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"MAINTENANCE AND REPAIR OF AIRCRAFT AND AVIATION ENGINES "

Subject: The method of evaluation of the influence of GTE operation conditions and its technical state of the damage accumulation in its turbine parts

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Project supervisor: Candidate of Technical Sciences, Associate Professor Yakushenko A.S.

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STATEMENT
on performance of the diploma paper
Chernenko Bogdan
(Full name of the graduate)

Subject: "The method of assessing the effect of operating conditions GTE and its technical condition of the accumulation of damages in the parts of the turbine" approved by order of the Rector on October 2020 ¹ /st.

2. Term of work: from October 05, 2020 to December 27, 2020.

3. Fundamental data of the work. Describe the components and basic characteristics of the typical GTE designs. Learn the basic requirements for the design and operation features of the turbine GTE. Analyse physical processes occurring in the turbine blade during GTE operation. To investigate the damage accumulation in the turbine parts, manifesting itself at the operation of aviation GTEs. To substantiate the assessment methods of the damageability of GTE parts. Carry out thermodynamic and gasdynamic calculation of gas turbine engines. Analyze the influence of operating and technological factors on the damageability. To suggest decision making on control actions based on the results of damage parameters monitoring.

4. Contents of the explanatory note: Analysis of GTE and their turbine operation. Method of assessing the influence of GTE operation conditions and technical condition of damage accumulation in its turbine parts. Estimation of the influence of GTE operating conditions and their technical state of accumulation

Health and Safety and Environmental Protection Conclusions. The list of sources of information. 5. List of mandatory illustrative material: tables, drawings, diagrams, charts, graphs.

Calendar plan

No	Task	Timeline	The completion mark
1.	Analysis of GTE operation and their turbines	11.10.2020.	Done
2.	Motor test stations and their structure, schemes and equipment	25.10.2020.	Done
3.	Method of estimation of influence of GTE operation conditions and its technical state of accumulation of damage of its turbine parts	01.11.2020.	Done
4.	Study of damage accumulation in the turbine parts, which manifests itself during the operation of aviation GTEs	15.11.2020.	Done
5.	Assessment of the impact of GTE operating conditions and their technical state of accumulation	30.11.2020.	Done
6.	Thermodynamic and gasdynamic calculation of turbofans.	04.12.2020.	Done
7.	Execution of occupational safety section	07.12.2020.	Done
8.	Execution of environmental protection section	08.12.2020	Done
9.	General work conclusions	11.12.2020.	Done
10.	Calculation and explanatory note	14.12.2020.	Done
11.	Presentation	18.12.2020.	Done
12.	Prior defense of the diploma paper	11.10.2020.	Done

7. Consultants for individual sections of work

Section	Consultant	Date, Signature	
		The assignment was given by	Assignment accepted
Labor safety	assistant professor Kazanets V.I.		
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Date of issue of the assignment October 05, 2022.

Thesis supervisor: ______________________ Yakushenko A.S.

Assignment received for execution _____________________ Paramonov V.G.

REFERAT

Explanatory note to the qualifying master's work "Method of evaluation of influence of conditions of GTE operation and its technical condition of accumulation of damage in its turbine parts".

126 p., figures 33, tables 6, bib. j. 47

In the qualifying diploma work the research of methods of evaluation of influence of GTE operation conditions and their technical state of accumulation of damage in its turbine parts is offered.

The aim of the research is to investigate the methods of assessing the influence of GTE operating conditions and their technical state of damage accumulation in its turbine parts.

The object of the research is GTE and its turbine parts.

The subject of the research is the method of estimation of influence of GTE operation conditions and their technical state of damage accumulation in its turbine parts.

When performing this qualification work such general scientific methods as: analysis, synthesis, analogy, formalization, abstraction, generalization, observation, description, comparison and mathematical modeling were used.

In the innovative part of the work the model of estimation of influence of GTE operating conditions and technical condition of damage accumulation in its turbine parts is analyzed and proposed for use.

The standardized methods for determining the relevant indicators are used in the sections of occupational safety and environmental protection.

OPERATION, GTD, TECHNICAL CONDITION, ACCUMULATION, DAMAGE, PART, TURBINE

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LIST OF ABBREVIATIONS

GTE – gas turbine engine;

AJE - air-jet engines;

TJE - turbojet engine;

TJEA - turbojet engine with an afterburner;

TPE – turboprop engine;

TFE – turbofan engine;

LPC - low pressure cascade;

HPC - high pressure cascade;

MPC - medium pressure cascade;

SD- straightening device;

WB – working blade.

INTRODUCTION

Relevance of the topic. Today gas turbine engines are an integral part of life of almost the entire population of the Earth. They are widely used in power plants of different types, aviation (civil and military), shipbuilding, etc. Work on improvement of gas turbine engines in different scientific branches, including development of new metals, is carried out.

One of the main fields of research in aviation of metals regarding gas turbine engines is the development of materials with lower weight or heat resistant alloys working at critical temperatures for the metal, i.e. above 1050°C.

Particularly acute is the issue of component heat resistance (thermal stability of material parts) operating under conditions that are almost constantly approaching critical temperatures. The reason for this is that the possibility of increasing the temperature at which the components of a gas turbine engine also increase its efficiency and power characteristics.

That is why the fact that the search for ways to improve GTE parameters is indisputably important, and confirms the relevance of the topic.

Relevance of the work lies in the fact that in Ukraine gas turbine engines are widely used. Due to the fact that gas turbine engines are manufactured in our country and are the object of successful export, as well as repaired foreign-made gas turbines (such as aircraft used in the aircraft operated by Ukrainian airlines), there is a need to explore new ways to improve the characteristics of gas turbine engine, which can be used both in the design and in repair.

The aim of the study is to investigate the methods of assessing the influence of GTE operating conditions and their technical condition of damage accumulation in its turbine parts.

The object of the research is GTE and its turbine parts.

The subject of the research is the method of estimation of influence of GTE operation conditions and their technical state of damage accumulation in its turbine parts.

The list of tasks, which the author solves in the presented master's work:

- to describe the components and basic characteristics of typical GTE designs;
- to study the main requirements, imposed to the design and peculiarities of GTE turbine operation;
- to analyze physical processes arising in the turbine blade during GTE operation;
- to study damage accumulation in the turbine parts, which manifests itself during the operation of aviation GTEs;
- to analyze the ways of GTE turbine blades protection at operation and the ways of their durability increase;
- to substantiate the methods of assessing the damageability of GTE parts;
- to describe the general information on PS-90A turbo-jet propulsion system;
- to make thermodynamic and gasdynamic calculation of GTE;
- to analyze the influence of operating and technological factors on the damageability;
- to suggest decision making on control actions based on the results of damage parameters monitoring.

Such general scientific methods as analysis, synthesis, analogy, formalization, abstraction, generalization, observation, description, comparison and mathematical modeling were used during performance of this qualification work.

