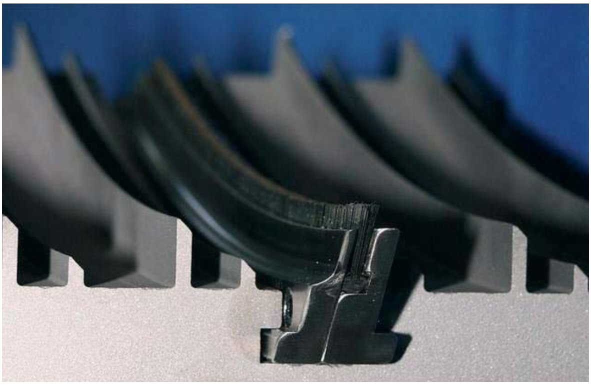


Fig. 5.17. Brush seal

The effectiveness of brush seals depends on the wear of the bristles, which in turn is determined by the circular speed and the quality of the coating at the contact points. An optimal coating (plasma-applied chromium carbide) allows you to reduce wear by half.

Brush seals have proven themselves well in stationary gas turbine engines from GE Vernova. In aircraft engines, the operating conditions in terms of multimode and cyclic loads are much more severe, but there is evidence of the successful use of such seals in this area as well.

The main problem with brush seals is their low service life in harsh operating conditions (high rotation speeds, temperatures, and pressure drops) due to wear, burning, and breakage of the bristles.

Non-contact **gap seals** are based on the creation of a very narrow gap between the sealing surfaces (see, for example, Fig. 5.8, *e*). In this case, complete isolation of the separated cavities is not ensured, and the flow through the gap occurs more, the higher the pressure difference between the separated cavities increases and the annular area of the gap.

Gap seals are typically used to separate cavities with a small pressure difference, for example, in the guide vanes of low-pressure stages of compressors.

Dry gas-dynamic seals operate on the principle of a gas-lubricated thrust bearing. In aircraft engines, such seals are currently used only for sealing compressor bearing supports at air temperatures up to 700 K, rotational speeds of 100-150 m/s, and diameters of 100-200 mm [15].

Since 1976, dry gas-dynamic seals from the English company "John Crane Inc." have become widely used in compressor bearings of industrial gas turbine engines.

A dry gas seal can be single, double (tandem), triple, etc. The seal from John Crane Inc. contains the following main parts (Fig. 5.18-5.21):

a) a heel (ring) 5 made of tungsten carbide (titanium carbide or silicon carbide), which is fixed to the shaft by means of a sleeve 6 and pins 7 and rotates with it. At the end of the contact surface of the heel, lifting platforms (microgrooves) with a depth of 5-20 microns are made (for gas) by ion bombardment or chemical etching. The shape and dimensions of the platforms are different and can vary depending on the design and purpose of the seal (Fig. 5.22). The company "John Crane Inc." prefers to use spiral microgrooves in its products;

b) stator ring 2, made of graphite impregnated with resin;

c) springs 3 made of nickel alloy, which press the stator ring 2 against the rotating ring 5;

d) rings 4, 9, and 12 made of fluorocarbon elastomer, which seal the gaps between the parts.

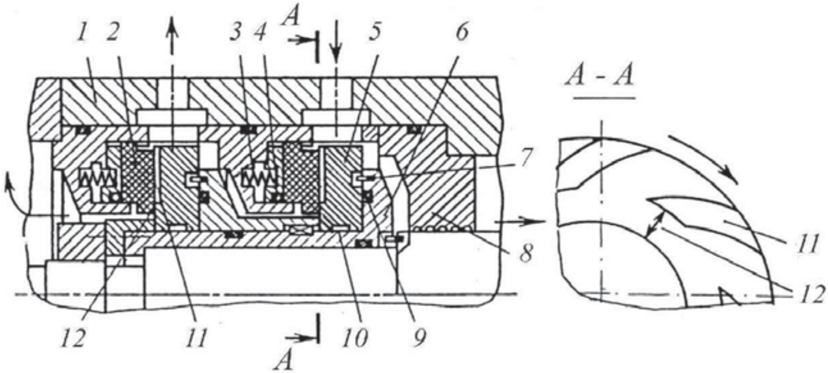


Fig. 5.18. Two-stage dry gas seal from John Crane Inc.:

1 – housing, 2 – graphite ring; 3 – springs; 4 – sealing ring; 5 – rotating heel (ring) made of solid material; 6 – sleeve; 7 – pins; 8 – labyrinth seal; 9 – sealing ring; 10 – bracelet spring; 11 – spiral grooves; 12 – sealing ring

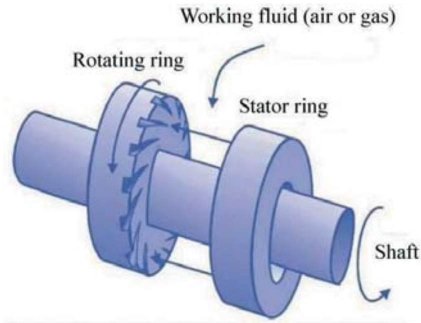
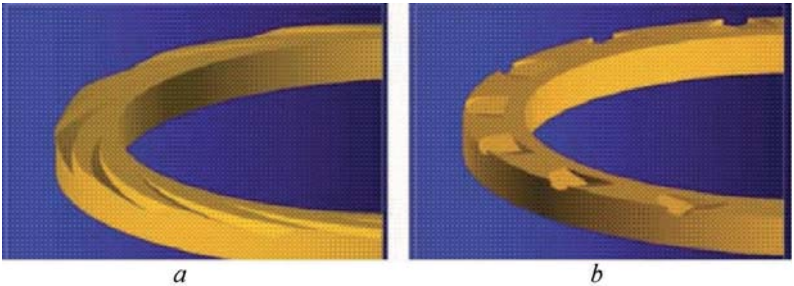


Fig. 5.19. On the principle of operation of a gas-dynamic dry seal

When stationary, the stator ring 2 is slightly pressed by springs 3 to the heel 5. During rotation, due to the resulting viscous gas-dynamic pressure between the sealing surfaces of the heel 5 and the ring 2, which is enhanced by the presence of microgrooves, the ring 2 floats above the heel 5 by several microns, and the seal operates on gas lubrication without wear (with a gap of approximately 0.005 mm).

Fig. 5.20. Two-stage seal from John Crane Inc.

Fig. 5.21. Elements of a dry gas seal



a *b*

Fig. 5.22. Rings with spiral (*a*) and U-shaped (*b*) gas-dynamic grooves

If the gas contains particles larger than 5 microns, the shut-off compressor gas entering the seal is pre-cleaned in a filter.

The seal can operate with axial movements of the shaft relative to the housing up to 2.5 mm and radial movements up to 0.6 mm.

In **dry gas-static seals**, increased pressure in the layer between the contact surfaces is created not by the rotation of the rotor, but by supplying air to it under excess pressure from a special blower.

Gas-static seals require less manufacturing precision than gas-dynamic seals, and they better withstand constant and pulsating loads, since their clearance is slightly larger than that of gas-dynamic seals.

The main drawback that limits their application is the need for an additional air blower that develops a pressure exceeding the pressure of the sealed medium.

The company "Stein Seal Co." specially for GE Vernova developed and conducted research on the design of a gas-static dry seal, which is called **an aspirator seal**.

5.3. Sealing of oil cavities

Reliable sealing of oil cavities significantly affects the reduction of oil consumption and the reliability of engine operation as a whole. The breakthrough of oil or its vapor into the flow part of the engine not only increases oil consumption, but also worsens the operating conditions of the combustion chambers, contributes to the deposition of carbon deposits on the parts of the flow part, pollutes the engine, and leads to other undesirable consequences. On the other hand, the ingress of hot gas or air into the oil cavities leads to additional heating of the oil and increases the mass of air (gas) that is removed during blowing.

The following types of oil cavity seals are used in gas turbine engines:

– labyrinth seals;

– ring seals;

– graphite contact seals;

– mechanical non-contact seals;

– threaded seals;

– lip seals;

– seals with oil-reflecting rings.

In gas turbine engines, non-contact **labyrinth seals** are used not only to seal air-gas cavities, but also to prevent oil leakage from oil and blow-by cavities into the flow part of the engine. Labyrinth seals are also installed near the bearing devices of turbines and compressors to prevent hot gases from entering them.

In the latter case, if there is a two-tier labyrinth specially organized for this purpose, the cavity between the labyrinths can be used to supply air from the compressor to support the oil cavity or to connect it with the atmosphere to remove hot gases that have penetrated through the first labyrinth (Figs. 5.23 and 5.24). This prevents bearing overheating, oil coking, and other malfunctions.

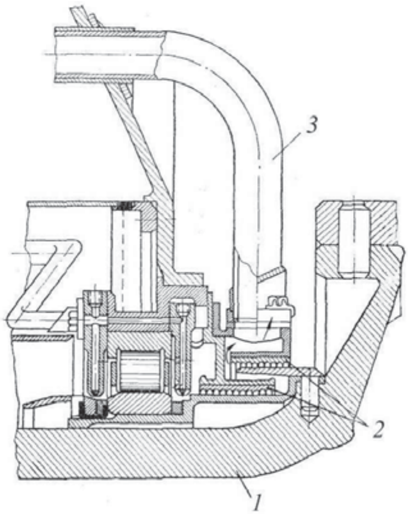


Fig. 5.23. Removal of gases that have penetrated through the first labyrinth of the turbine bearing [15]:

1 – turbine shaft; 2 – seal; 3 – gas outlet tube

Contact **ring seals** have become widely used in gas turbine engines. The main elements of such seals (Fig. 5.25) are separate elastic rings made of forged bronze with a high content of tin, as well as cast iron or steel. The rings are pressed against the cylindrical surface of the sleeve by elastic forces, and against the end surface of the groove on the rotor by excess pressure in the sealing cavity.

Fig. 5.24. Elements of two-tier labyrinth oil seals

The working surfaces of the rings are made with great precision, chrome-plated to reduce wear, and sometimes, for better finishing, they are coated with a layer of lead or cadmium 3-5 microns thick.

Grooves for sealing rings are most often made on a special sleeve (ring retainer), which is then installed on the shaft with tension, which simplifies heat treatment, facilitates the elimination of sealing defects, etc. To increase wear resistance, the end surfaces of the grooves are nitrided or cemented.

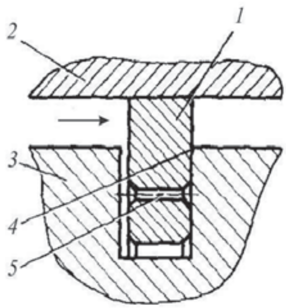


Fig. 5.25. Scheme of the ring seal:

1 – elastic ring; 2 – housing, 3 – sleeve; 4 – end surface of the groove; 5 – oil supply holes

The elasticity of the rings is selected so that when the shaft rotates they remain stationary or rotate slightly. In the latter case, the rings have a small wear on the end face. With significant rotation of the rings (due to insufficient elasticity), wear occurs on the cylindrical surface, which worsens the operation of the seal and makes it difficult to dismantle.

At high specific pressure, dry friction results in high heat generation, which reduces the efficiency of the seal. In such cases, a series of small diameter holes (approximately 1 mm) are made in the rings to lubricate the friction surfaces (see Fig. 5.25).

For higher reliability of the seal, not one, but two or three rings are usually installed (Fig. 5.26).

Despite the difficulty of manufacturing rings, especially large diameter ones, the difficulty of assembly and disassembly, and the danger of oil coking at high temperatures, ring seals are widely used in gas turbine engines, especially twin-shaft ones. They operate satisfactorily up to rotational speeds of 60-80 m/s and even more.

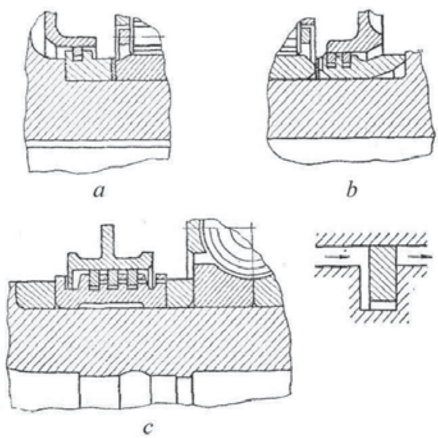


Fig. 5.26. Ring seals [15]:
a – with one ring; *b* – with two rings; *c* – with three rings

In modern gas turbine engines, **graphite contact seals** of various designs are used. These seals are characterized by high reliability and moderate wear at fairly high loads. The principle of operation of these seals is largely similar to the operation of the previously considered ring seals.

As a material, graphite is characterized by low cost, good lubricating properties (self-lubricating effect during friction), high corrosion resistance, low coefficient of thermal expansion, and good thermal conductivity. Its disadvantages include high fragility (including rapid failure under shock and vibration loads), low wear resistance, and low allowable specific pressure.

Sealing in such designs can be carried out from the radial side, from the end side, and simultaneously from both of these sides.

Fig. 5.27, *a* shows an example of the design of a radial sectional graphite contact seal. Graphite segments 3 are crimped onto the surface of the shaft 1 by means of a spiral spring 4, and are pressed against the end of the fixed housing 2 by springs 7. The graphite segments are held against rotation by stoppers 8.

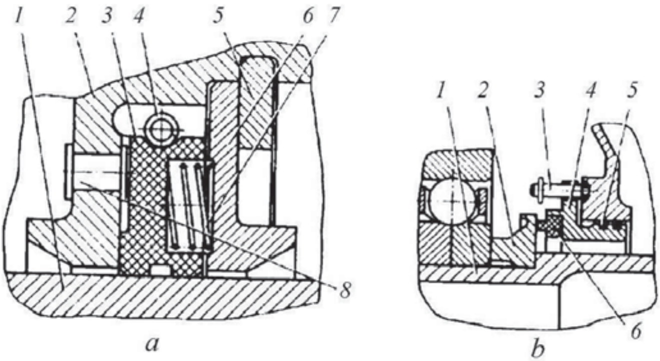


Fig. 5.27. Graphite contact seals [15]:

a – radial sectional seal: 1 – rotor shaft; 2 – seal housing; 3 – graphite segments; 4 – spiral spring; 5 – steel washer; 6 – release ring; 7 – axial spring; 8 – stopper;

b – end contact seal: 1 – rotor shaft; 2 – sealing sleeve, 3 – guide pins; 4 – clamping sleeve; 5 – sealing lips; 6 – graphite ring

The geometric dimensions, shape, and number of sealing elements vary widely. Thus, the sealing can be made single-row (as shown in the figure) or double-row with overlap of the joints between the rows by arranging them in a staggered order. The number of segments in one row is recommended to be between four and six. It is recommended to take no more than 0.25 MPa for the pressure drop acting on the seal.

Fig. 5.27, *b* shows the end contact seal. The sealing ring 6, glued to the steel sleeve 4, moves under the action of axial springs along the guide pins 3, creating contact with the end of the rotating steel sleeve 2 on the rotor shaft 1. The specific pressure in the contact must exceed the air pressure drop during operation of the gas turbine engine. The sleeve 4 is sealed in the support cover by rubber lips 5.

For reliable operation of such a seal, high precision of manufacturing of parts and absence of skew of the ends in contact are necessary; sleeve 2 must be cooled by a jet of oil (from 2.5 to 4.5 l/min). Relative axial movements in the seal are limited. Significant air pressure drops of up to 0.4 MPa are allowed (at a temperature of 473 K and a circular speed of up to 75 m/s).

Fig. 5.28 shows a combined radial and end contact seal, which consists of a split graphite ring 4, a steel sleeve and a heel 3. During operation, the ring 4 is pressed by air pressure with its left end against the rotating heel 3, and along the cylindrical surface, it is pressed against the steel sleeve by elastic forces. The heel 3 is cooled by oil, which is located in the oil cavity.

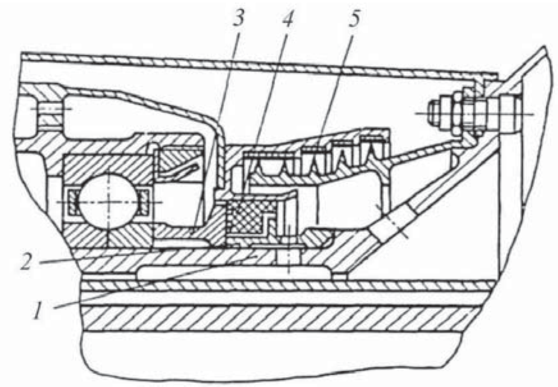


Fig. 5.28. Compressor support seal [15]:
1 – rotor shaft; 2 – steel ring, 3 – steel heel ring; 4 – graphite sealing ring; 5 – air labyrinth seal

The various elements of graphite contact seals are shown in Figs. 5.29 and 5.30.