SECTION 1.
ANALYSIS OF GTE AND ITS TURBINE OPERATION

1.1 Components and basic characteristics of typical gas turbine engine designs

One of the simplest designs of gas turbine engine, for understanding its work, can be represented as a shaft on which there are two disks with blades, the first disk - compressor, the second - turbine, in a gap between them the combustion chamber is installed.

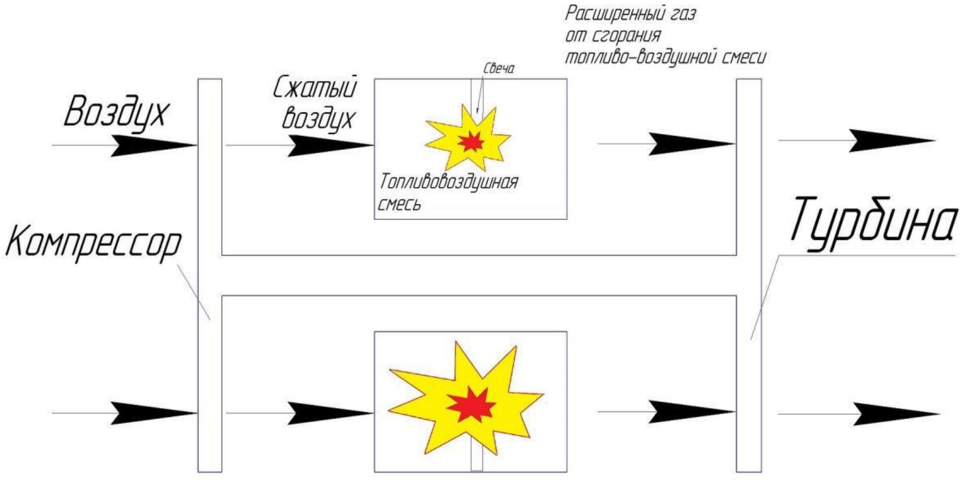


Figure 1.1 - Schematic representation of GTE operation [12]

In the study the turbofan engine with a high degree of dual-circuit is considered, because it is the most common in civil aviation and to date and displaced almost all other types of aviation GTE.

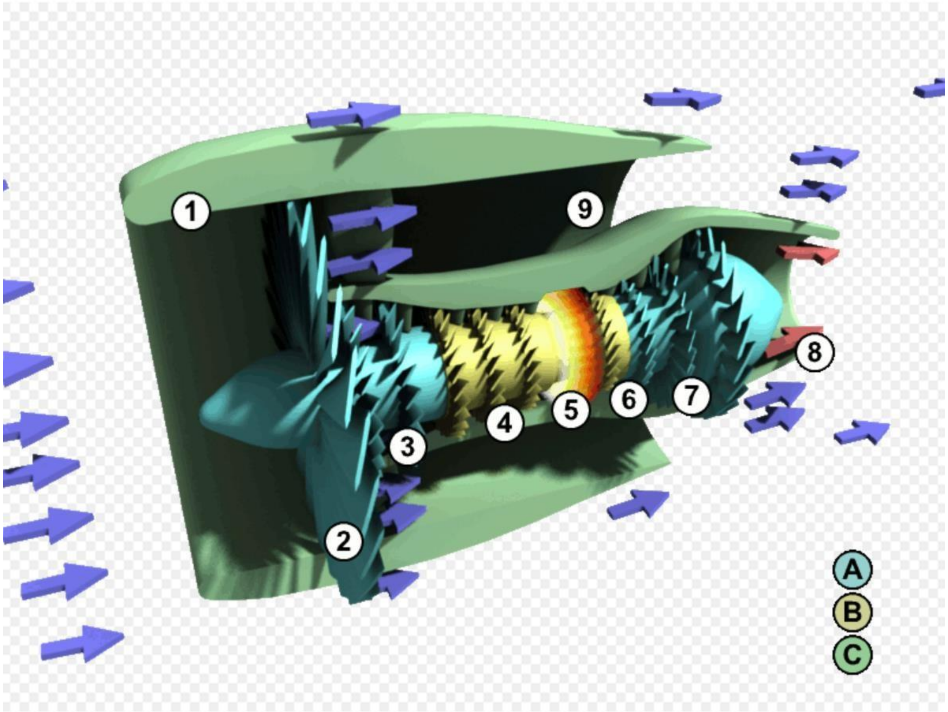


Figure 1.2 - Components of GTE [3]

The main components of a twin-turbofan engine with a high degree of dual circuit are: the low-pressure rotor, high-pressure rotor, stator components, nacelle, fan, low-pressure compressor, high-pressure compressor, combustion chamber, high pressure turbine, low gas nozzle turbine.

Among the components of a turbojet engine are such as the turbine. The turbine is one of the most loaded parts of the turbo-compressor, absorbing significant physical and thermal loads. For this reason, it is one of the most critical components, whose parts are repaired more often than other gas turbine engine components.

Despite the number of gas turbines sold in metal, the number of truly successful designs is not that great. Successful turbine designs:

- represent an entire family of structures;

- are sold in series or have the prospect of selling (at least on a single species scale) a sufficiently large series - several thousand pieces;

- proven reliability and durability in operation;

- ensure cost-effective production and maintenance [5].

The modern turbine GE90 is a basis for new developments (GE90-115B, GP7200). It is produced in series since 1995.

The main characteristics of the design are shown below (Fig. 1.3):

- two-shaft scheme (two-stage HPT + six-stage LPT);

- roller bearing and bearing HPT between HPT and LPT;

- bearing struts in the transition channel between turbines;

- two-stage highly loaded (expansion degree of about 5.5) HPT with working blades without shelves;

- high level of gas temperature before the rotor (estimated at 1850 K). For Rolls-Royce these are turbines of the RB211ATrent family, retaining all the main design characteristics for almost thirty years. These characteristics are exemplified by the RB211-535E4 turbine:

- three-wave scheme: single-stage HPT + single-stage TSD + three-stage THD;

- roller bearing and HPT support are combined with roller bearing and TSD support; common support is connected with TSD nozzle blades;

- single stage medium loaded (expansion ratio about 3.0) HPT with RL shelf;