Fig. 5.29. Elements of a graphite contact seal

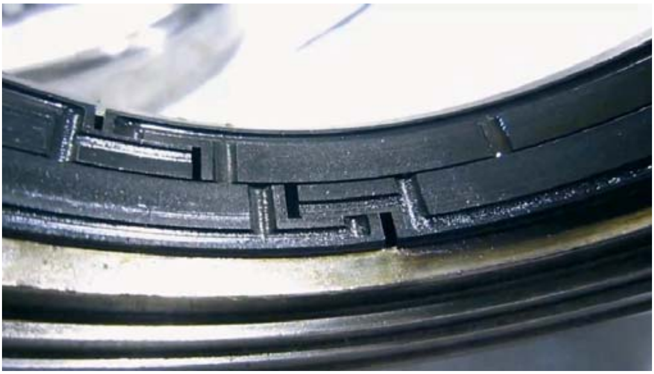


Fig. 5.30. Double-row graphite contact seal (section view of the rings)

The principle of operation of **mechanical non-contact seals** is similar to the principle of operation of dry gas-dynamic seals, discussed earlier in this book.

This seal (Fig. 5.31, *a*) consists of the same parts as the graphite contact seal (see Fig. 5.27); however, on the end surface of the heel, there are several dozen lifting areas with a depth of 10-15 microns, called **Rayleigh pockets**, the shape of which is shown in Fig. 5.31, *b*.

Due to the lifting of the Rayleigh pockets between the ends of the rotating heel 6 and the fixed ring 2 during rotation, an increased viscous gas-dynamic pressure ("air cushion") arises, as a result of which the ring 2 moves away from the heel 6 and the seal switches to the air (gas) lubrication mode with a gas film thickness of several microns. The result is almost complete elimination of wear of the end surfaces of the seal and thereby an increase in its service life. Secondary sealing is provided by a split graphite ring 3.

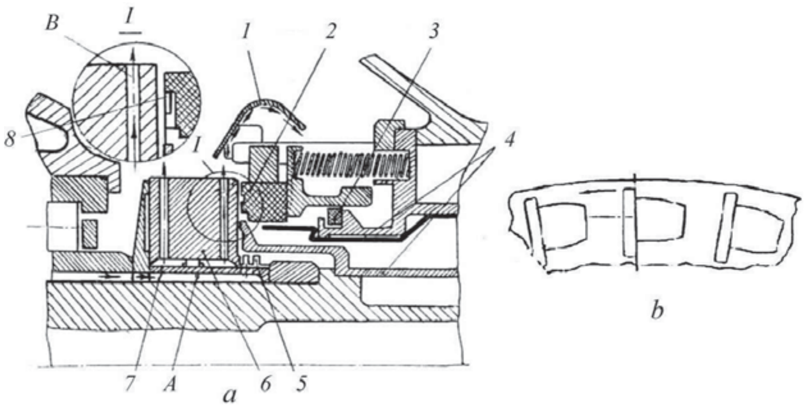


Fig. 5.31. Mechanical non-contact seal of the compressor support (*a*) and its lifting platforms (Rayleigh pockets) (*b*) [15]:

1 – deflector; 2 – graphite stator ring; 3 – split graphite ring; 4 – heat-insulating screens; 5 – elastic elements; 6 – rotating ring (heel); 7 – sleeve; 8 – Rayleigh pockets; A – sleeve groove; B – oil pumping channels

To reduce the effect of high air temperature on the deformation of parts in the seal, heat-insulating screens 4 can be used. For the same purpose, the rotating heel 6 is made of an alloy containing 97% molybdenum, 1.5% titanium, and 1.5% zirconium. This combination of materials has high thermal conductivity and a low coefficient of thermal expansion.

Studies conducted for a shaft with a diameter of 168 mm [15] show that a Rayleigh pocket seal provides a gap seal 2.5 times better than a graphite contact seal and 6.5 times better than a conventional five-comb labyrinth seal.

The mastered parameters for mechanical non-contact seals are: temperature up to 600-700 K, pressure drop 0.3-0.5 MPa, circular speed up to 100 m/s.

The materials used for sealing rings are carbon graphite (for gas temperatures up to 573 K), pressed graphite and pyrolytic graphite (pyrographite) (for temperatures 623-673 K) [15]. Zorya-Mashproekt has been using mechanical non-contact seals in its engines since the early 1990s. As noted in [15], replacing labyrinth seals with mechanical non-contact seals allows reducing the inflow of air into the oil cavities by 10-15 times. Due to this, oil consumption is significantly reduced due to the reduction of its irreversible losses in the blowing system. A significant disadvantage of such seals is that to ensure a resource of about 25,000 hours, which corresponds to the engine resource before overhaul, in the range of operating modes the pressure drop across the seal should not exceed 0.10-0.13 MPa.

Examples of the use of mechanical non-contact seals by Zorya-Mashproekt are shown in Figs. 5.32 and 5.33.

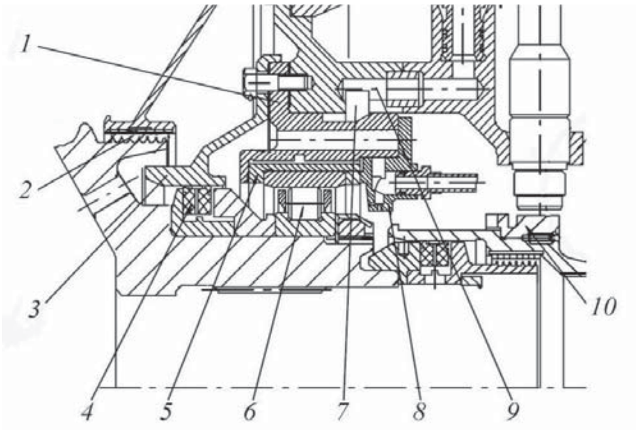


Fig. 5.32. Rear support of the UGT25000 low-pressure compressor [20]:

1 – housing of the rear support of the low-pressure compressor; 2 – labyrinth cover; 3 – cover; 4 – sealing graphite ring; 5 – sleeve; 6 – roller bearing; 7 – annular oil distribution channel; 8 – oil supply ring; 9 – oil channel; 10 – gear-inductor

Non-contact **threaded seals** (Fig. 5.34) have the appearance of a ribbon spiral (thread) cut on the shaft or in the housing. Accordingly, the housing or shaft is made smooth and cylindrical, and the radial gap in the seal is small (0.2-0.4 mm).

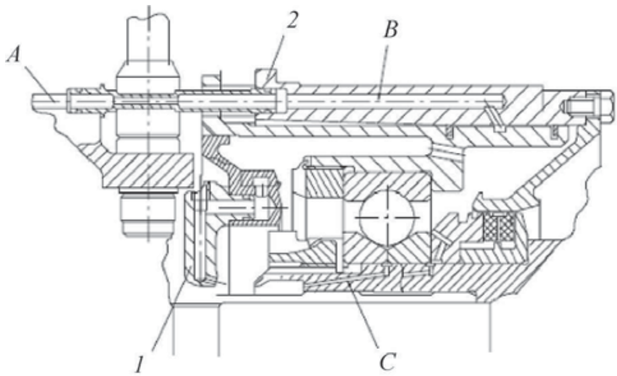


Fig. 5.33. Front support of the UGT25000 high-pressure compressor [20]:
A – oil channel; B, C – oil supply channels; 1 – oil nozzle; 2 – restrictor cover

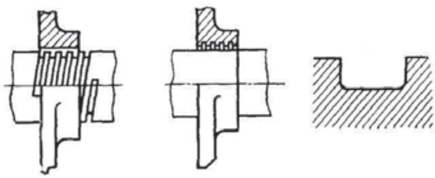


Fig. 5.34. Threaded seal

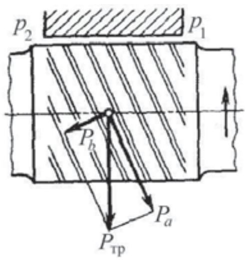


Fig. 5.35. Non-contact threaded seal

The principle of operation of the seal is that any particle of liquid (oil) will be subjected to a friction force P_{tr}, which arises between the sealed liquid and the housing (Fig. 5.35). The direction of the thread and the rotation of the shaft must be coordinated so that the component P_a of the friction force acting along the groove is directed towards the sealed oil cavity. The pressure p_2 in the air cavity must be 0.05-0.07 MPa higher than the pressure p_1, so that a certain support is created, but air leaks through the gap are minimal.

It is recommended to perform spiral cutting on the shaft or in the housing with multiple cuts (up to three or four cuts) of a rectangular, and sometimes triangular, profile.

The threaded seal can be used in combination with a labyrinth seal, which provides the necessary pressure difference across the threaded seal.

Contact **lip seals** operate satisfactorily at rotational speeds not exceeding 25-30 m/s. Lip seals are widely used in drives of auxiliary units, as well as for sealing the ends of the output shaft of the gearbox and parts that perform reciprocating movements.

The lips are made of special grades of rubber, leather, and elastic plastics. The lip is installed on the shaft with pre-tension and, in most cases, is additionally pressed against the shaft using a spiral spring.

The specific pressure at the contact surface must be equal to or higher than the pressure difference on both sides of the seal. This condition provides a good seal, but the heat generated by friction during operation can cause the seal to fail.

Limited rotational speed, significant friction, and short service life limit the use of lip seals in gas turbine engines.

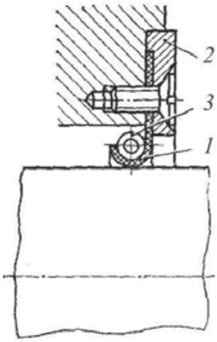


Fig. 5.36. Lip seal:
1 – lip; 2 – plate; 3 – spiral spring

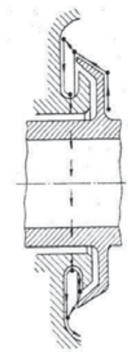


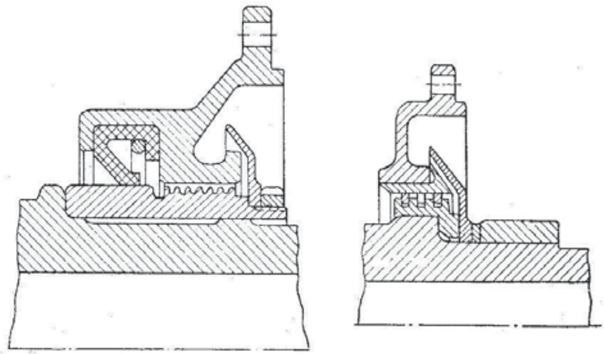
Fig. 5.37. Principle of operation of the oil-reflecting ring

Fig. 5.36 shows one of the possible variants of the lip seal [15]. A leather or rubber lip 1 is attached to the housing with a plate 2 and screws. A spiral spring 3, welded into a ring, is put on the flange of the lip, which compresses the rotating shaft. The leather from which the lip is made must be specially treated, and the rubber must be oil-resistant. The elasticity of the spring is selected experimentally to satisfy two contradictory requirements: the spring force must be large enough to prevent oil from passing through, and at the same time it must be small enough so that the heat generated by friction does not damage the lip.

The principle of using centrifugal forces to retain oil is widely used in **seals with oil-reflecting rings** (Fig. 5.37).

Oil that has hit the reflecting ring, under the action of centrifugal and viscous forces, moves along the surface of the ring to its edge, is thrown onto the housing, and flows to the drain point. To prevent oil from getting into the gap, it is advisable to make an annular protrusion on the cylindrical part of the housing.

Such seals do not provide tightness and are used mainly as auxiliary ones – to reduce the amount of oil that enters other types of seals. Figs 5.38, 5.39, and 5.40 show examples of combined seals using oil-reflecting rings.



a *b*

Fig. 5.38. Combined seals [15]:

a – with a lip, thread and oil-reflecting ring; *b* – with contact rings and oil-reflecting ring

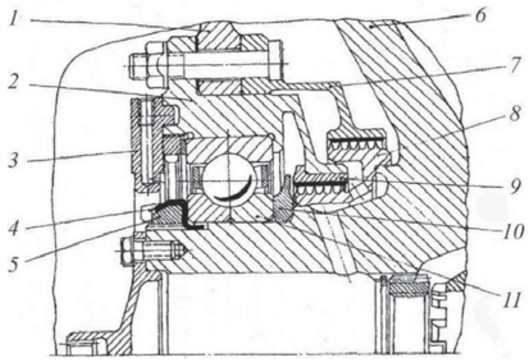


Fig. 5.39 Front support of the D050 low-pressure compressor [15]:

1 – inner wall of the front housing; 2 – front support housing; 3 – oil collector, 4 – lock; 5 – clamping nut; 6 – front trunnion; 7 – cover, 8 – labyrinth sleeve; 9 – sleeve; 10 – oil-reflecting ring; 11 – bearing

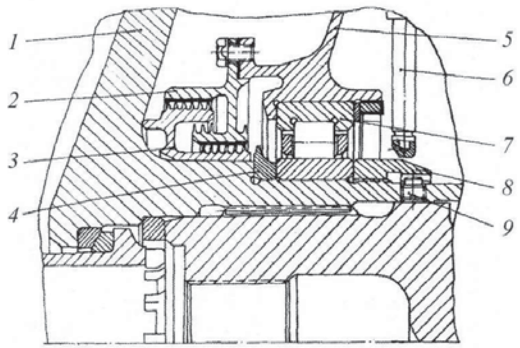


Fig. 5.40 Rear support of the D050 low-pressure compressor [15]:

1 – rear trunnion; 2 – rear support cover; 3 – labyrinth sleeve; 4 – oil-reflecting ring, 5 – rear support wall; 6 – oil supply manifold; 7 – roller bearing; 8 – clamping nut; 9 – pin

Control tasks and questions

1. Explain the change in radial clearances between the housing and the tips of the turbine blades at different operating modes.

2. Explain the need to seal mounting radial and axial clearances in gas turbine engines.

3. How are peripheral radial clearances sealed in gas turbine engines?

4. Describe the main types of seals for air-gas cavities of gas turbine engines.

5. Explain the structure and operation of a labyrinth seal.

6. Explain the structure and operation of a brush seal.

7. Explain the structure and operation of dry gas-dynamic seals.

8. Describe the main types of oil cavity seals.

6. GAS TURBINE ENGINE HOUSINGS

6.1. Inlet devices

The inlet device is designed for the smooth and uniform supply of atmospheric air to the compressor. Circular unevenness of the velocity field at the inlet can lead to vibration of the first-stage blades and their possible failure.

The most favorable is the use of an **axial inlet to the compressor.** In this case, the circular symmetry of the inlet can be broken only by the radial struts that connect the outer housing with the bearing housing. However, with a sufficient distance of the struts from the first stage blades and with their appropriate profile, the influence of the struts on the velocity field can be minimized.

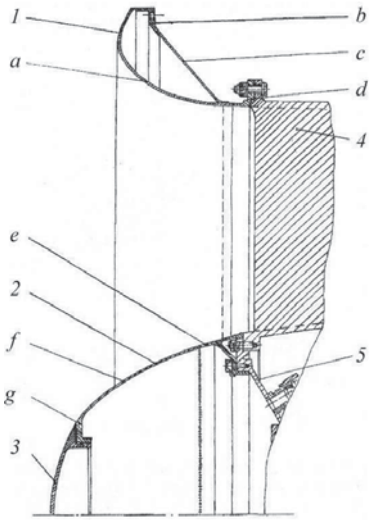


Fig. 6.1. Inlet device of the D050 engine [15]:

a, c – walls; b, d – flange; e, g – flanges; f – wall; 1 – outer fairing; 2 – inner fairing; 3 – fairing cover; 4 – front housing; 5 – wall

Fig. 6.1 shows the axial-type inlet device of the D050 marine gas turbine engine from the Zorya-Mashproekt. It consists of outer and inner fairings. The annular channel between them serves as the beginning of the flow path of the engine.

The outer fairing 1 is made welded and consists of flanges "b", "d" and walls "a", "c". Using flange "d", the fairing is attached to the front housing. Flange "b" has holes for connection to the inlet duct.

The inner fairing is also welded, consisting of a flange "g", a wall "f" and a flange "e". It is attached to the front housing with the help of a flange "e". The flange "g" has a hole for mounting the fairing, which is closed by a fairing cover 3.