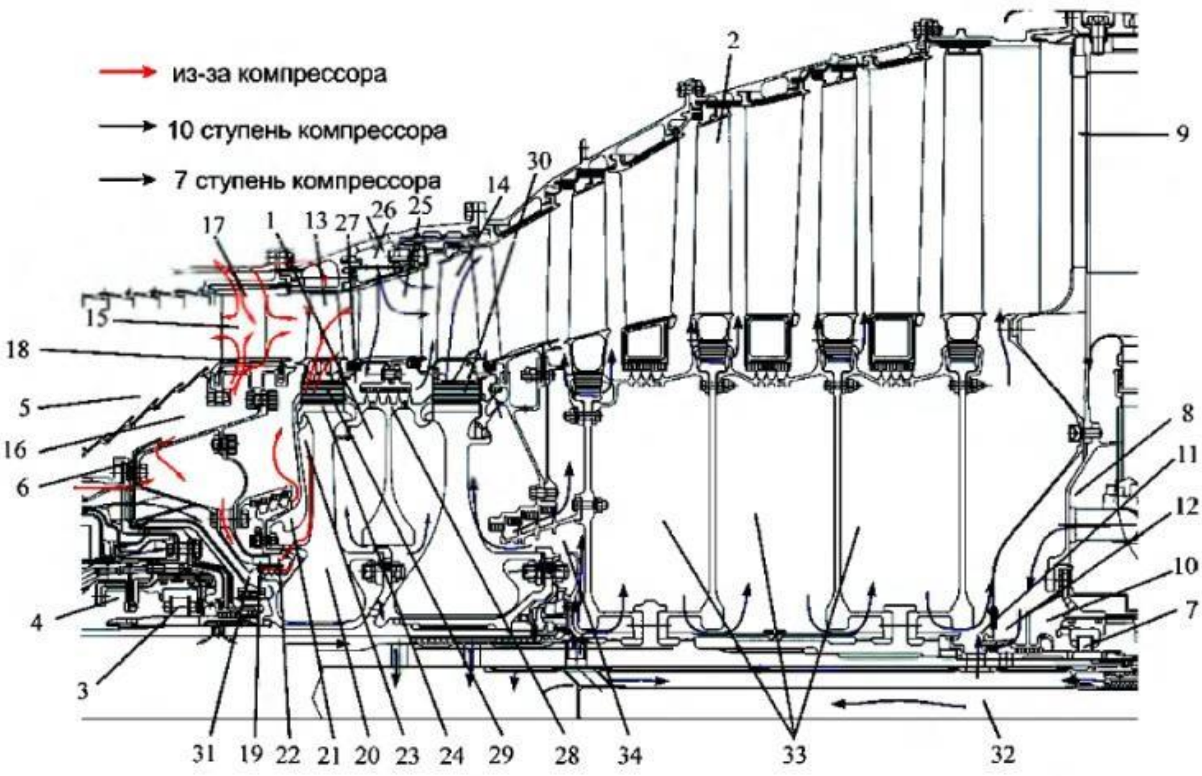
- moderate pre-rotor gas temperature (estimated at 1550 K for RB211-535E4). For Pratt & Whitney, these are the V2500, PW6000, and F119 models. The V2500 is the second largest in the world (after the CFM56). The turbine layout is basically the same as the PW2000 and PW4000. The main characteristics of the turbine design are:

- two-shaft scheme (two-stage HPT + five-stage LPT);

- roller bearing under CS, HPT resistances are connected to the CS housing;

- two-stage highly loaded (expansion degree about 5.0) TVD with sexless working blades;

- high gas temperature before the turbine (1700K).



1 - PDE; 2 - HPF; 3 - PDE bearing; 4 - oil cavity of PDE bearing; 5 - CS; 6 - CS housing; 7 - HPF bearing; 8 - turbine rear support; 9 - rear support stand; 10 - oil cavity of HPF bearing; 11 - air communications; 12 - supercharging cavity; 13 - PDE 1RL;

14 - 2RL PWB; 15 - 1SL; 16 - secondary zone of CS; 17 - upper shelf of 1SL; 18 - lower shelf of 1SL; 19 - spinning apparatus; 20 - first stage disc; 21 - first stage diffuser; 22 - cavity behind spinning apparatus;

23 - cavity under the deflector; 24 - cavity under the blade lock;

25 - 2SL; 26 - annular cavity above 2CA; 27 - cavity under 2CA; 28 - intermediate disk; 29 - lock 1RL; 30 - lock 2RL; 31 - cooling air cavity for rotor; 32 - cold air pipes for HPF; 33 - rotor cavity of ND; 34 - cavity between the rotors of HPF and HPF

Figure 1.3 - PS-90A2 engine turbine [12]

1.2 Turbine design requirements

A turbine is a vane machine in which the kinetic energy and/or internal energy of the working fluid (steam, gas, water) is converted into mechanical work on the shaft. The jet of working fluid acts on the blades attached to the circumference of the rotor and sets them in motion. Thus, turbines allow to turn energy of liquid in our case gas into useful work.

In aircraft engines, the turbine is directly connected by a shaft to the compressor. Thus, the turbine transmits its torque along the shaft, thereby giving it to the compressor and not allowing the engine process to stop.

The aviation turbine itself, simply put, is a structure consisting of a rotor, stator and various auxiliary structural elements. The stator consists of an outer housing, nozzle bodies and rotor bearing housings. The rotor is usually a disk structure in which the disks are connected to the rotor and to each other through various auxiliary elements and mounting methods. [12]

Sometimes the motor is multi-shaft. In this case, there are several consecutive turbines, each driving a different shaft. The high-pressure turbine (the first one after the combustion chamber) always drives the engine compressor, and the subsequent ones can drive both the external load (helicopter or ship propellers, powerful generators, etc.) and the additional stages of the compressor of the engine itself, located before the main one. Separation of the compressor into cascades (low-pressure cascade, high-pressure cascade - respectively LPC and HPC, sometimes placed between them by medium-pressure cascade, MPC, as in the engine NK-32 Tu-160) can partially avoid surges mode voltage.

Specific requirements for turbine design can be formulated as follows:

1. Maximum efficiency.

2. Minimum cooling air consumption. Cooling air flow is really as important for specific engine parameters as turbine efficiency. In addition, increased cooling costs worsen turbine efficiency and make it difficult to obtain such engine environmental performance as low CS emissions. In modern aircraft turbines, the cooling air flow can be as high as 30% of the air flow through the ATC. It is necessary to distinguish between the cooling air flow at the turbine GU and the air

flow entering the flow section behind the section, i.e. at the turbine rotor, from which the direct power extraction takes place. The GU cost (10...12% of the cost through the ATC) can be considered part of the GU in a thermodynamic sense and does not directly affect specific engine parameters, but the gas temperature level at the GU (in Chapter 4) and energy losses in the GU. The temperature difference ranges from 80 to 120 K.

3. Minimum production cost. The share of turbines (HPT and LPT) in the cost of a medium thrust engine (CFM56 and V2500 type) is about 30%. For industrial ground engines based on the aircraft prototype gas generator, in which the fan and LPC are removed and LPT is replaced by CT (type PS-90GP-1, -2, -3), the share of turbines is about 40%.

4. Minimal maintenance costs. The share of turbine maintenance costs in engine maintenance costs (the main part of which is the cost of spare parts and shop repairs) is about 60%. For turbines of short- and medium-range aircraft engines, as well as turbines of so-called "aircraft" ground engines (up to 50 MW), the total cost of maintenance ranges from $50 to $150 per flight hour. For large turbine engines (over 40 tons), thrust support is much higher. The cost of maintenance became independent and very important in the 1990s after the widespread use of the engine maintenance system by the manufacturer on the basis of a fixed fee per flight hour.