Figures 6.2 and 6.3 show gas turbine engines with axial-type inlet devices.

Fig. 6.2. Zorya-Mashproekt UGT16000 engine with axial-type inlet device [12]

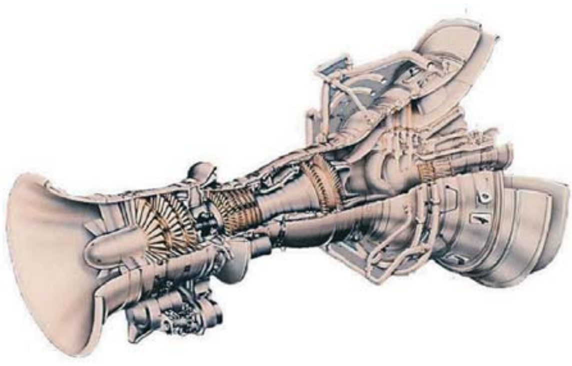


Fig. 6.3. GE Aerospace LM1600 engine with axial-type inlet device [13]

Axial-type inlet devices are widely used in aircraft gas turbine engines and engines converted from aircraft engines.

The axial inlet to the compressor involves air suction most often from a separate compartment, separated from the engine room by a partition. An example of such a design is shown in Fig. 6.4.

Despite the obvious advantages of axial air supply to the compressor, its use in stationary and transport gas turbine engines is limited, since with axial supply it is necessary to provide a separate room (compartment) for air intake, the possibility of locating the drive mechanism on the compressor side is excluded, access to the front support of the compressor and its disassembly are complicated, since inlet devices of this type are often made without horizontal connectors.

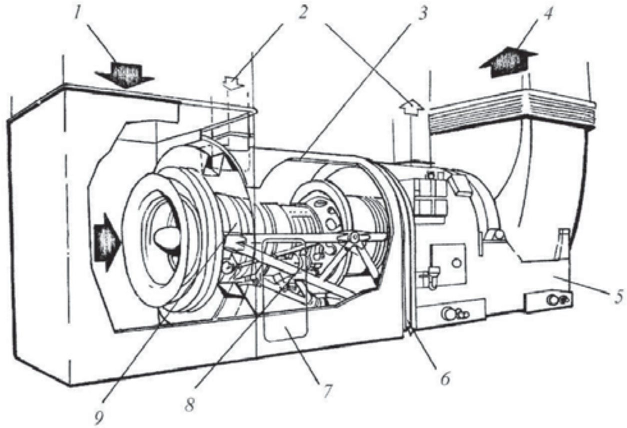


Fig. 6.4. Placement of the Rolls-Royce Olympus TM3B engine in a heat and sound insulating container:

1 – working air inlet; 2 – air for cooling the gas turbine engine; 3 – heat and sound insulating container of the gas generator; 4 – exhaust gases; 5 – power turbine frame; 6 – flexible spacer between the containers of the power turbine and gas generator; 7 – inspection hatch; 8 – gas generator frame, cantilevered with the power turbine frame; 9 – gas turbine engine

The ring-type air supply to the compressor has become somewhat widespread in gas turbine engines. The air flow enters the ring-type inlet device in a radial direction from the periphery to the center, and then inside it turns in an axial direction to the compressor inlet (Figs. 6.5 and 6.6).

Such a design usually involves air suction directly from the engine room (or from a special compartment), so it can be used either for low-power engines or for stationary and transportable high-power engines placed in separate boxes, access to which is prohibited during gas turbine engine operation.

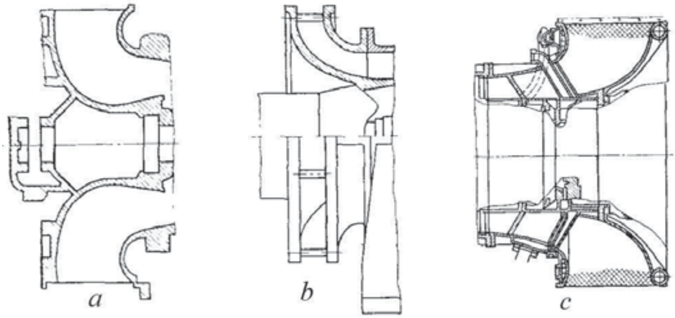


Fig. 6.5. Ring-type inlet devices [15]

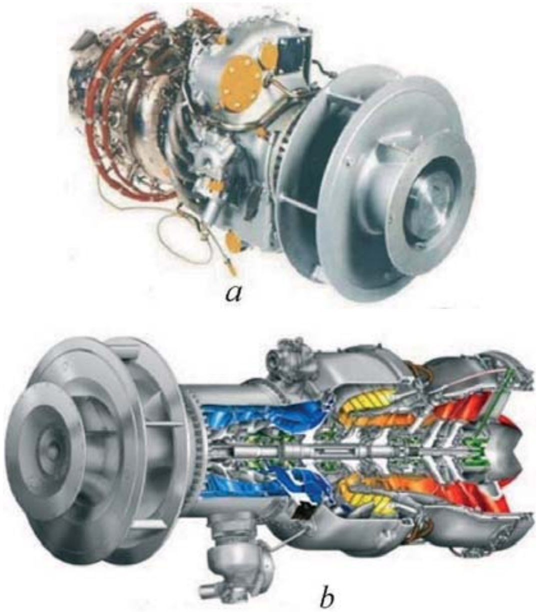


Fig. 6.6. Engines with ring-type inlet devices:
a – ST18A; *b* – ST40 (both from Pratt & Whitney Canada Corp.)

The advantage of ring-type inlet devices is the freedom of arrangement of auxiliary mechanisms and access to the front support of the compressor. The elements that connect the support body to the outer housing of the compressor can be made more rigid and durable than the hollow radial struts in an axial-inlet compressor. In addition, these elements can be carried to a much greater distance from the compressor blades.

Inlet devices with one-sided radial supply have been widely used in gas turbine engines. Air to such devices is usually supplied through air ducts of rectangular or circular cross-section from an air intake or a separate room, where it is filtered, and the suction noise is suppressed. However, it is also possible to supply it directly from the engine room.

In radial-type inlet devices, two zones can be distinguished: in the first, the flow is turned and evenly distributed to the annular cross section in front of the confuser; in the second zone, the flow is accelerated in front of the compressor blades in the confuser.

The radial-type inlet devices are divided into knee-shaped and snail-type ones.

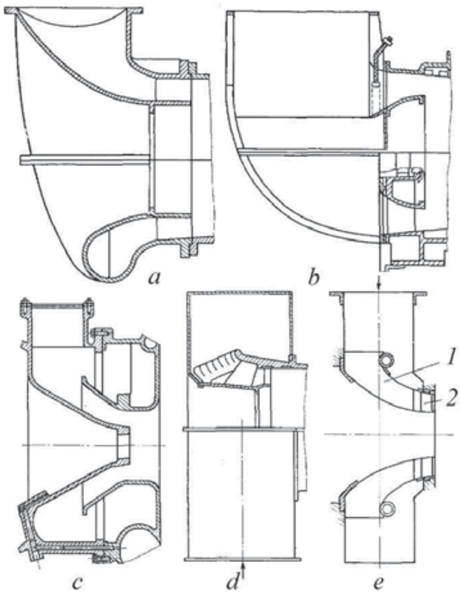


Fig. 6.7. Radial-type inlet devices [15]:
1 – radial ribs; 2 – radial blades

The knee-shaped inlet devices shown in Fig. 6.7, *a*, *b* are distinguished by the fact that both of their zones have a smooth transition. Therefore, aerodynamic pressure losses in them will be minimal, and the uniformity of air supply around the circumference will be maximum for radial-type devices.

However, such inlet devices are characterized by relatively large axial dimensions, which complicates the fastening of the front support of the compressor and access to it. They are mainly made cast to create the necessary smooth configuration. The connection of both circuits of the inlet devices is carried out using radial ribs. In the designs of some devices, only the confusing part is cast, and everything else is made of thin-walled sheet steel. Thus, the inlet device of a ship's gas turbine engine shown in Fig. 6.7, *b* is welded from thin sheet steel to reduce mass. Since the rigidity of the device is insufficient to perceive the load from the rotor, a hole is made in its lower part, inside which the front support and the drive of auxiliary mechanisms are placed.

An example of an engine design with a knee-shaped inlet device is shown in Fig. 6.8.

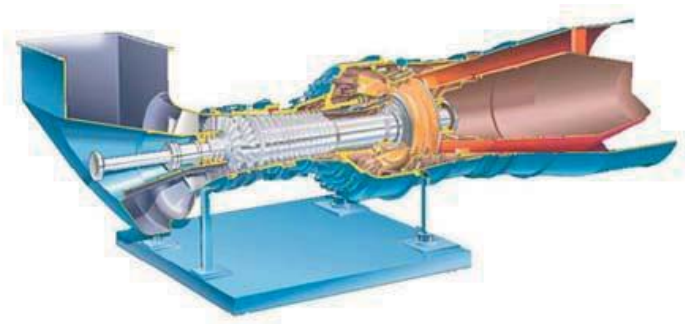


Fig. 6.8. Siemens Energy SGT-800 engine with knee-shaped inlet device

The complexity of attaching the front support and accessing it led to the fact that, despite the most favorable aerodynamic shapes, these inlet devices were not widely used.

The predominant distribution in all branches of stationary and transport gas turbine engineering is **snail-type inlet devices**, which partially cover the compressor housing (Fig. 6.7, *c*, *d*, *e*). These devices are characterized by the fact that, due to the significant free space above the compressor housing, a uniform supply of air to the annular cross-section of the confuser is achieved with relatively small pressure losses. As a result, the length of the inlet device is reduced, and access to the front support is freed.

Since snail-type inlet devices are used mainly because of their small axial size, the confuser is in most cases made curved with a minimum turning radius. Some designs include annular guide ribs (see Fig. 6.7, *d*).

Figures 6.9, 6.10, and 6.11 show gas turbine engines with snail-type inlet devices.

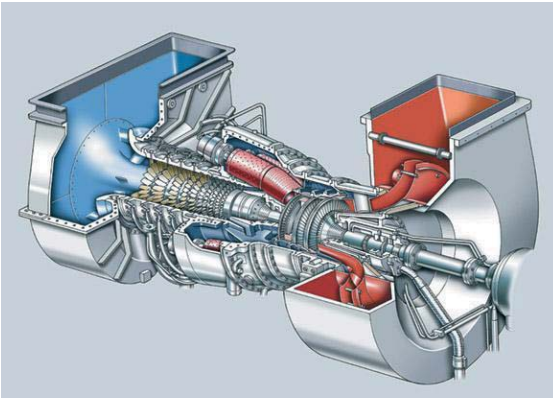


Fig. 6.9. Siemens Energy SGT-100-2S engine with a snail-type inlet device

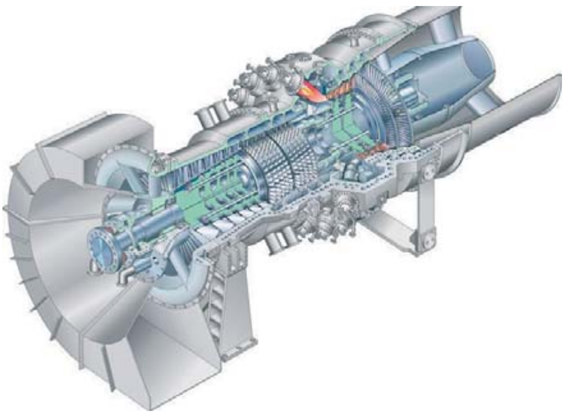


Fig. 6.10. Siemens Energy SGT6-6000G engine with a snail-type inlet device

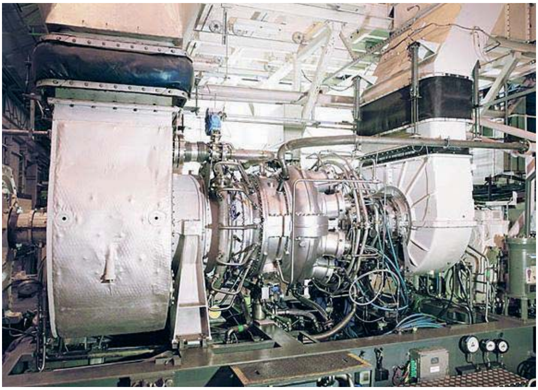


Fig. 6.11. Siemens Energy SGT-200 engine with a snail-type inlet device

6.2. Axial compressor housings

The compressor housing is one of the main parts of the power circuit of a gas turbine engine. It is usually a stationary thin-walled or thick-walled shell with ribs, flanges, and pipes. Thin-walled housings are typical for compressors of aircraft and some transport engines; thick-walled ones are mainly for stationary gas turbine engines.

Inside the housing, the rotor is mounted on support bearings, and the guide devices are attached. Outside, the drive boxes, engine mounting units to the frame, and other power structural elements are installed on the housing. The compressor housing is affected by the forces of gravity and inertia of the rotor, the excess pressure of the working fluid on the internal surfaces, and the torques transmitted from the guide devices.

The following **basic requirements** are imposed on compressor housings:

– sufficient strength and rigidity at low weight (to withstand the internal pressure of the working fluid and ensure fixed radial and axial clearances in the flow path);

– ease of installation of the rotor and guide devices;

– ease of manufacture;

– tightness.

The compressor housing most often consists of three parts: front, middle, and rear housings. The front housing is sometimes combined with the inlet device. In two-spool compressors, an intermediate housing (adapter) may be installed between the spools.

The design options for compressor housings are very diverse, since the level of stresses in them is insignificant and a wide variety of technological methods are acceptable for their implementation.

All parts of the compressor housing can be made as a single unit or separately, with a connection via vertical transverse connectors.

Fig. 6.12 shows a compressor housing made by casting as a single unit (including inlet and outlet devices). The manufacture of such complex castings and their processing are associated with technological difficulties, therefore, as a rule, compressor housings are made of several parts connected by vertical transverse connectors using bolts (pins) or welding. It should also be taken into account that when casting the housing, it is very difficult to accurately make the channels of the inlet confuser and the outlet diffuser, respectively, at the inlet and outlet of the compressor. Since the inner surface of these channels becomes inaccessible for machining, such a design requires a high level of foundry production. Usually, to reduce pressure losses in the air duct, curved inlet and outlet channels are subjected to machining, both to achieve accurate geometric dimensions and to obtain a clean and smooth surface.

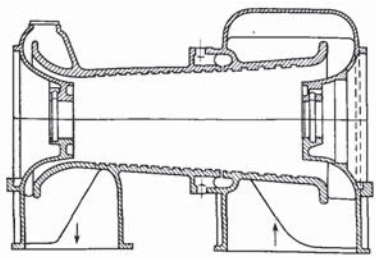


Fig. 6.12. Compressor housing made by casting as a single unit

The front housing of the compressor is usually a casting of steel, magnesium, or aluminum alloys, in which the central part (box) is connected by means of cast hollow struts to the outer ring.

The front housing usually houses the compressor front support, the inlet guide device, the heating system, and various communications. It also houses the central transmission bevel gears and drive boxes, for which there are special flanges on the housing for installation and centering.

Fig. 6.13 shows the front compressor housing of the Zorya-Mashproekt D050 marine engine. It houses the front support of the low-pressure compressor 5 and the inlet guide device 2. The outer and inner walls of the front housing 1 are connected to each other by six struts.

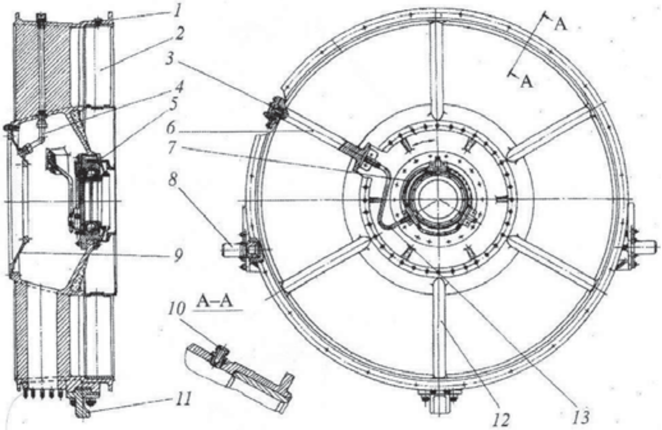


Fig. 6.13. Front housing of the D050 engine compressor [15]

On the outer wall are fixed two trunnions 8 for fastening the engine to the frame, a retainer 11 for transverse fixation of the engine, six fittings 10, to which the compressor flow path flushing manifold is connected, and two fittings for measuring static pressure in the compressor.

Oil for lubricating the bearing of the front support of the low-pressure compressor 5 is supplied through drilling in the side inclined strut 6, tube 13, and collector 7. A filter 3 is installed at the oil inlet to the rack. Oil is drained from the oil cavity of the support through the lower hollow strut 12.

The inlet guide device 2 consists of inner and outer rings and vanes. The outer ring is fixed in the front housing with screws.

Pipe 4 is designed to lead wires from the tachometer sensor.

The front housings can be structurally simplified when using a supersonic stage as the first stage of the compressor. In this case, the drives to the units are placed not in the front, but in the rear housing of the compressor. The front housings can be made of sheet steel in the form of a thin outer ring-shell and a small-diameter inner box or housing for placing the front support bearing. The ring is connected to the box with thin struts by welding. The role of the struts can be performed by the inlet guide vanes (Fig. 6.14).

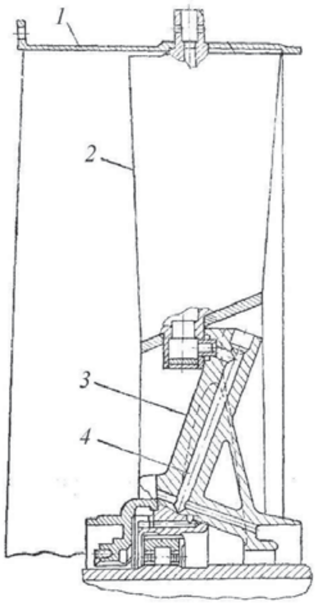


Fig. 6.14. Power connection through the 1st stage guide vanes [15]:

1 – ring-shell; 2 – guide vane; 3 – bearing housing; 4 – oil supply channel

The middle housing of the compressor provides a power connection between the front and rear housings. Guiding vanes are mounted inside the middle housing. The walls of the middle housing together with the rotor form the flow path of the compressor.

The middle housings of axial compressors are made non-separable and separable (Fig. 6.15).

Non-separable housings have the lowest mass, uniform rigidity along the circumference and are easy to manufacture. However, the installation of the compressor and guide vanes with a non-separable housing is complicated and requires disassembly of the rotor. It is also difficult to control the gaps and inspect the compressor blades and vanes.

With a small wall thickness, the housing is provided with longitudinal and transverse ribs to increase rigidity.

The most common are **separable housings**. The housing can be separated in one or more longitudinal and transverse planes.

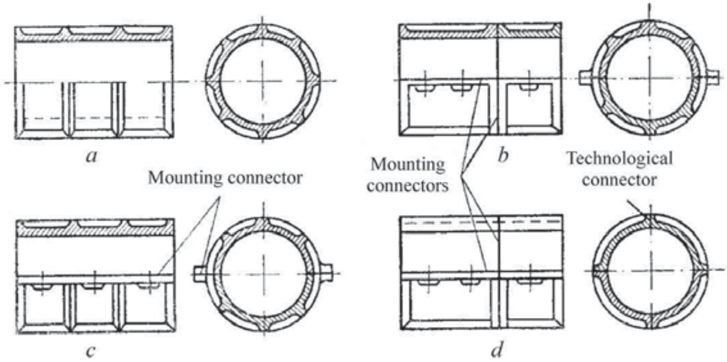


Fig. 6.15. Design of medium compressor housings:

a – non-separable; *b* – with longitudinal horizontal and transverse connectors; *c* – with longitudinal horizontal connector; *d* – with two longitudinal (vertical and horizontal) and one transverse connector

The presence of a longitudinal connector facilitates assembly, since in this case the guide devices are assembled in the halves of the housing and both halves, when assembled, are fastened with bolts along the flanges (using the so-called **mounting connector**).

With a large diameter of the housing, to facilitate casting, each of its halves is made of two parts, which, after processing, are fastened together at the flanges, forming **a technological connector**, and are then mounted as integral parts.

The presence of transverse connectors improves the manufacturability of the housing and facilitates the choice of material for each of its parts. For example, if the front part can be made of aluminum or even magnesium alloys, then the rear part, which perceives significant pressure and loads and is exposed to high temperatures, is most often advisable to be made of steel.

Thus, separable housings facilitate the assembly of the compressor but increase its weight. Additional flanges increase the volume and complexity of machining, require fixing and centering of parts, and ensure the tightness of joints.

Mutual fixation of the halves of the separable housings is usually ensured by bolts, and the tightness of the joint is achieved by careful processing of the flanges and the installation of connecting bolts or studs with a small pitch.

In the case of a disk-drum type rotor with disks connected by end splines, it is advisable to assemble the housing from separate rings (Fig. 6.16). This simplifies the attachment of guide devices, increases the manufacturability of the housing, and reduces its weight.

According to the manufacturing method, middle housings are distinguished by cast and welded.

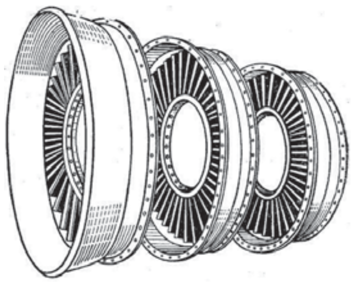


Fig. 6.16. Compressor housing, which consists of separate rings

The walls of cast housings are made of variable thickness to create uniform strength, which varies within 6-10 mm. The smallest wall thickness is limited by the casting properties of the material and the thickness of the elements combined in it (flanges, ribs, bosses). A significant thickness of the casting is impractical due to the deterioration of the casting structure, which reduces the strength limit of the material.

Housings made of titanium are used in aircraft and marine engines. The good weldability of this material allows you to connect housings made of sheet material with flanges and obtain a light, strong, and rigid housing structure.

Steel housings are made either from forgings by machining the internal and external surfaces (Fig. 6.17, *a*), or by welding from sheet steel 2-3 mm thick

(Fig. 6.17, *b*). Longitudinal and transverse ring flanges are welded to the sheets. T-shaped rings 3 can be welded to the inside of the housing to provide higher rigidity and to install the external band rings of the guide devices. After welding and subsequent heat treatment, the flanges and other ring profiles are bored. Then the housing is cut with a disk cutter along the longitudinal flanges. Therefore, during assembly, gaskets 7 are installed in the connector to preserve the shape of the housing (Fig. 6.17, *c*).

Windows are made in the walls of the housings, which are necessary for the bleeding of part of the air into the atmosphere (Fig. 6.18). The bleeding windows must be located evenly around the circumference so as not to cause a strong distortion of the velocity and pressure field in front of the blades. Otherwise, the blades may vibrate and break.

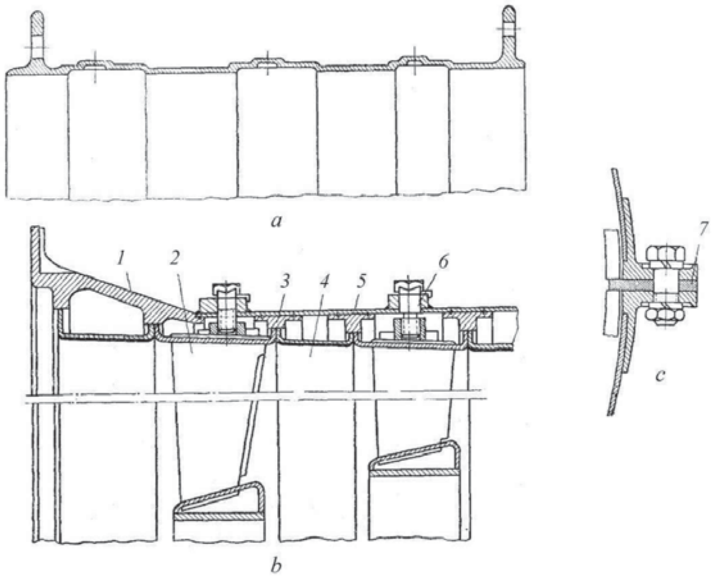


Fig. 6.17. Steel compressor housings [15]:

a – machined housing; *b* – welded housing; *c* – connection of welded housing halves; 1 – flange; 2 – guide vane; 3 – T-shaped ring; 4 – tract ring, 5 – housing wall; 6 – guide mounting screw; 7 – gasket

It is advisable to place the windows not above the blades, but behind them or above the guide vanes. When placing the windows in front of the blades, the axial gap between the guide vanes and the blades is increased.

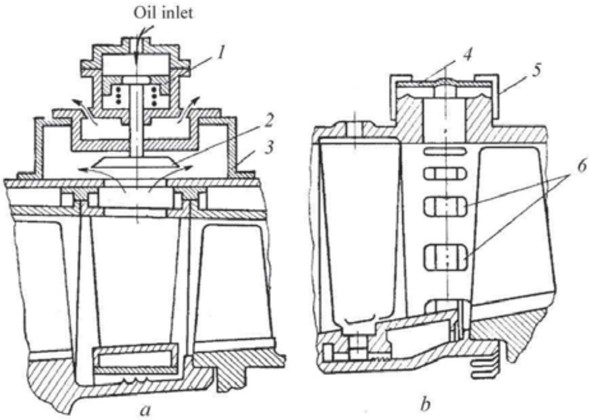


Fig. 6.18. Air bleeding from the compressor to the atmosphere [15]:
a – collector with bleeding valve; *b* – bleeding strip; 1 – piston; 2 – valve; 3 – collector, 4 – bleeding strip; 5 – strip opening limiter; 6 – bleeding windows

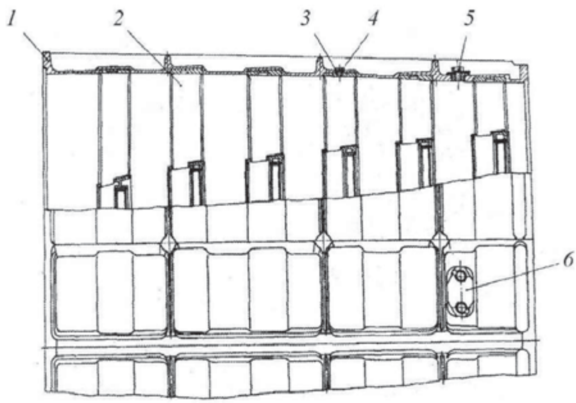


Fig. 6.19. Low-pressure compressor housing of the D050 engine [15]:
1 – compressor housing; 2 – guide device; 3 – lock; 4 – screw; 5, 6 – plugs

Above the windows, a collector is welded to the housing, at the outlet of which anti-surge bleeding valves or throttle valves are installed. The windows can also be closed using steel strips that cover the housing.

In some designs, inspection windows are made above the blades to monitor them during operation.

Fig. 6.19 shows the middle housing of the low-pressure compressor of the Zorya-Mashproekt D050 marine engine.

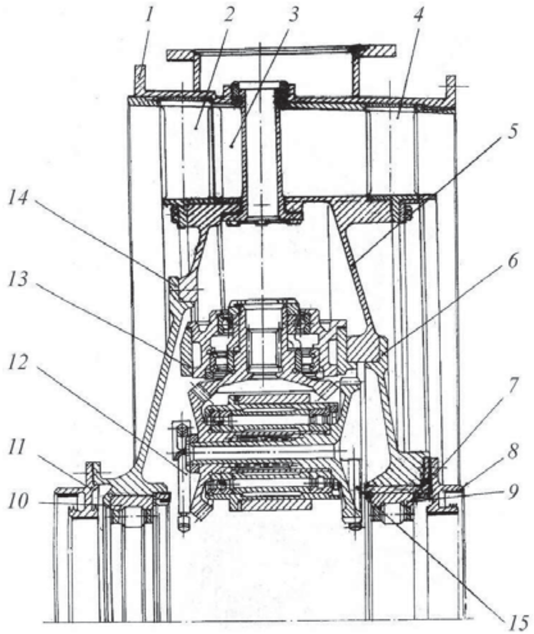


Fig. 6.20. Adapter between compressors of the D050 engine [15]:

1 – outer wall; 2 – 7th stage guide vanes of the low-pressure compressor; 3 – guide vanes at the outlet of the low-pressure compressor; 4 – inlet guide vanes of the high-pressure compressor; 5 – inner wall; 6 – wall of the front support of the high-pressure compressor; 7, 11 – labyrinth covers; 8 – sealing layer; 9 – front support of the high-pressure compressor; 10 – rear support of the low-pressure compressor; 12 – oil collector of the rear support of the low-pressure compressor; 13 – central drive; 14 – wall of the rear support of the low-pressure compressor; 15 – oil nozzle of the front support of the high-pressure compressor

The housing of the low-pressure compressor is made in the form of a hollow truncated cone, reinforced with stiffening ribs. The housing is separable in the horizontal plane. The rotor is fixed through three equally spaced holes during assembly and disassembly of the engine. The holes are closed with plugs 5. Plug 6 closes the window for inspecting the blades of the last stage of the low-pressure compressor.

Inside the housing, guide devices 2 are installed in six cylindrical grooves, which are attached to the housing with screws 4.

An example of the design of **an intermediate housing between low- and high-pressure compressors (adapter)** is shown in Fig. 6.20. Walls 6 and 14 are attached to the inner wall of the adapter, in which the bearings of the front support of the high-pressure compressor 9 and the bearings of the rear support of the low-pressure compressor 10 are installed.

The power connection between the bearing housings and the outer housing of the adapter is carried out using the guide vanes of the 7th stage of the low-pressure compressor 2 and the inlet guide vanes of the high-pressure compressor 4.

A central drive 13 with a manifold 12, which supplies oil to the rear support of the low-pressure compressor, is also attached to the inner wall of the adapter.

The oil cavity of the adapter is sealed by covers 7 and 11, the layer 8 on the surfaces of which, together

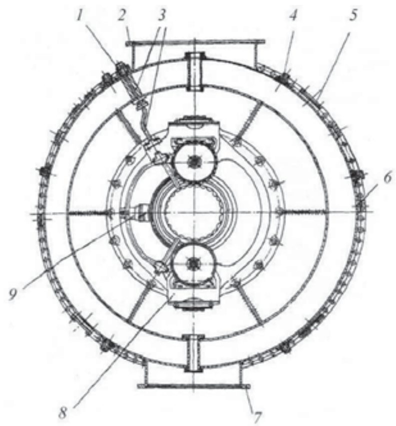


Fig. 6.21. Adapter between compressors of the D050 engine [15]:

1 – filter; 2 – upper horizontal platform; 3 – oil supply pipes; 4 – flushing fitting; 5 – air extraction flange; 6 – air parameter measurement fitting; 7 – lower horizontal platform; 8 – central drive; 9 – loading device

with the combs made on the rotors of the low-pressure compressor and the high-pressure compressor, forms labyrinth seals.

On the outer wall of the adapter are made (Fig. 6.21):

– upper horizontal platform 2 for mounting the upper drive box;

– lower horizontal platform 7 for attaching the lower drive box;

– four flanges 5 for air extraction from the flow path for ship needs;

– four fittings 6 for measuring air parameters in the flow path;

– six fittings 4 for flushing the flow path.

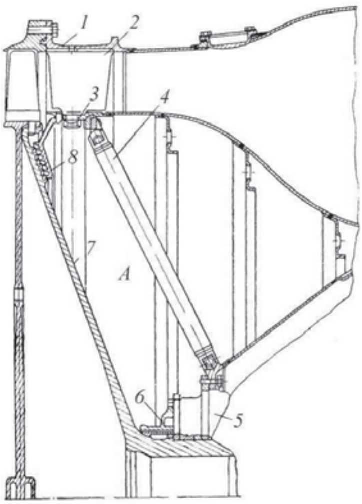


Fig. 6.22. Rear housing [15]:
1 – outer power ring (bandage), 2 – guide vane; 3 – inner power ring; 4 – brace; 5 – bearing housing; 6 – labyrinth seal of the bearing; 7 – rear trunnion of the compressor; 8 – labyrinth seal of the unloading cavity

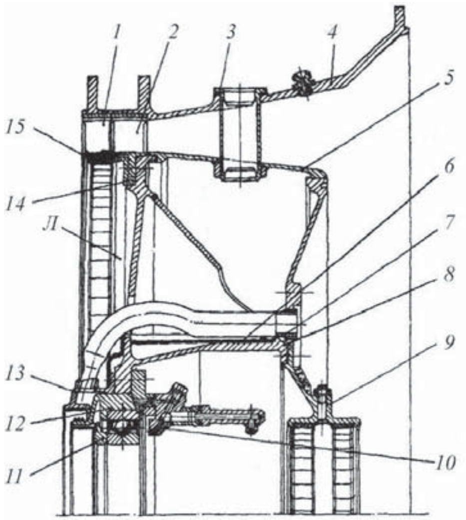


Fig. 6.23. Rear compressor housing of the D050 engine [15]:
1 – 7th stage high-pressure compressor guide vanes; 2 – exit straightening vanes of the high-pressure compressor; 3, 7 – blow-off pipes; 4, 8 – outer and inner walls of the diffuser; 5 – casing; 6 – heat-insulating casing; 9 – labyrinth cover; 10 – rear support collector; 11 – oil-reflecting ring; 12 – rear support housing; 13 – rear support cover; 14 – inner ring; 15 – insert

The rear compressor housing serves as a power connection with the combustion chamber housing and is one of the main parts of the engine power circuit.