Ensuring the service life of the main parts necessary for the competitiveness of the engine. It is the resource of the main parts of the turbine that determines the overhaul life of the entire engine. The service life of blades is most often measured in hours (less often - in cycles). The life of rotor parts (disks, deflectors and shafts) is measured in cycles. The best modern aircraft turbine designs have a blade life of up to 15,000 hours, and service life of rotor parts of the turbines is up to 20,000 cycles. [6]

Existence of a gas temperature reserve upstream of the turbine. The design reserve on the gas temperature before the turbine is a value selected in the design by which the design gas temperatures before the turbine (before the turbine rotor) are increased in thermal and robust calculations.

Designs of modern turbines are based on the highest technologies in the field of materials science. Progress in the field of heat-resistant alloys largely determines the parameters of gas turbines. Typical materials for an aircraft engine turbine are characterized by high specific strength at high temperatures and satisfactory reproducibility of mechanical properties in production. All these materials must be approved by state certification organizations. [7]

A gas turbine design includes four major groups of parts that determine engine reliability: disks, rings and other rotating rotor parts, blades, shafts, and housing parts.

The major design aspects requiring increased attention from the designer are:

1. The location of the roller bearing of the HPT and ways to ensure tightness and protection against overheating of its oil cavity. There are three basic options for locating the bearing and its support: in front of the LPT, between the LPT and the THA, and along the THA. Each solution has its own advantages and disadvantages in terms of cost, reliability, operating experience, as well as providing the necessary conditions for the oil cavity. All of them are discussed below on the example of the present designs.

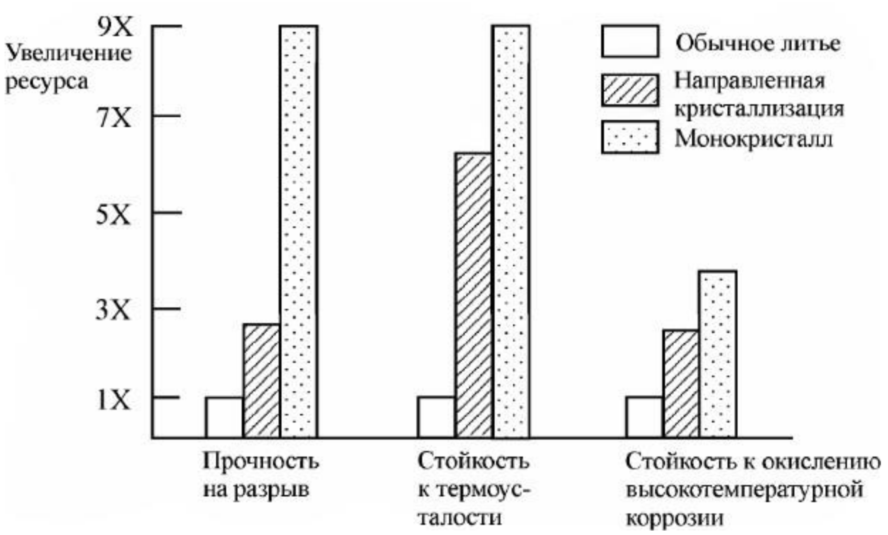
Each turbine rotor must have two supports. One of them can be combined with the compressor support (i.e., this support is a common turbine and compressor shaft). Usually, a ball bearing is installed in the compressor resistance, which excludes axial movements of the rotor and absorbs its axial forces.

A roller bearing is installed in the turbine support. It accepts only radial forces and allows relative axial movements of the rotor and casing. These movements are inevitable due to axial aerodynamic forces and the difference in thermal expansion of the rotor and housing. The magnitude of the axial movements from the cold state in the assembly to the operating state increases with the removal of the ball bearing from the turbine. These displacements from "cold" state to "hot" state must be taken into account in the design.

2. Number of rotors (shafts) - single-shaft, twin-shaft, or strength chains. The number of rotors has an obvious and significant effect on design complexity. An additional rotor means additional bearing resistance and the need to solve the problem of its placement. In modern aircraft engines, the turbine has at least a two-shaft and consists of a HPT and a LPT. In a three-stroke aviation turbine, a MPT appears between the HPT and the LPT, serving to drive a separate compressor stage.

In industrial engines, the third rotor may be free of mechanical connection to the compressor and have a free turbine, simultaneously being a CT to drive the power devices.

3. The structural scheme of the HPT (single-stage or two-stage, the presence of bandage shelves on the working blades of the HPT). In modern aircraft engines, the main role is played by the HPT, which serves as the ATC drive. HPT operates at the highest temperatures and in most cases cools. Cooling and high stresses greatly complicate the HPT design and force the use of expensive high-temperature alloys for blades and disks. A single-stage HPT with the same expansion rate as a two-stage HPT (a typical expansion rate for a modern HPT is 4.0...5.5) should have a circumferential speed in the average diameter 1.4 times greater than for the same UIC load. Increasing the peripheral speed increases the centrifugal forces and thus increases the mass of the structure to provide stresses of acceptable levels. Increasing the weight of rotor parts (especially the disk) also increases rotor inertia and exacerbates the problems of radial clearance control, disk quality control and powder



alloy deflector. Reducing the number of grids by half increases the degree of expansion and velocity levels in each grid. Energy losses and gas loads on all elements of the structure increase. The use of a banded shelf on the working blade of the HPT means increased stress levels, design complexity, and increased cooling air flow to cool the shelf. The shelf gain through increased efficiency must outweigh the increased air consumption and potential durability issues.

Figure 1.4 - Progress provided by alloys with directional crystallization and monocrystalline alloys on reliability and durability of turbine blades [6].

4. Gas temperature level before turbine rotor and efficiency of the cooling system. The level of gas temperature before the rotor and the required efficiency of the cooling system both have a decisive influence on the complexity of the crown blade cooling technologies used and the design of the turbine cooling system. The gas temperature level means the maximum temperature level (for an average new engine) on a hot day (at +30°C). The Redline temperature level will be higher depending on the available gas temperature margin. The design must supply all cooling vanes and cooling elements with the necessary air volume of the lowest possible temperature with a margin of pressure (relative to gas pressure). Air pressure supply is necessary to prevent gas penetration into the cooled parts, its escape into the flowing part. [7]

The turbine vane of a gas turbine engine is a set of static nozzles and working blades. The nozzle and blades are the hottest parts of the turbine. The blades are subjected to strong static stresses when breaking and bending, as well as dynamic stresses. Thermal stresses occur in the cooled blades during turbine transients. Nozzles and working blades work in the gaseous environment of high temperature, containing, besides oxygen, other aggressive substances, including particularly dangerous - vanadium and sulfur. These substances contribute to the development of gas corrosion, destroying the blades.

Therefore, nozzle and blade materials must be not only temperature-resistant, but also temperature-resistant, i.e. resistant to corrosion in atmospheric conditions and in the gas environment at operating temperature. In addition to thermal resistance and heat resistance, gas turbine blade material should have low sensitivity to stress concentration, withstand thermal fatigue, satisfactory machining, have reasonable cost.