The space between the rear housing and the compressor rotor is often used to organize **an unloading cavity,** which allows you to reduce the axial loads on the rotor supports. Fig. 6.22 shows the unloading cavity "A", which is separated from the gas path by a labyrinth.

The rear housing, shown in Fig. 6.22, is made of steel, welded construction, and consists of an outer power ring 1, an inner ring 3, the vanes of the guide device of the last stage of the compressor 2, which carry out the power connection between the outer and inner power circuits, and conical walls 4. The latter form a rigid truss for fastening the housing of the rear support of the compressor.

The rear housing becomes significantly more complicated when the drives of the units are located in it, since the placement of the gears and rollers of the central transmission requires the sealing of many cavities.

The intermediate and rear compressor housings are usually made by casting.

Fig. 6.23 shows the rear compressor housing of the D050 marine engine. It is an annular diffuser, through which the compressed air from the high-pressure compressor, reducing its speed, passes into the combustion chamber. It consists of outer 4 and inner 8 walls and a casing 5, interconnected by the exit straightening vanes 2.

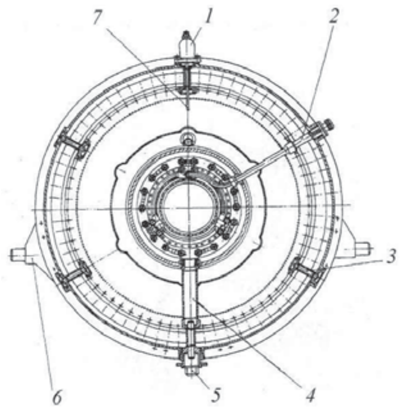


Fig. 6.24. Rear compressor housing of the D050 engine [15]:

1 – square; 2, 3, 4, 7 – pipes for oil supply, bleeding, oil draining and pressure measurement in the unloading cavity of the high-pressure compressor, respectively; 5 – retainer; 6 – trunnion

The outer wall of the diffuser is a power connection between the compressor housing and the combustion chamber. Two trunnions 6 on the outer wall (Fig. 6.24) are designed to fasten the engine to the frame, the retainer 5 is designed to transversely fix the engine.

The exit straightening vanes 2 (see Fig. 6.23) have a power function. Each of its vanes is bolted to the wall 8, and its shelf enters the groove of the outer ring, which is fitted with tension into the bore of the outer wall 4.

The guide vanes of the 7th stage of the high-pressure compressor 1 have not a power function. Its outer ring is seated in the bore of the outer wall of the diffuser with a gap. In addition, the outer rings of the vanes 1 and 2 are connected to the outer wall of the diffuser by radial pins.

The ring 14, fixed to the diffuser wall with the same bolts as the vanes 2, has metal-ceramic inserts 15 assembled in the annular groove, which, together with the combs made on the rear trunnion of the high-pressure compressor rotor, form a thin labyrinth seal that separates the rear unloading cavity "L" from the flow path of the compressor.

In the unloading cavity, the pressure is maintained, the magnitude of which determines the axial force on the thrust bearing of the high-pressure compressor rotor. To maintain the required pressure in the unloading cavity, air is vented from it through pipe 3 into the flow path behind the high-pressure turbine.

The pressure measurement in the unloading cavity is performed through tube 7 (see Fig. 6.24) installed in square 1.

The compressor housing designs considered earlier were single-walled. However, **double-walled compressor housings** are also quite often used in stationary and transport gas turbine engines (Fig. 6.25).

The outer housing of such structures is intended mainly for perception of internal pressure in the compressor; it is not subject to high requirements for manufacturing accuracy. The inner housing serves to accommodate the guiding devices. Its walls can have a minimum thickness, since they do not perceive pressure drops.

Double-walled housings have increased rigidity, which is especially important for multi-stage compressors with a significant length, since with low housing rigidity, it is possible for the blades to catch on the housing due to its deflection under the action of its own weight.

In double-walled compressor housings, the space between the outer and inner parts can be usefully used as a chamber for further air extraction to anti-surge valves or for turbine cooling.

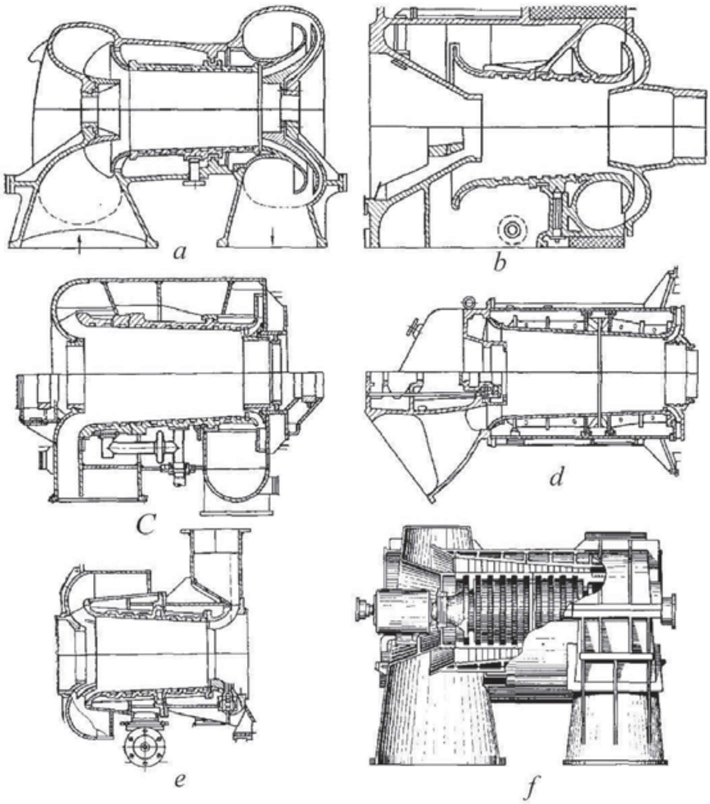


Fig. 6.25. Double-walled compressor housings [15]

6.3. Axial turbine housings

The turbine housing is a cylindrical or conical shell that provides placement and attachment of nozzle devices and a power connection with the combustion chamber housing and exhaust device (gas outlet). The inner walls of the housing limit the outer contour of the flow path of the turbine.

The turbine housing is loaded with torque and axial force from the nozzles and the exhaust device, internal excess pressure, and is also subjected to vibration loads.

Uneven heating of the housing along the thickness and especially along the circumference leads to its deformation, local reductions in radial clearances, and sometimes to the blades catching on the housing and the labyrinth combs on the sleeve.

The specified operational features determine the following **requirements for axial turbine housings**:

– the housing must be strong enough so that if the blade breaks or the disk breaks, its pieces do not pierce it;

– the housing must have significant and uniform rigidity in the axial and circular directions, as well as reliable centering in the cold and hot states, in order to ensure reliable operation of the turbine with small radial gaps between the housing and the tips of the blades and in the labyrinth seals.

From the latter follows **the main task of designing turbine housings** of gas turbine engines of all types - maintaining their original cylindrical shape during numerous changes in temperature conditions during operation.

The design of the housing depends on the location of the turbine rotor support, the method of attaching the nozzle devices, the manufacturing technology and method of cooling the parts of the nozzle devices and the walls of the housing itself, the number of turbine stages, and the design of the rotor.

Turbine housings of gas turbine engines are usually made of composite, with transverse and horizontal mounting connectors. **Transverse connectors** provide more uniform circular rigidity of the housing. In aircraft-type engines, they are most often made for each stage.

When using **a horizontal connector**, installation, inspection, and even blade replacement are facilitated without disassembling the turbine. However, with a horizontal connector, the circular rigidity of the housing is reduced, and there is a risk of its distortion.

The connection of the turbine housing parts is carried out by flanges, ensuring reliable centering along the shoulders. The flanges of the transverse connectors are also the stiffening ribs of the housing.

Housings without horizontal connectors are usually used in aircraft-type engines. In cross-section, such a housing is a cylindrical ring, a ring in the form of a truncated cone or in the form of a shell of a stepped profile, depending on the design of the flow path of the turbine and the number of its stages. The housing is usually assembled from several sections along flanges located perpendicular to the turbine axis. The advantage of this design is the comparative simplicity of manufacture. For example, aircraft-type turbine housings are made from a strip of constant thickness; it is rolled into a ring, the ends are welded, and the turned flanges are welded by gas or contact welding. Housings are also made from profiled rolled strip. By rolling the strip into a ring and butt-welding it, a housing with minor allowances is obtained, after which it is mechanically processed.

Housings made from cast billets are more expensive due to the significant amount of machining and significant material waste in chips.

Horizontally connected housings are more typical for stationary gas turbine engine turbines. The presence of a flange bolted connection limits the speed of starting a turbine with hot housings, since, due to uneven heating, the housing loses its cylindrical shape over time.

To increase the rigidity of the housings, they are sometimes made with annular ribs. Individual parts of the housing are centered using cylindrical collars or bolts. With a significant difference in temperature deformations of individual parts of the housing, the bolt holes are made elliptical with a radially located major axis.

Fig. 6.26 shows the turbine part of the Zorya-Mashproekt UGT10000 engine as an example of such a design.

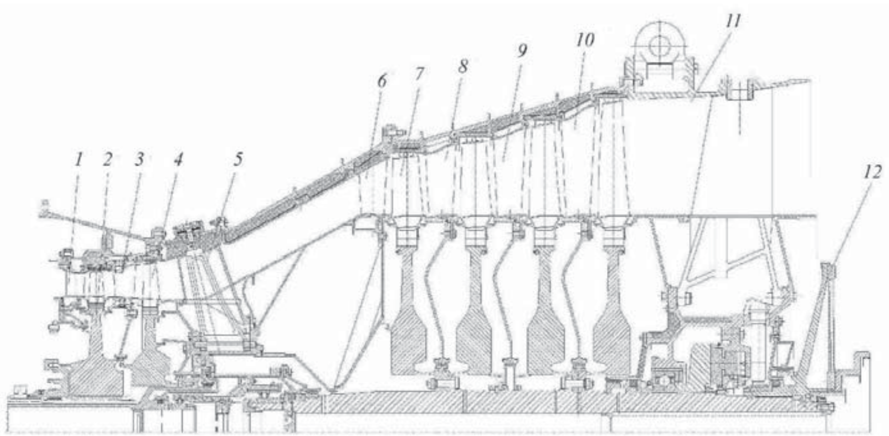


Fig. 6.26. Turbine part of the Zorya-Mashproekt UGT10000 engine [12]:
1, 3, 6, 8, 9, 10 – nozzle devices of turbine stages; 2 – trunnion-disk of high-pressure turbine; 4 – rotor of low-pressure turbine; 5 – rear support of low-pressure turbine; 7 – rotor of power turbine; 11 – rear support of power turbine; 12 – coupling

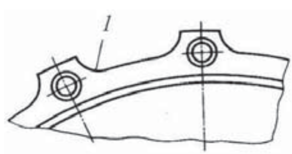


Fig. 6.27. Turbine housing flange

To reduce thermal stresses in the flanges, and therefore, their less distortion and prevent shrinkage, recesses 1 are milled in the flanges between the holes for the mounting bolts (Fig. 6.27), which reduce the weight of the housing.

A more radical way to prevent shrinkage and warping of the housings is to cool their inner walls with air taken from the combustion chamber cavity. The air enters the cavity formed externally by the power housing, and from the inside by the nozzle device rings and intermediate rings forming the flow path, or by the nozzle vane shelves extended along the gas flow. A turbine of this type is shown in Fig. 6.28.

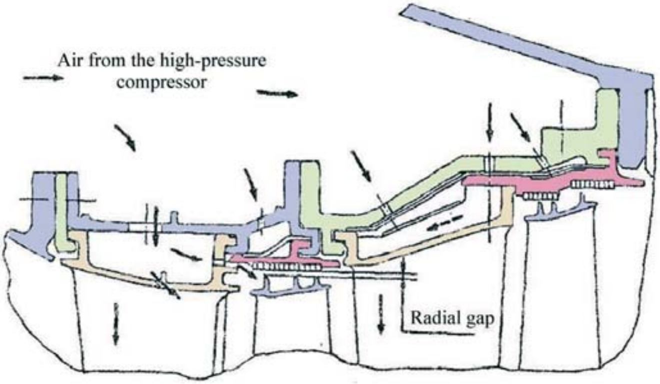


Fig. 6.28. Cooling scheme of high and low pressure turbine housings of the Zorya-Mashproekt UGT15000 engine [15]

Housings of such turbines can be single-walled or double-walled.

In **a double-walled housing design**, the outer housing withstands the full gas pressure but is insulated from the hot flow; the inner housing is in contact with the gas, takes on its temperature, but is not under pressure. The outer housing, as less heated, can be made of carbon steel or pearlite steel, and the inner housing can be made of heat-resistant steel or alloy. To maintain concentricity, the outer and inner housings must be free to move relative to each other.

The space between the walls of a double-walled housing can be used for blowing cooling air (as in the first stages of multi-stage turbines) or for filling with thermal insulation material (as in the subsequent stages).

Blowing cooling air in operating modes provides the housing with a constant "cold" state and stability of thermal expansion during a sharp change in operating mode. The fact is that gas turbine engine turbines have specific operating modes associated with a sharp increase in temperature during start-up and sudden cooling during an emergency stop. The latter mode determines the value of the minimum radial gap between the tips of the blades and the housing: the cold working fluid (no longer gas, but air) is directed into the turbine, and since the housing has less thermal inertia than the rotor, it cools faster and decreases in diameter. The use of a double-walled housing with cooling in the main operating mode allows you to avoid excessive shrinkage of the housing and the selection of radial gaps in specific modes.

For the same purpose, many modern turbines have a special radial clearance adjustment system (discussed earlier in that book).

In the design shown in Fig. 6.29, cooling air is supplied to the space between the outer housing 2 and the inner inlet pipe 6. The reflecting sheet 5 directs the air along the housing; then the air passes through the holes in the inner housing and then through the holes in the body of the nozzle vanes 3, and enters the periphery of the rotor, flowing around the fastening of the blades. This creates a film of cold air over the entire surface of the rotor in the area of the first two stages. Air is also supplied through special holes to cool the edges of the blades of the 1st stage.

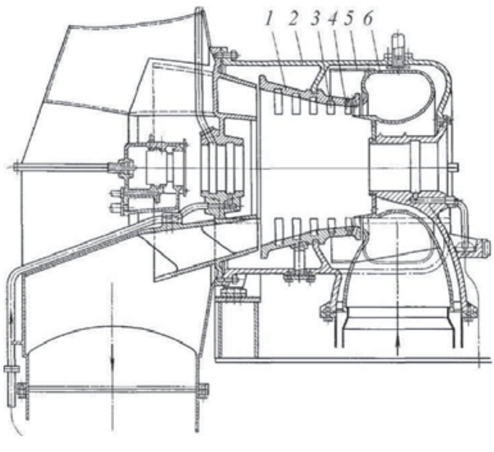


Fig. 6.29. Turbine with a double-walled housing [15]:

1 – inner housing; 2 – outer housing; 3, 4 – nozzle vanes; 5 – reflecting sheet; 6 – inner inlet pipe

Similar to the double-walled housings is the design of **single-walled turbine housings with internal inserts** (see Fig. 6.28). They are used mainly in gas turbine engines designed for rapid load build-up. The outer "cold" housing perceives only the pressure of the working fluid, and the elements that limit the flow path - its temperature. The role of the inner housing is played by individual segmental inserts (for example, nozzle vanes), suspended from a relatively cold power housing. To allow for the freedom of expansion of the inserts, a tangential gap is provided between them.