Nickel alloys ZhS-6K, ZhS-6F, ZhS-6UVI have been used in the USSR for casting nozzles and working blades since the 1960s. It is recommended to use these alloys up to temperatures of 1050...1100K. Huge progress in turbine parameters and durability of nozzle and blades has been achieved due to introduction of directional crystallization and monocrystalline alloys. The main idea of the alloy with directional crystallization is the elimination of boundaries between grains perpendicular to the direction of centrifugal forces, i.e. eliminating the possibility of creep and fracture at the grain boundaries. The monocrystalline part has no grain boundaries at all, so it has optimal strength characteristics.

The blades obtained by the directed crystallization method increased the strength by 2.5 times, increased the resistance to thermal fatigue by 6 times and the resistance to oxidation and corrosion by 2 times. For monocrystalline blade, strength and thermal fatigue resistance improved by 9 times and oxidation and corrosion resistance by 3.5 times. [7]

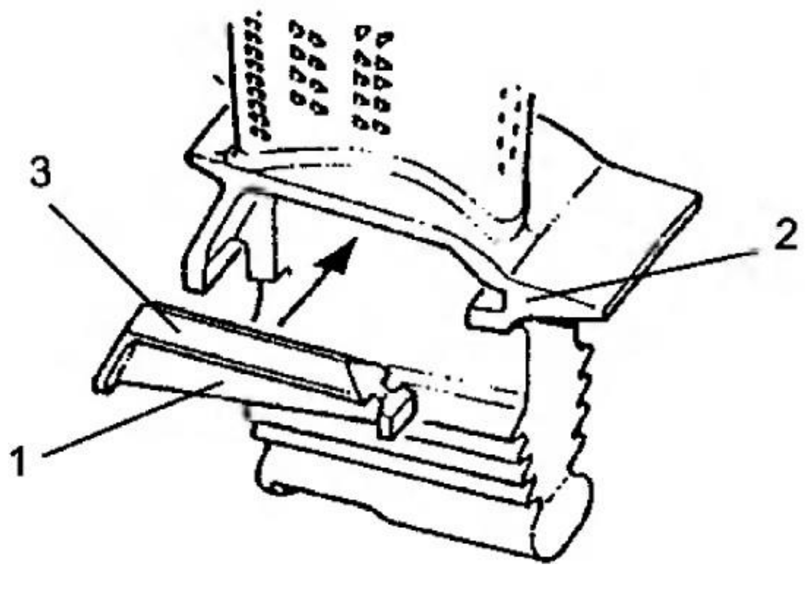
Blades are complex and expensive turbine parts. Like nozzle blades, they are affected by high-temperature gas flow. In addition, unlike nozzle blades, blades are subject to centrifugal forces, rotating at up to 20,000 rpm and peripheral speeds of up to 600 m/s. Stresses from centrifugal forces make the blades more sensitive to vibration loads. The need to withstand all these loads determines the design of the blades.

The blade consists of a profile part, a lock, lower and upper (bandage) flanges, and a leg that connects the profile part and the lower flange to the lock. The profile part, the lock and the lower shelf are the main and obligatory parts of the working blade. The profiled part of the working blade when the blades are installed in the disk forms the crown of the blade, which provides the necessary rotation and expansion of the working fluid flow with minimal losses - that is the main task of the blade.

The working blade lock ensures that the blade is secured in the disk - in the grooves between the protrusions on the disk rim. Connection of the blade to the disk is carried out by means of the fixing connection of the so-called "herringbone" type. Number of teeth in "herringbone" lock may vary from 1...2 (for blades working at low stresses from centrifugal forces), up to 5 - for blades with high stress level. The stress level depends on the circumferential speed (rotor speed and flow diameter) and the mass of the blade itself. In rotor blades of the turbine aircraft the main value is the speed (up to 20000 rpm), for the last stages of turbines of high-power stationary engines of 200...400 mW (at 3000...3600 rpm) is mainly the mass of the blade.

The direction of the groove in the disk may not correspond to the axis of rotation of the turbine. This angle is determined when designing the root section of the blade. If the root section does not fit into a rectangle (which is the bottom flange with a straight groove), the side edges of the bottom flange are angled at the right angle to the turbine's axis of rotation. At the same angle to the axis of rotation and cut a stop groove in the disk. This is less preferable (than a straight groove on the disc), but sometimes it is unavoidable. If the design permits, you can allow some difference in the angles of the lock and the lower shelf of the blade (up to 15°), keeping the straight groove in the disc.

The lower shelf of the blade is necessary to form the inner contour of the turbine flow part. In addition, under the lower shelf you can place flaps - loads, due to their centrifugal force they reduce (damping) through the lower shelf the vibration stresses in the blade (Fig. 1.5).



1 - damper; 2 - lower blade flange; 3 - damper contact surfaces

Figure 1.5 - Damper and its installation in the blade [8]

The foot (located between the lower shelf and the lock) of the blade can be virtually absent. Increasing the length of the stem reduces the diameter of the blade rim and generally reduces the diameter and weight of the blade (increasing the length of the stem may prevent a reduction in the cross section of the blade protrusion and increase the weight of the blade).

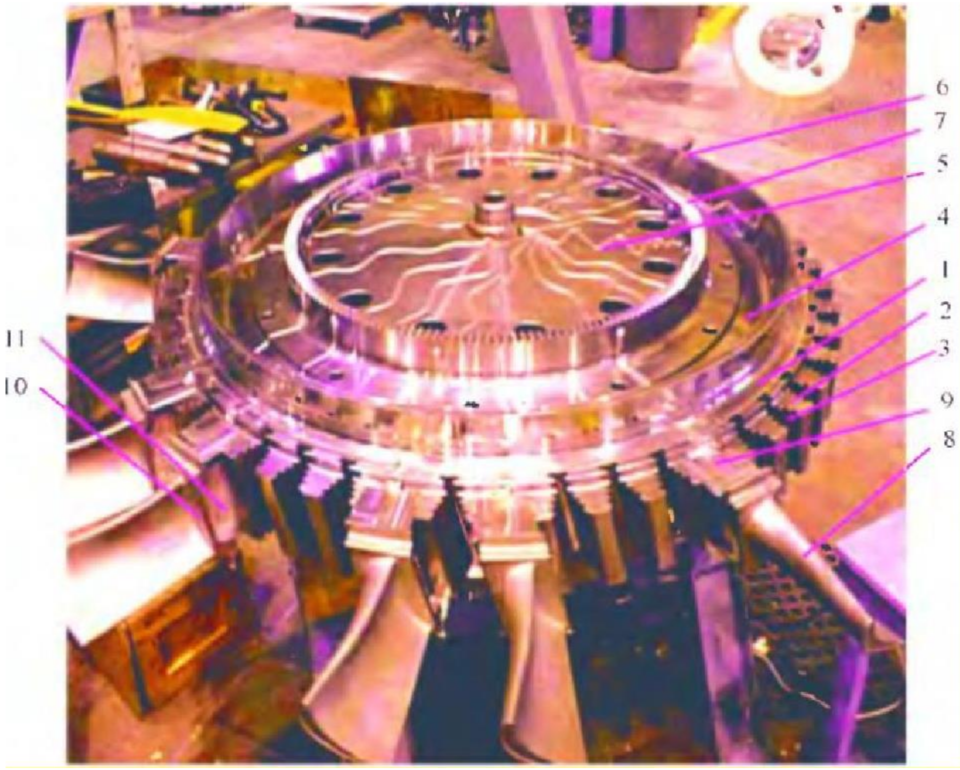
Increasing the length of the stem also reduces the heat flow from the flow and profile portion of the blades into the disc. In addition, by minimizing the level of vibration stresses in the blade, changing its length due to the stem provides some opportunities to control the vibration characteristics of the blade.