The single-walled design of the housing does not allow for such cooling and, therefore, the housing in this case can be described as "hot". However, the design scheme of a turbine of this type is very simple and therefore has some applications. In the turbine shown in Fig. 6.30, the radial gap between the tips of the blades and the housing is about 1.8 mm, and its value is controlled by several electronic sensors and one optical sensor, which is installed tangentially in the area of the 3rd stage. The optical sensor allows direct visual observation of the gap value.

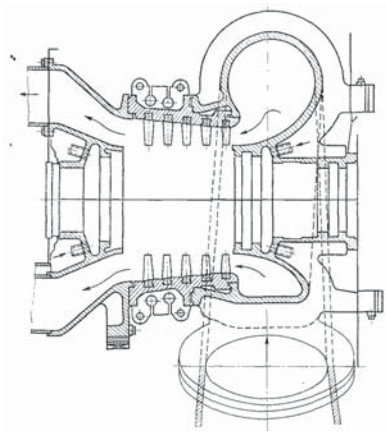


Fig. 6.30. Turbine with a single-walled housing [15]

Cooling of single-walled housings, however, can also be done externally – by blowing air or pumping water through boxes or casings welded to the outer surface of the housing. The air for such cooling can be taken from the compressor or supplied from a separate blower – the latter option is associated with lower energy losses.

Fig. 6.31 shows one of the possible solutions for external air cooling of the housing. Cooling air passes into the gap between the housing and the thin-walled casing or through channels formed by halves of pipes welded to the outer surface of the housing.

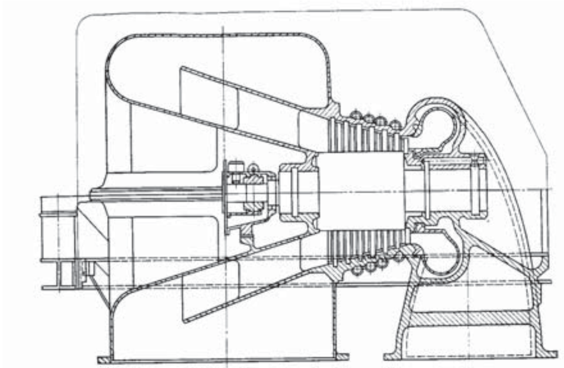


Fig. 6.31. Turbine with a single-walled housing and external cooling [15]

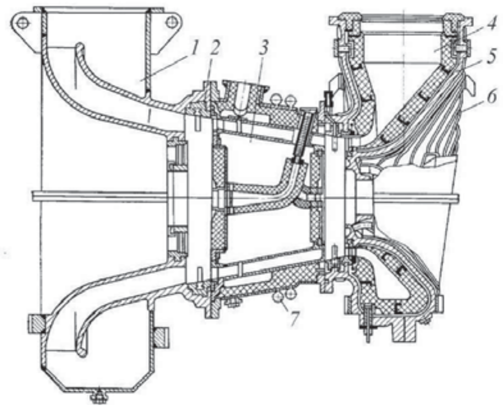


Fig. 6.32. Turbine housing of GTU-20 [15]:

1 – outlet duct, 2 – nozzle vane holder of the low-pressure turbine; 3 – intermediate duct; 4 – inner part of the inlet duct; 5 – high-pressure turbine housing; 6 – channels for water passage; 7 – compensator

Water cooling of the housing is more efficient than air cooling and provides practically unchanged dimensions of the housing with minimal energy consumption. To avoid contamination of the cooling channels, closed circulation systems are used, filled with purified water, which is cooled in a special water-to-water heat exchanger. Such a system is used, for example, in the GTU-20 gas turbine power plant installed on the ship "Paris Commune" (Fig. 6.32). In it, the cooling channels are formed by halves of pipes welded to the housing with a pitch of 110 mm. In this power plant, at an initial gas temperature of 1023 K, the temperature at different points of the outer surface of the housing was within 323-423 K (50-150 °C) [15].

The power connection between the turbine housing and the bearing housing is carried out in various ways, the choice of which is largely determined by the design of the combustion chamber. Nozzle vanes, due to significant temperature deformations, are usually not included in this power connection. Other elements (struts, pins, spokes, etc.) are used for this.

Let's consider some examples of the power connection between the turbine outer housing and the bearing housing.

The support housing of the D050 marine engine is shown in Fig. 6.33. This housing serves to accommodate the rear support of the high-pressure turbine rotor and the front support of the low-pressure turbine rotor; in fact, it acts as an adapter between these turbines. It consists of an outer housing 1, a bearing housing 16, and six radial struts 5. The struts with the outer housing 1 are connected by flanges 3, which are attached to it with screws and pins. Flange 3 is placed on the strut with a gap and is attached to it with a screw 4. The struts with the bearing housing 16 are connected by pins 7 and 9.

In the bearing housing, there are slots drilled at the front and rear for mounting sleeves 14 and 17, into the bores of which the outer rings of roller bearings 15 and 18 are installed, secured by rings 12 and 20. Labyrinth covers 11 and 21 are attached to the bearing housing at the front and rear.

The outer 2 and inner 24 casings insulate the housing from hot gases and form the flow path.

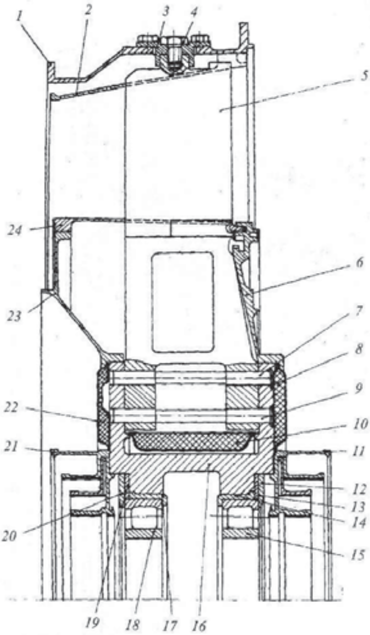


Fig. 6.33. Support housing of the D050 engine [15]:

1 – outer housing; 2, 24 – outer and inner casings; 3 – flange; 4 – screw; 5 – strut; 6 – rear wall; 7, 9 – fingers; 8, 10, 22 – casings; 11, 21 – labyrinth covers; 12, 20 – clamping rings; 13, 19 – adjusting rings; 14, 17 – sleeves; 15, 18 – bearings; 16 – bearing housing; 23 – front wall

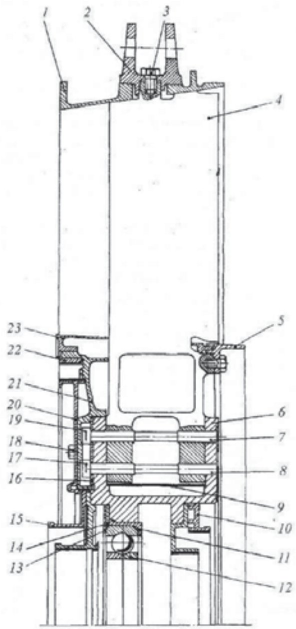


Fig. 6.35. Rear support housing of the D050 power turbine [15]:

1 – outer housing; 2 – rim flange; 3, 18 – screws; 4 – strut; 5, 20, 21 – walls; 6 – bearing housing; 7, 8 – fingers; 9 – casing; 10, 15 – labyrinth covers; 11 – sleeve; 12 – bearing; 13, 14 – clamping and adjusting rings; 16, 22 – rings; 17, 19 – sleeves; 23 – inner casing

Between the casings in the flow path of the support housing, there are three fairings (Fig. 6.34), through which oil pipes pass. The fairings enter the slots on the inner and outer casings.

The rear support housing of the D050 power turbine is shown in Fig. 6.35. It is designed to accommodate the rear support of the power turbine rotor and forms the flow path between the turbine and the exhaust device. It consists of an outer housing 1, a bearing housing 6 and seven struts 4. The struts with the outer housing 1 are connected by means of flanges, which are attached to the housing with screws and pins. The flanges are placed on the struts with a gap and are attached to them with screws 3.

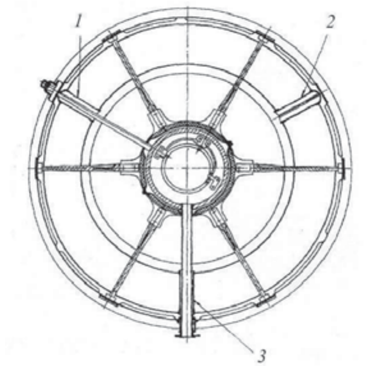


Fig. 6.34. Cross-section of the support housing of the D050 engine [15]:
1 – oil supply pipe fairing; 2 – fairing; 3 – oil drain pipe fairing

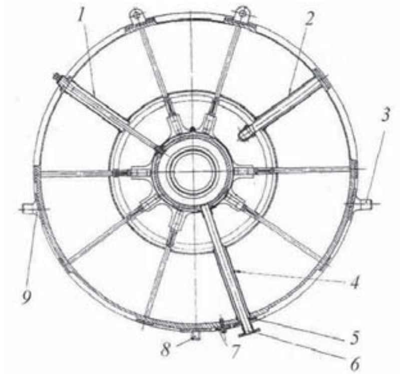


Fig. 6.36. Cross-section of the rear support housing of the D050 power turbine [15]:
1 – fairing of the oil supply pipe and limitation of the power turbine spin-up; 2 – fairing of the air supply pipe; 4 – fairing of the oil drain pipe; 3, 9 – trunnions; 5 – flange; 6 – oil drain pipe; 7 – fitting; 8 – retainer

The struts with the bearing housing 6 are connected by pins 7 and 8.

A slot is bored in the bearing housing for mounting the sleeve 11, into which a ball bearing 12 is installed, secured by a clamping ring 13. Sealing labyrinth covers 10 and 15 are attached to the bearing housing at the front and rear, and walls 20 and 21 are attached at the front with screws 18.

The flow path is formed by the outer housing 1 and the inner casing 23. In the flow path of the support housing there are three fairings 1, 2, 4 (Fig. 6.36), through which oil, air pipes and a pipe for limiting the speed of the power turbine spin-up pass. The fairings enter the slots of the inner casing, and are fixed on the outer housing with flanges 5.

Consider the support housing of the low-pressure turbine of the Zorya-Mashproekt UGT25000 engine, shown in Fig. 6.37. It consists of a outer housing 2, six radial struts 6, connected to the outer housing by means of thermal expansion compensators 4, an inner housing 9, connected by pins to the struts 6 and in the lower part by bolts to the bearing housing 11 and the cone. The housing, in which the rear support of the low-pressure turbine rotor is located, is attached to the bearing housing 11 with screws.

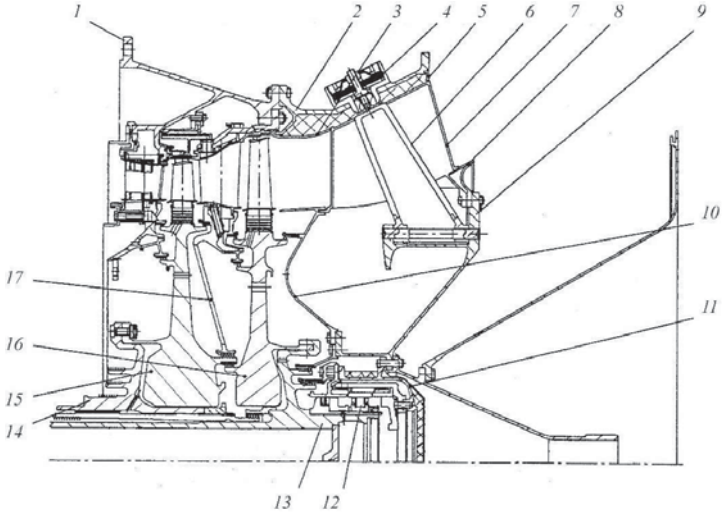


Fig. 6.37. High and low pressure turbines of the UGT25000 engine [12]:
1 – turbine housing; 2 – outer housing; 3, 5 – outer casing; 4 – compensator; 6 – strut; 7 – fairing; 8 – inner casing; 9 – inner housing; 10 – front wall; 11 – bearing housing; 12 – roller bearing; 13 – low-pressure turbine shaft; 14 – high-pressure turbine trunnion; 15 – high-pressure turbine disc; 16 – low-pressure turbine disc; 17 – diaphragm

The flow path in the support housing is organized by external casings 3 and 5 and the front wall 10. To reduce thermal expansion of the housing 2, the cavity between it and casings 3 and 5 is filled with thermal insulation basalt fiber. Three fairings are placed between the external and internal casings, through which the air supply pipe, the oil supply pipe and the oil drain pipe pass.

The rear support of the low-pressure turbine rotor is a roller bearing. A damper is fitted to the outer ring of the bearing with tension, designed to shift the critical number of rotor revolutions to lower values and to damp the energy of rotor vibrations. To increase the pressure in the oil gap on the oil outlet side of the damper, a sealing ring is provided. An adjusting ring is used to regulate the oil flow through the damper.

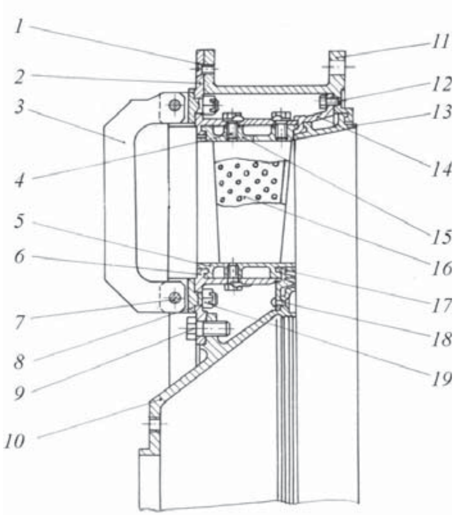


Fig. 6.38. Nozzle device of the 1st stage of the DI59 engine [6]:

1 – screw; 2 – outer housing; 3 – spoke; 4 – screw; 5 – inner shoe; 6 – inner housing; 7 – finger; 8 – eyelet; 9 – screw; 10 – screen; 11 – power casing; 12 – screw; 13 – insert; 14 – power ring; 15 – outer shoe; 16 – deflector; 17 – screen ring; 18 – rivet; 19 – nut

Fig. 6.38 shows the nozzle device of the first stage of the Zorya-Mashproekt DI59 engine. The nozzle device consists of a power 11, external 2 and internal 6 housings, nozzle vanes, external 15 and internal 5 shoes, screen 10, spokes 3, power ring 14 and insert 13.

The power connection between the housing 11 and the screen 10 is carried out using radial spokes 3.

6.4. Gas exhaust devices

The axial flow velocity at the outlet of the last stage of the turbine of a gas turbine engine is, as a rule, 150-180 m/s[1]. At the same time, the kinetic energy of the exhaust gases reaches 10% of the useful energy. (For comparison, in steam turbines, the axial velocity at the last stage reaches 250 m/s, but the kinetic energy of the steam flow does not exceed 1.2-1.5% [19].) Therefore, the development of a rational design of the diffuser of the gas exhaust device (gas outlet) for turbines of gas turbine engines is a specific and important problem.

Despite the variety of design features, t**he main requirements for gas exhaust devices** include:

– minimum total pressure loss;

– minimum dimensions and weight;

– manufacturability;

– reliability in operation.

The importance of ensuring minimal total pressure losses is explained by the fact that an increase in gas outlet pressure by 100 mm Hg (approximately 1 kPa) reduces turbine power by approximately 1% [11].

Due to the high temperature of the exhaust gases, the walls of the exhaust devices must be reliably insulated: thermal insulation creates normal working and safety conditions for the operating personnel; in addition, it prevents heating of the bearing housings located inside the exhaust, gearbox, shaft-rotating device, drive mechanism, etc., as well as foundations, walls, control panels, and other adjacent elements. In power plants with exhaust heat recovery, insulation of the exhaust reduces unwanted heat loss to the environment.

[1] For example, for Zorya-Mashproekt engines: UGT6000 (modification DV71) – 72 m/s, UGT15000 (DG90) – 148 m/s, UGT16000 (DT59) – 169 m/s, UGT2500 – 196 m/s [9].

A layer of insulating material is usually applied to the outside of the exhaust device, and in power plants with double-walled exhaust devices, the insulation fills the space between the walls. Instead of a layer of mineral insulation, an air gap or shielding sheets can be used.

Minimal pressure losses are provided by **axial-type gas exhaust devices**. Figures 6.39 and 6.40 (also see Figures 6.8 and 6.9) show examples of gas turbine engines with similar devices.

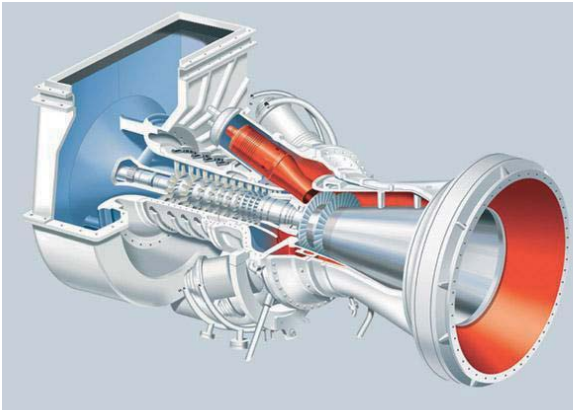


Fig. 6.39. Siemens Energy SGT-300 engine with axial-type exhaust device

It should be taken into account that the gas is usually discharged through vertically arranged gas ducts (except in cases where the gas is directed to horizontally arranged waste heat boilers). Therefore, the flow must still be turned in a radial direction. When designing, it is important to choose the distance at which this turn will be made, i.e., determine the length of the axial diffuser.

Examples of gas exhaust devices with a shorter axial diffuser and radial rotation are shown in Fig. 6.41. In these designs, the flow is turned at an angle of 90° using guide vanes with a quarter-circle cross-section.

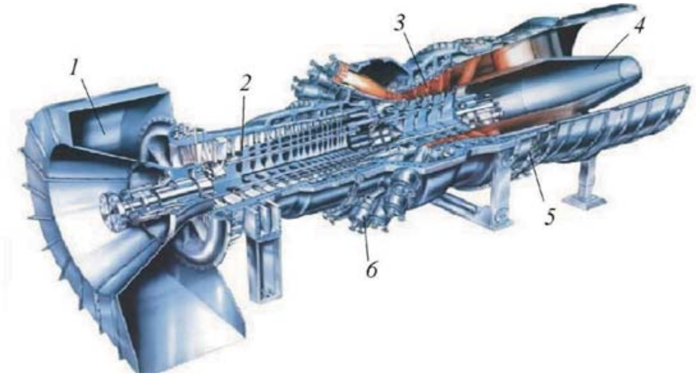


Fig. 6.40. Siemens Energy SGT6-6000G engine with axial-type exhaust device:
1 – inlet device; 2 – compressor; 3 – turbine; 4 – gas exhaust device; 5 – housing; 6 – combustion chamber

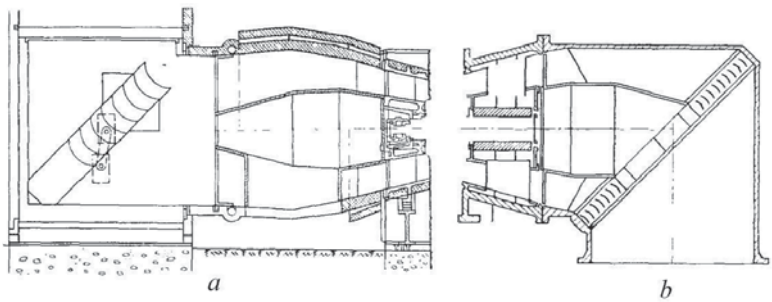


Fig. 6.41. Axial-type gas exhaust devices [15]

Axial-type exhaust devices are typical for gas turbine engines in which the driven mechanism is located on the compressor side. In most gas turbine engines, axial-type exhaust devices cannot be implemented due to the limited length of the engine or due to the fact that the driven mechanism is installed behind the turbine. Therefore, in the vast majority of gas turbine engines, **radial-type exhaust devices** are used, which consist of two zones: the first is a diffuser to reduce the speed, the second is a gas collector to divert the flow from the annular section behind the diffuser into a rectangular or circular section of the gas duct.

Depending on the design of the gas collector, **radial-type gas exhaust devices are divided into snail-type and knee-shaped ones**.

A design feature of **a snail-type exhaust device** is the presence of an axial-radial diffuser (gas enters it in the axial direction and exits in the radial direction), a gas-collecting snail, and an exhaust section that ends with an ejection nozzle or flange. Examples of gas turbine engines with a snail-type exhaust device are shown in Figs. 6.42 and 6.43.

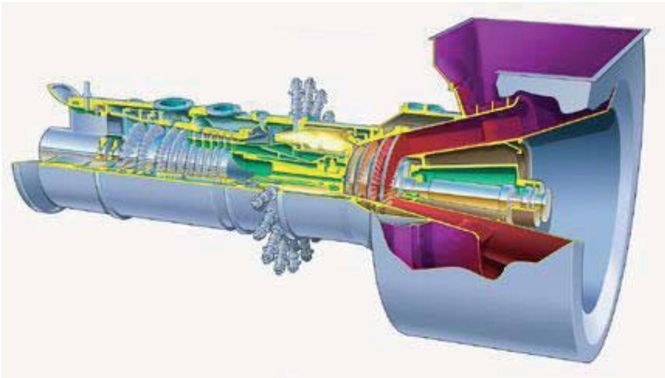


Fig. 6.42. Siemens Energy SGT-600 engine with a snail-type exhaust device

A distinctive feature of the snail-type exhaust device is its relatively small length with developed radial dimensions. The ratio of the length to the outer diameter of the gas inlet into the device, according to [9], is approximately 0.7-1.1; the ratio of the outer diameter of the snail (gas collector) to the outer diameter of the gas inlet is 1.5-2.0; the ratio of the outer diameter of the snail to its length is 1.35-2.85.

The snail-type gas exhaust devices are mainly made of 3 mm-thick chrome-nickel steel sheet and thin flanges, which provide a power connection between the diffuser and the snail. The snail is equipped with profiled stiffening ribs on the outside.

Fig. 6.43. GE Aerospace LM2500 engine with a snail-type exhaust device

It should be noted that pressure losses in a snail-type exhaust device increase significantly with decreasing snail diameter and length.

Knee-shaped radial exhausts have a higher length than the snail-type exhausts (the ratio of length to the outer diameter of the exhaust inlet is 1.2-2.0), with radial dimensions close to the outer diameter of the turbine. As noted in [9], this is not always acceptable from the point of view of limiting the permissible outer shaft length, but reducing the length of the knee leads to increased pressure losses.

An example of a gas turbine engine with a knee-shaped exhaust is shown in Fig. 6.44.

Knee-shaped exhaust devices are often used in gas turbine engines of hovercrafts and hydrofoils. **Their main advantages** (compared to the snail-type devices) are significantly lower weight and diameter. As noted in [9], knee-shaped exhaust devices for the same type of engine turned out to be about three times lighter with similar efficiency.

Knee-shaped devices are considered especially effective for structures with a flow rotation of less than 90°.

Fig. 6.44. Zorya-Mashproekt UGT2500 engine with a knee-shaped exhaust device

They are usually made of 1.5 mm-thick chrome-nickel steel sheet with stamped stiffening ribs of semicircular cross-section from the same sheet. The connection to the engine on the outer diameter is carried out through a floating ring, on the inner one, telescopically.

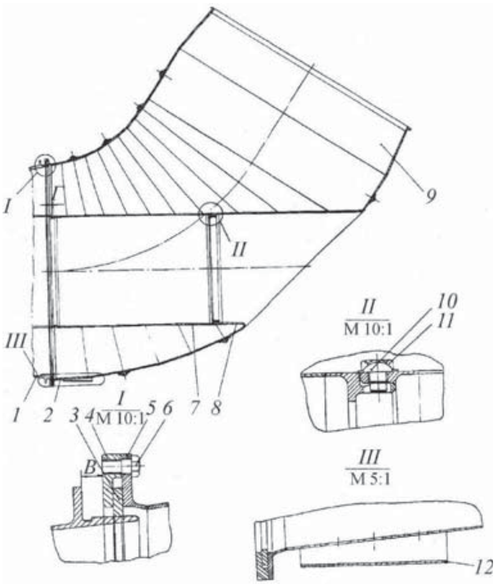


Fig. 6.45. Knee-shaped exhaust device of the D050 engine [15]:

1 – turbine housing; 2 – casing; 3 – floating ring; 4 – ring; 5 – flange; 6, 11 – screws; 7 – inner casing; 8 – rear wall; 9 – exhaust device housing; 10 – lock washer; 12 – rib

The exhaust of the D050 marine engine is shown in Fig. 6.45. It is designed to discharge exhaust gases into the ship's smokestack with a flow turn at an angle of 70° to the engine axis. It consists of a housing 9, a rear wall 8, and an inner casing 7.

The exhaust device housing 9 is a welded construction made of sheet steel and is combined with the rear wall 8 into an integral unit. Its upper part is conical and freely enters the ship's smokestack.

The rear wall 8 is a hollow truncated cylinder made of sheet steel; its rear end is welded to the exhaust device housing, and the front end ends with a flange.

The inner casing 7 is made in the form of a hollow truncated cone with telescopic rings at the ends. Its connection with the turbine housing and the rear wall is of the telescopic type. The position of the inner casing is fixed by six screws 11 screwed into the front flange of the rear wall 8, which enter the holes of the rear ring of the casing 7 (see element II).

The connection of the exhaust device with the outer housing of the turbine is carried out through a floating ring 3, which is pressed against the flange 5 of the exhaust device housing by ring 4. Ring 4 is attached to the housing flange by screws 6.

To ensure the possibility of thermal expansion at the junction of the exhaust device housing and the inner casing with the turbine housing, mounting gaps are provided.

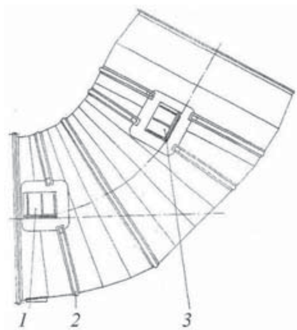


Fig. 6.46. D050 exhaust device housing [15]:
1, 3 – supports; 2 – ribs

In the lower part of the exhaust device (see element III), three drainage holes are made to drain fuel and oil that may enter from the engine into the heat-insulating casing of the exhaust device.

To increase the rigidity, the exhaust device housing has ribs 2 (Fig. 6.46). To attach the exhaust device to the heat-insulating casing, four supports 1 and 3 with bolt holes are welded to its housing.

6.5. Frames and heat-insulating containers (casings)

To ensure higher rigidity, **the gas turbine engine is mounted to the foundation on a frame**. The design of the frame and the method of its connection to the engine must take into account the possible thermal expansion of the engine housing, and for transport gas turbine engines, also minimize possible dynamic loads (for example, from side and keel pitching when used on ships, etc.).

Let us consider as an example the frame of the D050 marine engine. The location of the engine and auxiliary mechanisms on it is shown in Fig. 6.47.

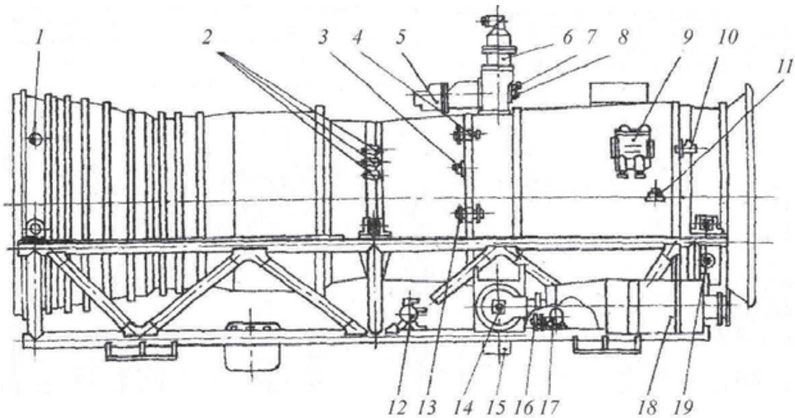


Fig. 6.47. Location of the D050 engine and auxiliary mechanisms on the frame (right side view) [15]:

1 – speed sensor; 2, 10, 19 – pressure sensors; 3, 4 – shut-off and drain valves; 5 – fuel pump-regulator; 6 – centrifuge; 7 – tachometer sensor; 8, 14 – centrifugal sensors; 9, 17 – oil and fuel filters; 11, 16 – pressure sensors; 12 – electric oil injection pump; 13 – electromagnetic valve; 15 – oil unit; 18 – turbostarter

The frame (Fig. 6.48) is a welded structure consisting of two side trusses 1 of tubular cross-section, interconnected in three places by transverse ribs.

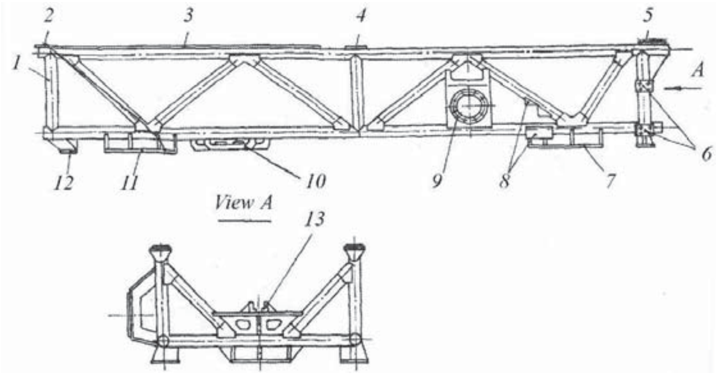


Fig. 6.48. Engine frame D050 [15]:

1 – side trusses; 2, 4, 5 – engine mounting platforms; 3 – casing mounting platforms; 6 – turbostarter mounting plates; 7, 11, 12 – frame mounting platforms to the foundation; 8 – electric starter bracket mounting plates (for modification D050L1); 9 – remote drive box mounting flange; 10 – oil tank mounting bracket; 13 – retainer box

On the upper part of the frame, six platforms 2, 4, and 5 are welded for attaching the engine supports, and platform 3 is welded for attaching the engine casing. On the lower part, six platforms 7, 11, and 12 are welded, with the help of which the frame is attached to the ship's foundation, and a bracket 10 for attaching the oil tank.

On the right side of the frame, a flange 9 is welded to the remote bracket for mounting the remote drive box, and on the front pillar, there is a plate 6 for mounting the turbostarter (for the D050L engine modification) or a plate 8 for mounting the electric starter bracket (for the D050L1 modification).

Three boxes of retainers 13 are welded to the transverse ribs, with the help of which the engine is fixed from transverse movements.

The engine is installed and mounted on the frame on six supports.

The front and middle supports (Fig. 6.49, *a*) are similar in design; they support the weight of the engine, allowing axial displacement during thermal elongation of its housing.

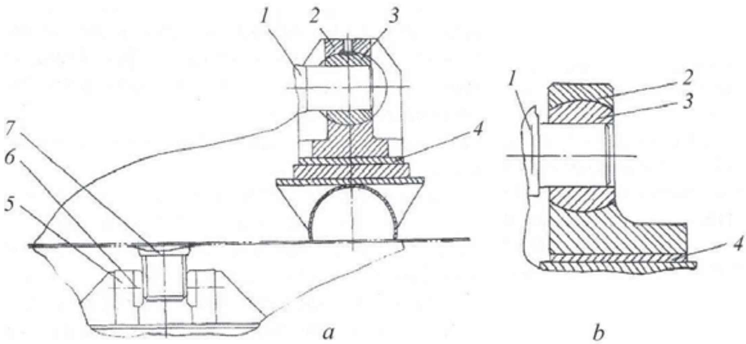


Fig. 6.49. Front (a) and rear (b) supports for attaching the D050 engine to the frame [15]:

1 – trunnion; 2 – support housing; 3 – liner; 4 – gasket; 5 – retainer box; 6 – adjustment strips; 7 – retainer

The weight is perceived by trunnions 1, fixed (on the right side and on the left side) on the front housing of the low-pressure compressor and on the rear housing of the high-pressure compressor. The trunnions enter the cylindrical liners 3, installed in the housings of the supports 2.

The supports are attached to the frame: the front ones with four screws, the middle ones with four bolts each. Between the support and the frame, gaskets 4 are installed, designed to adjust the engine height.

The axial displacement of the engine is ensured by the sliding ability of the liner 3 in the support housing.

Transverse (circular) fixation is achieved by installing retainers 7, fixed on the front housing of the low-pressure compressor and on the rear housing of the high-pressure compressor in the grooves formed by the adjusting strips 6, installed in the retainer boxes 5.

Two rear supports (Fig. 6.49, *b*), mounted on the support housing of the power turbine, take the weight of the engine and fix it in the axial direction.

The trunnions 1, fixed on the support housing of the power turbine, enter the spherical liners 3, installed in the housings of the rear supports 2. Each support is attached to the frame with four bolts. The purpose of the gaskets 4 is similar to the purpose of the gaskets of the front and middle supports.