The bandage shelf serves to accommodate the sealing of the labyrinth radial gap above the blade (one to three teeth can be placed on the shelf). In addition, the bandage shelf in most cases serves as a means of controlling the vibration characteristics of the blades. When designing the bandage shelf, it is necessary to place its center of gravity in the center of gravity of the upper section of the feather. In this case, the edges of the shelf should have a minimum dangle relative to the profile - to reduce the bending stresses from centrifugal forces.

The turbine disc is the place of blade installation. The disc is used to install the working blades, which create the torque and to transmit this torque from the blades to the shaft.

The connection between the blades and the disc is a tense and critical part of the turbine's design. Nowadays the blades are fastened in the disc in the form of a “herringbone” lock.

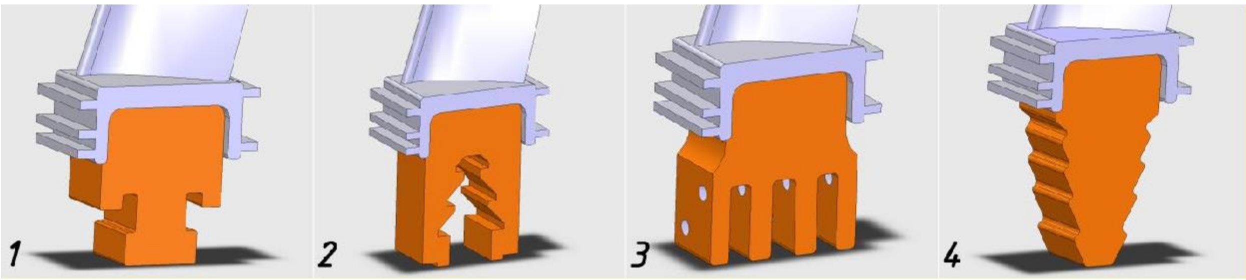
Turbine disks (Fig. 1.6) in a general case have a dinner plate 1 with herringbone-shaped projections 2 which form grooves 3 for blade fastening, a blade 4 and a hub 5, and also flanges 6 for fastening to other disks and to the shaft. Other disks, deflectors 7, labyrinths 8, balancing weights can be attached to the disk flanges.



1 - disc lunch; 2 - disc protrusion; 3 - blade slot; 4 - blade;
5 - hub; 6 - flange; 7 - deflector; 8 - profile part of the working blade; 9 - lock; 10 - bottom flange; 11 - leg

Fig. 1.6 - disk with herringbone type grooves for installation of working blades [10]

When designing discs, you need to ensure a number of requirements. Yes, it is necessary to provide sufficient margin of tensile strength for any possible operating conditions, since the disk failure always leads to catastrophic consequences and cannot be localized within the turbine housing. The next requirement is a minimum probability of disc failure from overheating (i.e. protection from direct contact with high-temperature gas and reliable - with the necessary reserves - operation of the cooling system). The disks of fuel injection engines are usually protected from contact with gas and heat flow along the flowing part both structurally (by baffles and intermediate disks) and by the cooling system. The connection of the blades with the disk is a tense and responsible place in the turbine design. Nowadays the blades are fixed in the turbine disk in the form of so called "herringbone" lock. Constructive form of herringbone lock with three pairs of teeth is shown in Fig. 1.7.



1. T-shaped shank; 2. Mushroom shank (dovetail); 3. Fork-shaped shank; 4. Spruce shank

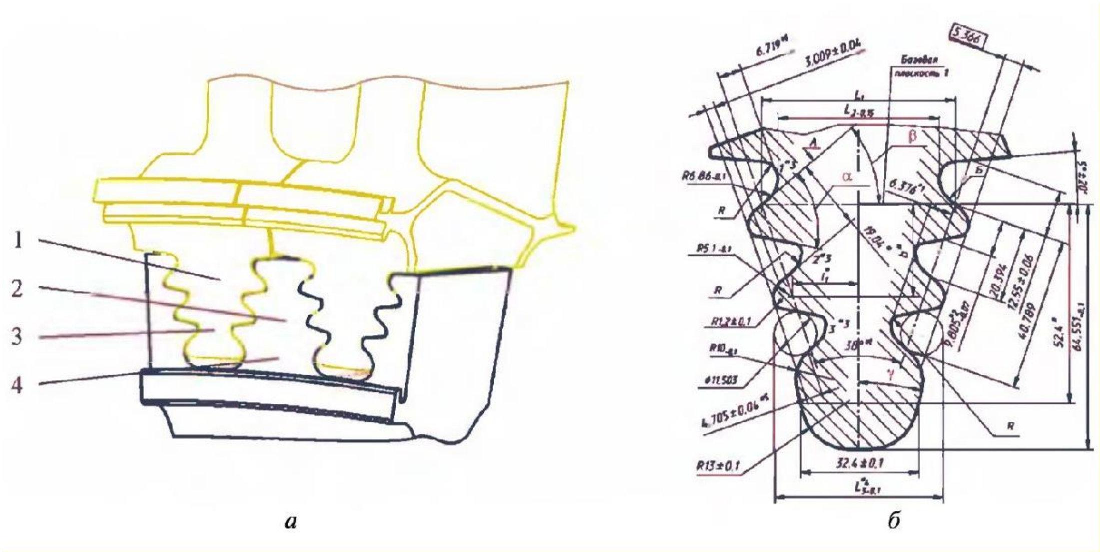
Figure 1.7 - Basic types of blade shanks [10].

The teeth under the action of centrifugal force and bending moments work for shearing, bending and buckling, while the sections at the blade lock hollows and protrusion hollows in the disk work for stretching. The teeth in the blade and disc are manufactured with great accuracy to ensure uniform tooth contact over the entire surface - both in length and width. For example, the tooth pitch tolerance is about 0.008...0.016mm. This ensures uniform loading of all teeth of the joint and avoids dangerous overloading of individual parts of the lock.