In the transverse (circular) direction, fixation is carried out by a retainer, the design of which is similar to the front retainer.

It should be noted that the reference scheme assumes an axial shift of the engine housing during heating in the direction of the air inlet. At the same time, no additional loads will be transmitted during thermal expansion towards the drive mechanism (located behind the engine).

Let's consider the mounting on the frame of the Zorya-Mashproekt UGT25000 engine (Fig. 6.50).

The frame consists of longitudinal 1 and transverse 2 I-beams. In the lower part of the frame there are support platforms 3, with which it is attached to the engine, in the upper part - platforms 4 and 5 for attaching pedestals 12 and 13 under the front flexible supports; in the front part – platforms 6 for attaching the remote drive box, in the rear part - support platforms 7 and 9 for attaching the rear engine supports and platform 8 for the engine retainer 24.

The engine is attached to the frame using four flexible supports 11, 14, 17 and 18.

The front supports 11 and 14 take the weight of the engine and fix it in the transverse direction. The supports consist of two flexible plates 15, attached to the upper and lower support flanges by means of clamping strips and bolts. The upper part of the front supports 11 and 14 is attached to the compressor housing with screws, the lower part is attached to the pedestals 12 and 13 through adjusting gaskets 16. The pedestals 12 and 13 are mounted on the frame through adjusting gaskets 19, which ensure the interchangeability of the engine on the frame.

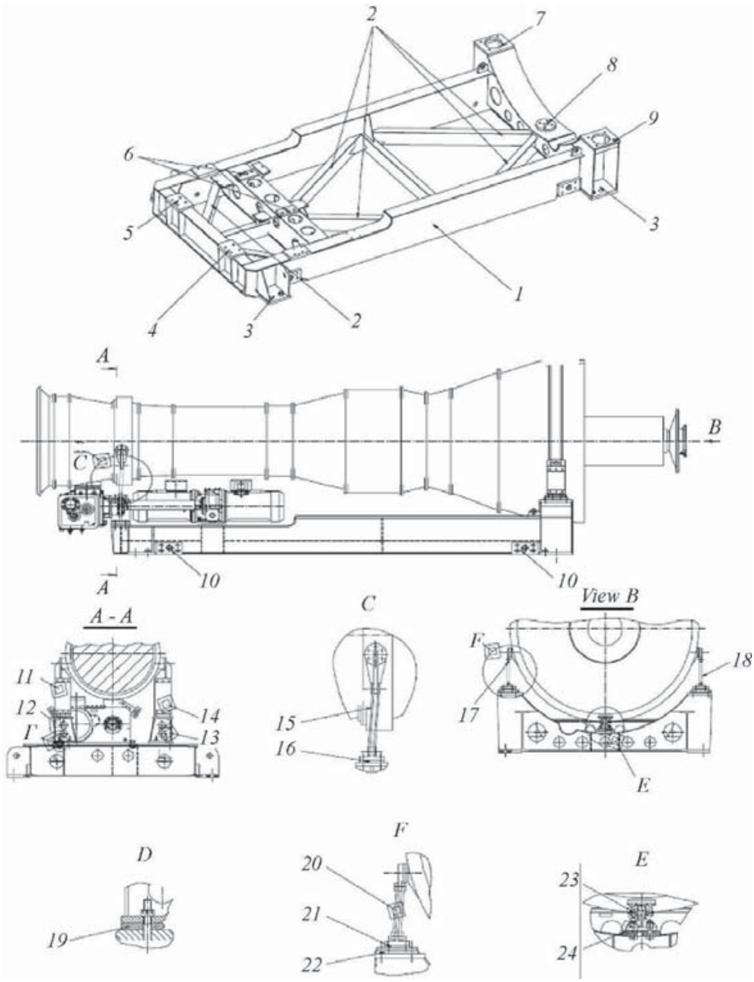


Fig. 6.50. UGT25000 engine frame and supports [20]

The rear supports 17 and 18 take the weight of the engine and fix it in the longitudinal (axial) direction. The supports consist of two flexible plates 20, attached to the upper and lower support flanges by means of clamping strips and bolts. They are attached to the trunnions of the support housing of the power turbine and are installed through the adjusting gaskets 21 and the intermediate plate 22 on the frame. The intermediate plate 22 ensures the interchangeability of the engine on the frame, and the adjusting gaskets 21 - adjustment of the engine position in height.

In the transverse direction, the engine is fixed by a retainer 24. The slats 23, secured with screws, are used to adjust the gaps in the retainer in the transverse direction.

On the sides of the frame there are four boards 10 for attaching technological skids for rolling out the engine.

A gas turbine engine is a source of significant thermal radiation and high-intensity noise. Each of these undesirable effects is usually strong enough to make it impossible for people to be in the engine room unless the engine is placed inside **a special container or casing**.

When designing such containers, it is necessary to take into account that it is not possible to enclose the engine in a thick layer of insulating material, since in this case the heat transfer from the engine housing to the outside will deteriorate and the housing will overheat.

The main elements of the containers are panels made of sound-absorbing and heat-insulating materials. Access to the interior of the container must be provided. The panels must be easily removable if necessary to facilitate rapid engine replacement.

Cooling of the space inside the container, usually carried out by air, is necessary due to the high temperature of the gas turbine engine surfaces. Fig. 6.51 shows a graph of the surface temperature of the housing of the GE Aerospace LM2500 marine engine when operating in nominal mode and taking into account the pumping of cooling air through the container.

Cooling air can be pumped through the container either by the ejector action of the outgoing stream of flue gases in the exhaust device or by the operation of a fan.

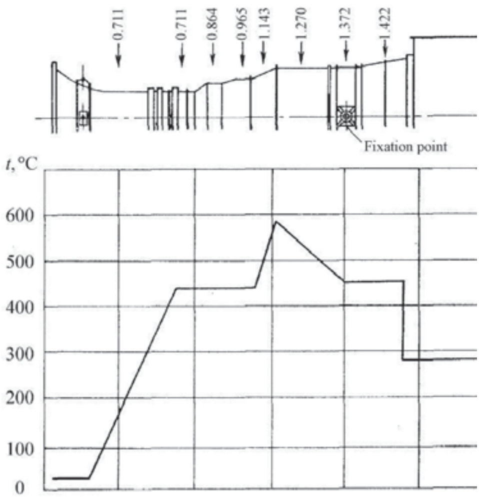


Fig. 6.51. Surface temperature of the LM2500 engine housing at nominal mode (the upper part of the figure shows approximate diameter values in meters) [9]

Figs. 6.52 and 6.53 (also see Fig. 6.4) show the placement of various marine gas turbine engines in thermal and acoustic insulating containers.

In some cases, the heat-sound insulating container is installed only above the hot part of the engine, which usually includes the last stages of the compressor, the combustion chamber, and the turbine. In this case, the term **heat-sound insulating casing** is often used to designate it. An example of such a design is the casing of the D050 marine engine (Fig. 6.54).

The engine casing 1 is attached to the engine frame by means of bolts. It consists of one section with a longitudinal connector in a vertical plane. The halves of the casing section are connected by flanges by means of bolts.

The casing 1 has a hatch 2 in the upper part for inspection and replacement of thermocouples. The casing has no rigid connection with the engine.

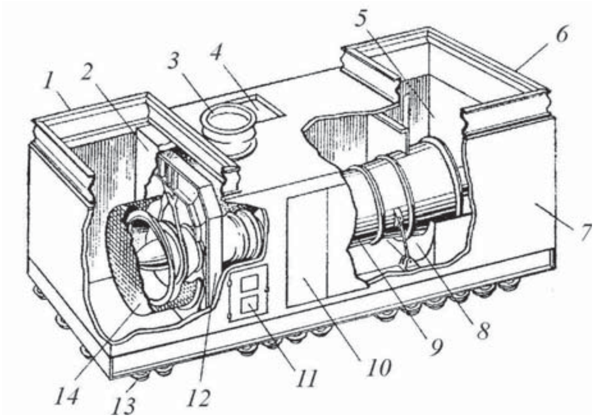


Fig. 6.52. Placing the LM2500 engine in a heat- and sound-insulating container:
1 – air intake duct compensator; 2 – container partition; 3 – air intake for cooling the container; 4 – hatch, 5 – gas exhaust device; 6 – exhaust device compensator; 7 – container; 8 and 12 – gas turbine engine supports; 9 – engine; 10 – removable panel; 11 – door; 13 – shock absorbers; 14 – inlet grid

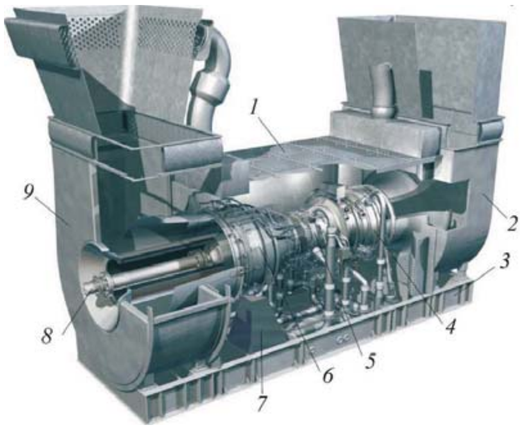


Fig. 6.53. Placement of the Rolls-Royce MT30 engine in the container:
1 – container; 2 – air intake; 3 – frame; 4 – engine gas generator; 5 – combustion chamber; 6 – power turbine; 7 – lubrication system unit block; 8 – power take-off shaft; 9 – gas exhaust device

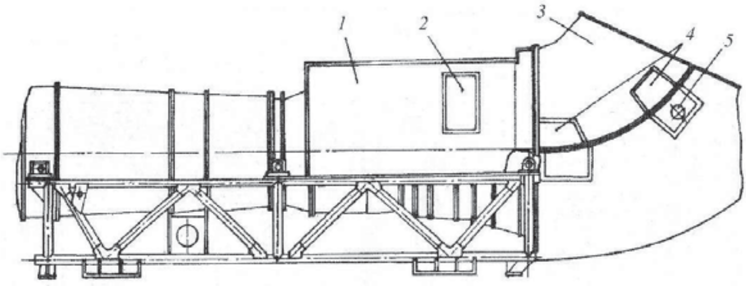


Fig. 6.54. D050 engine casing and exhaust device [15]:
1, 3 – engine and exhaust casings; 2, 4 – hatches, 5 – trunnion

The casing of the exhaust device 3 is heat-sound insulating, forms an air cavity around the exhaust device. Through this cavity, due to the ejection action of the exhaust gases, cold air is pumped, which provides cooling of the exhaust device. The casing consists of two parts - upper and lower, connected by flanges with bolts. On the lower part, there are two removable trunnions 5 for fastening to the ship's foundation, and on the upper part, four hatches 4 for access to the exhaust device supports.

The exhaust device casing 3 is flanged with bolts at the top to the engine casing 1, and at the bottom to the frame. The ship's gas duct is attached to the flange of the upper part of the casing.

A special **fire-extinguishing system** is often installed under the heat-sound insulating container (casing). Such a system is necessary because fuel, lubricating oil, and hydraulic control oil are supplied under pressure through the engine pipelines, and hot surfaces of the housings can become a source of their ignition in the event of a leak or rupture of the pipeline. Visual detection and manual extinguishing of a fire are difficult because the engine is inside the container (casing). For this reason, it is important to provide remote fire detection and activation of the alarm, as well as the supply of fire-extinguishing agents using a remotely controlled system.

The most effective way to extinguish burning petroleum products is with water (due to the water mist produced by boiling). Carbon dioxide, dry chemicals in the form of foam, and halogenated hydrocarbons are also suitable for this purpose.

Control tasks and questions

1. Explain the purpose of the inlet devices of gas turbine engines.
2. Explain the structural features of axial-type, ring-type, and radial-type inlet devices.
3. Explain the structural features of axial compressor housings.
4. Describe the structural features of turbine housings.
5. Describe the structure of axial-type gas exhaust devices.
6. Analyze the structure of radial-type gas exhaust devices.
7. Explain the purpose and structure of gas turbine engine frames.
8. Describe the purpose and structure of heat-sound insulating containers of gas turbine engines.

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У навчальному посібнику розглянуто конструкції, принцип роботи та типові схеми суднових газотурбінних двигунів, що застосовуються на морському транспорті. Проаналізовано конструктивні особливості та сфери застосування основних складових суднових газотурбінних двигунів, зокрема елементів проточних частин компресорів і турбін, роторів осьових турбомашин, підшипників і ущільнень, а також корпусних деталей. Наведено огляд сучасних конструкцій, що використовуються в суднових енергетичних установках. Видання призначене для здобувачів вищої освіти спеціальності G11 «Машинобудування», а також для студентів, які навчаються за освітньою програмою «Морська енергетична інженерія та технології».

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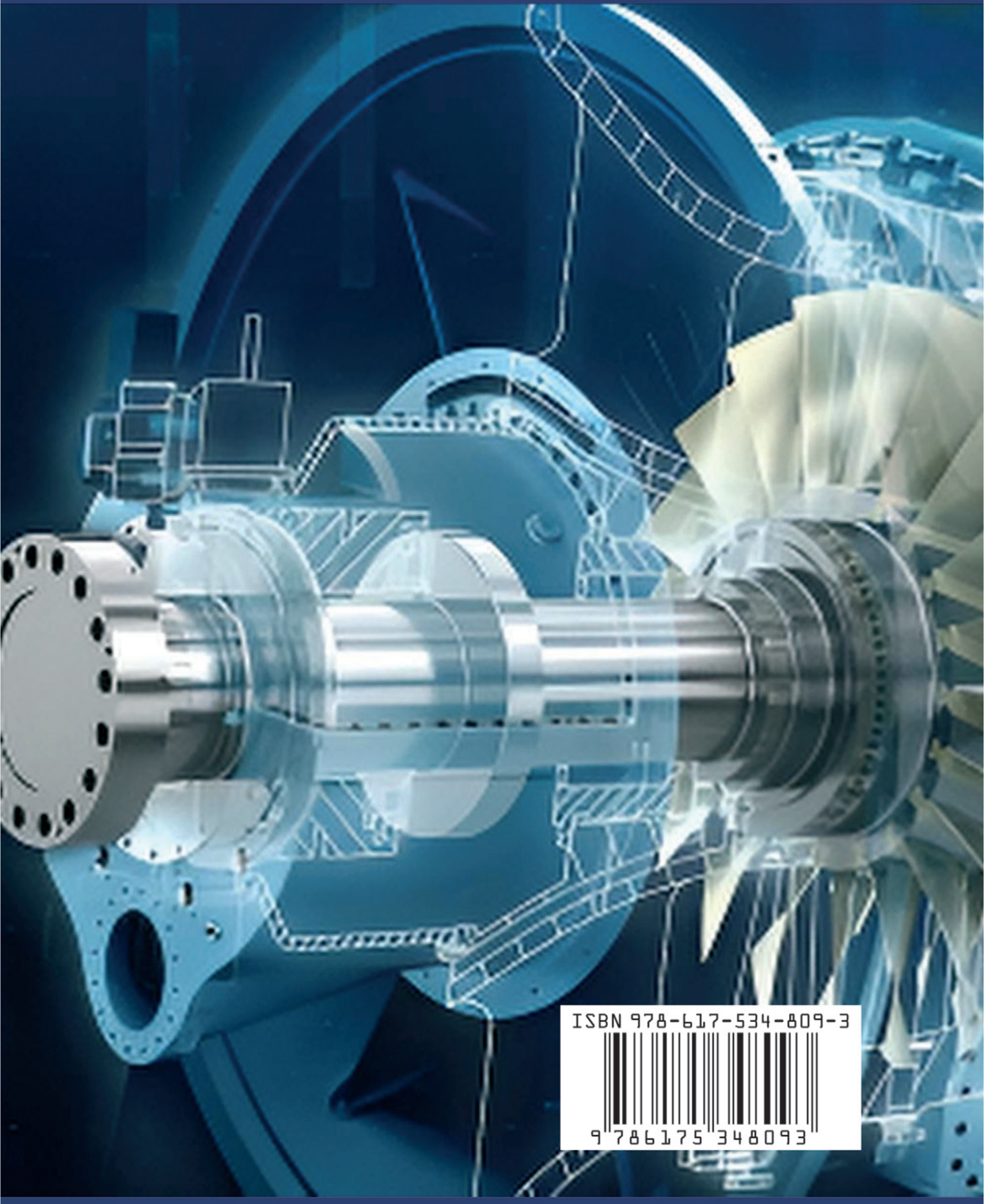

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