"The herringbone lock (Fig. 1.8) has become very useful in practice because of the following advantagesЖ

- the wedge-shaped shape of the locking part of the blade and the peripheral part of the disc ensures a near-equal distribution of stresses (the thickness of the rim and the weight of the disc with the blades are minimal).
- free fit of the blade in the lock (with a gap) eliminates the appearance of thermal stresses in the connection;
- easy replacement of the blades in the wheel when partitioning the assembly or their damage is possible.

When building a "herringbone" lock, the main parameters are the blade pitch on the outer diameter of the disc, the angle of the wedge and the number of teeth.



a - external view of the lock connection;

b - fragment of the blade lock drawing; 1 - blade lock; 2 - disc tab; 3 - bridges between hollows of the blade lock; 4 - bridges between hollows of the disc tab

Fig. 1.8 - Locking connection of vanes with a herringbone disk type [12].

- The angle of the wedge of the lock is chosen, as a rule, in the range of 30 ... 45 degrees, the number of teeth - from two to five. With increasing the number of teeth, the magnitude of the load on each tooth decreases, but the concentration of stresses increases, because with a greater number of teeth the value of the radius of the teeth hollows decreases. Therefore, fewer pairs of teeth are better.

- The most stressed section in the lintel of the disc protrusions is at the base of the protrusions, and the blades are in the first cavity of the lock. To reduce stresses in the elements of the locking connection, the width of the disc rim is made somewhat thicker than the blade.

The grooves in the disk are made by broaching, and the locking part of the blades is made by milling or grinding.

Thus, the design issues that need to be addressed in design, fabrication, and assembly can be noted:

- Secure the blade in the disc so that they absorb uniform stresses during operation.

- Ensuring that the groove is manufactured to the acceptable standards of angular accuracy for further blade attachment

- In addition to its direct function in forming the inner contour of the turbine's flow path, the bottom shelf must provide for the installation of the flaps.

- The flap shelf requires a well-defined position relative to the center of gravity and is a means of controlling the vibration characteristics of the blade.

Both centrifugal and gas forces act on the blades - gas flow forces arising from the pressure difference in front of and behind the blade crown, as well as from the pressure difference between the chute and the rear part. These forces result in bends acting together with the tension of centrifugal forces. However, for operating conditions, when designing a blade profile, it is possible to almost completely compensate bending stresses due to centrifugal forces. In this case, centers of gravity of design sections are placed on the line inclined to radial direction - so that the moment arising under the action of centrifugal forces is directed directly opposite to the action of the total moment of gas forces.

Shift of centers of gravity from radial direction is determined during strength calculations. Full compensation of gas forces is possible only at certain operating conditions (mode), because the value of gas forces changes depending on absolute parameters of turbine mode.

Practical evidence shows that cases of blade failures due to lack of static strength are very rare and most often result from any manufacturing deviations. At the same time, adjustment of blades to reduce vibration stresses is a common phenomenon. This is due to the fact that the accuracy of analytical predictions of average temperature and average stress level (necessary to determine the static strength of the blade) is much higher than in determining the resonant velocity and vibration stress. Moreover, even the necessary experimental work to determine the vibration stresses in the blades does not give full confidence in the form of oscillations, their frequency range, level and place of occurrence.

The fact is that for the correct conduct of such experiment (choice of methodology, type of sensors and their location), the preliminary modeling of the vibration characteristics of blades in a high-level engineering package (ANSYS, NASTRAN) is required. However, the accuracy of analytical modeling (construction of grids, setting the correct boundary conditions) cannot be guaranteed in advance.

Consequently, of such great importance in the design of blades is to ensure an acceptable level of vibration and the necessary reserves from resonant frequencies. Sources of vibration can be combustion chamber nozzles, injector blades, intermediate bearing struts, etc. Since it is often impossible to accurately predict the source of the dangerous frequency, it is advisable to exclude the possibility of oscillations through the blade design. In general, it is necessary to design.

However, eliminating the possibility of vibration through blade design is not always possible, in which case two means of controlling (with limited capability) the level of vibration in the blades are used.

1. The first of them is the use of bandage shelf on the upper end of the working blade, which has meshing with the shelves of neighboring blades with special contact surfaces. The contact surfaces of the bandage shelves of the blades are connected in the wheel by a fastening "tension" and damp the vibrations of the blades due to friction.

2. The second means of controlling the level of vibration stresses are flaps 7, which are mounted under the bottom flange 2 of the blade. For blades, which do not have a bandage shelf (or have a bandage shelf of "aerodynamic" purpose - without contact faces), they are used. The contact surfaces 3 of the flaps are pressed (due to centrifugal forces) to the lower surfaces of the flanges 2 and due to friction reduce (dampen) the vibrations of the blades. Dampers influence mainly the level of vibration stress and relatively weakly - on the natural frequency of vibration of the blade.

Modern turbine blades are almost exclusively cast on melted models. HPT blades can be manufactured using a special casting and cooling process - to produce a cast with directional crystallization or monocrystallization. In working blade castings, only the "herringbone" lock connection surfaces, the connecting surfaces of the gangplank shelf and (in film-cooled blades) the perforations are machined.

When designing blades, special attention should be paid to production capabilities. This applies to the selection of the profile thickness in each section, the diameter of the inlet opening and especially the outlet edge. For chilled blades, minimum wall thickness and the ability to manufacture inner cavity and exit edge cooling channels are important. This is the only way to ensure a cost-effective percentage of appropriate castings.

1.3 Physical processes occurring in the turbine blade during GTE operation

In the process of the previously mentioned interaction with the blades there is a transformation of the kinetic energy of the flow into mechanical energy, rotating the shaft of the engine. This transformation in the axial turbine can occur in two ways:

1. Without changing the pressure, and therefore the value of the relative velocity of the flow (only its direction changes significantly - the rotation of the flow) in the turbine stage;
2. With a drop in pressure, an increase in relative flow rate and some change in its direction in degrees.

Turbines operating by the first method are called active. The flow of gas actively (pulse) affects the blades as a result of changing its direction during their flow. In the second method are reactive turbines. Here, in addition to the pulse effect, the flow also affects the blades indirectly (simply put) by the reactive force, which increases the power of the turbine. Additional reactive effect is achieved by special profiling of the blades.

In modern aviation gas turbine engines only jet axial turbines are used.

The peculiarity of the turbine's blade apparatus is that its operating conditions are almost always close to critical. In modern engines, the gas temperature (after flashover) at the combustion chamber outlet can reach 1650 °C (with a tendency to increase), so special precautions are required for normal turbine operation at such high thermal loads:

1. Application of heat-resistant materials both metal alloys and (further) special composite and ceramic materials used for manufacturing the most loaded parts of the turbine - nozzle and working blades and disks. The most loaded of them are the blades.

The heat-resistant alloys can be based on aluminum, titanium, iron, copper, cobalt and nickel. The most widely used in aircraft engines are heat-resistant nickel alloys made from working and blades of nozzles, turbine rotor disks, combustion chamber parts, etc. Depending on the manufacturing process, nickel heat resistant alloys can be cast, deformed and powders. The most heat-resistant cast nickel-based alloys are capable of operating at temperatures of 1050-1100 ° C for hundreds and thousands of hours under high static and dynamic loads.

In modern heat-resistant and high-temperature alloys various alloying elements are added to 16 names in order to obtain maximum high-temperature characteristics.

These include, for example, chromium, manganese, cobalt, tungsten, aluminum, titanium, tantalum, bismuth and even rhenium or ruthenium. Particularly promising in this regard is rhenium (used in Russia), which is now used instead of carbides, but it is extremely expensive and its reserves are small. The use of niobium silicide is also considered promising.

2. The use of a thermal barrier layer on the blades. Thermal barrier coating (TVS) significantly reduces the amount of heat flow in the blade housing (thermal barrier functions) and protects it from gas corrosion (thermal resistant functions).

Conclusions to Chapter 1

The following results were obtained as a result of the analysis of GTE and its turbine operation peculiarities:

Component parts and basic characteristics of typical GTE designs have been analyzed and described.

The main requirements to the design and peculiarities of GTE turbine operation have been described, namely: maximum efficiency; minimum cooling air flow rate; minimum production cost and maintenance cost; provision of the service life of the main parts necessary for engine competitiveness.

Physical processes arising in the turbine blade during GTE operation have been studied. Without changing pressure, and hence the value of the relative flow velocity (tangibly changing only its direction - the turn of the flow) in the degree of the turbine. With pressure drop, growth of relative flow velocity and some change of its direction in degree.

SECTION 2.
METHOD OF GTR OPERATION CONDITIONS AND ITS TECHNICAL CONDITION EVALUATION OF DAMAGE ACCUMULATION IN THE TURBINE PARTS

2.1 Investigation of damage accumulation in the turbine parts, manifested during aviation GTE operation

On modern turbojet engines, the level of gas temperature before the turbine is almost equal to the level of gas temperature before the turbine of military turbochargers.

The maximum gas temperature before the HPT rotor reaches 1700...1850 K. Engines for short medium-range aircraft (CFM56, V2500) have much lower temperature levels. Working and nozzle turbine blades work in direct contact with high-temperature gas, and the allowable temperature of the blade alloys is lower than the working gas temperatures before each corona by 200...500 K. The greatest difficulty is to ensure the reliability of the blades, especially in high-pressure turbines. They, together with nozzle blades, are subject to thermal fatigue, vibration, gas corrosion and erosion, exposure to gas loads. In addition, the blades are exposed to centrifugal forces. Taking all this into account, the average metal temperature of the blades must not exceed 900 ... 1000°C, and the maximum level - 1100°C, in order to ensure the reliable operation. The level of acceptable operating temperatures depends directly on the characteristics of the blade material applied.

Some rotor and stator turbine parts are also directly exposed to gas: the housing, part of the disc rim, labyrinths and other less stressed parts. To ensure their reliable operation for this service life, they use:

- special heat-resistant, heat-resistant and corrosion-resistant alloys capable of resisting sulfide-oxide corrosion;
- manufacturing of blades by the directional crystallization method or from monocrystals;
- coating to improve the thermal resistance of the material (e.g., alumina);
- metallic multicomponent coatings to improve the corrosion resistance of the material, e.g. four-component coatings (nickel-chromium-aluminum- yttrium);
- thermal protection coatings made of ceramic materials with low thermal conductivity - to reduce heat flow into the blade metal;
- different schemes of air (for industrial turbines, sometimes even steam) cooling. [11]

An optimal turbine design in terms of engine life cycle cost implies an optimal combination of all the above-mentioned basic efficiency methods. The use of expensive heat-resistant alloys increases the material cost, but reduces the need for cooling. Using a more sophisticated and efficient turbine blade cooling system increases the cost of the turbine, but allows the use of less expensive materials.

Developing the optimal cooling system involves a consistent search for a reasonable compromise at all stages of the project.

Modern engines are operated according to their technical condition until their service life expires or any malfunction occurs.

During bench testing and in operation, turbine malfunctions can be detected by engine diagnostics (e.g. temperature rise across the turbine) or during maintenance, using special tools and control methods. Such tools are optical endoscopes (for inspection of parts of the flow part), instruments for ultrasonic and current vortex control of deflectors and disks. In order to facilitate diagnostics in turbine housings the hatches are made to provide access to control elements.

The turbine, as the most thermal engine unit, is also the most frequent source of malfunctions, which leads to repairs and limits the service life. For example, with an average life of 11,000 hours, two out of every three times the RChU4000 family (all models) is sent in for repair because of a problem (burnouts and cracks) with the HPT blades.

A CPM56-3 engine with an average service life (for second and subsequent repairs) of about 10,000 hours is sent for repair due to reaching the cyclic life limit (35%) due to exceeding the temperature limit for the turbine (31%) and due to TVD blade durability problems (lCA burns and cracks, 1RL axial and radial cracks). Fig. 3-204 shows the average operating time of the new CF6-80C2 engines (GE Aircraft Engines). [10]

Figure 2.1 - Average MTBF of new CF6-80C2 engines on removal from wing - by turbine blade defects and by all defects taken together on "removal from wing" due to turbine blade failure, by all defects [12]

Figure 2.1 shows that it is turbine blade defects that overwhelmingly cause engines to be taken out of service and sent in for repairs.

While problems with GTE blades account for the bulk of engine problems, the vast majority of defects in the blades themselves are burnouts, thermo-fatigue cracks, high-temperature oxidation, corrosion and erosion. Fig. 2.2 and 2.3 show typical defects of the first stage GTE nozzle blades.

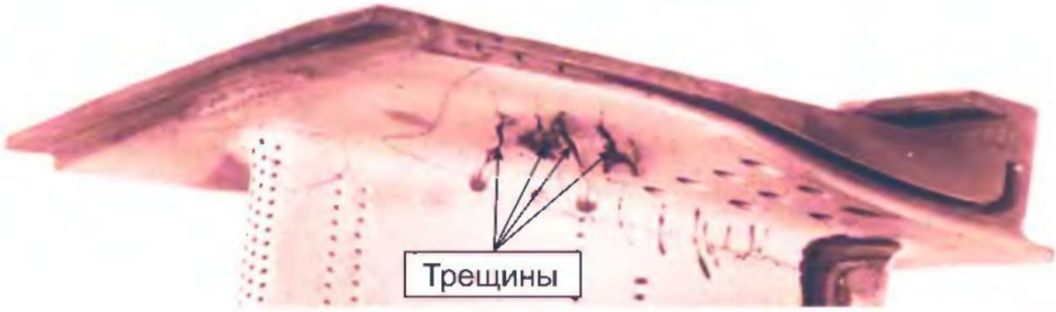


Figure 2.2 - Cracks on the profile and the shelf of the nozzle blade of the first stage of GTE

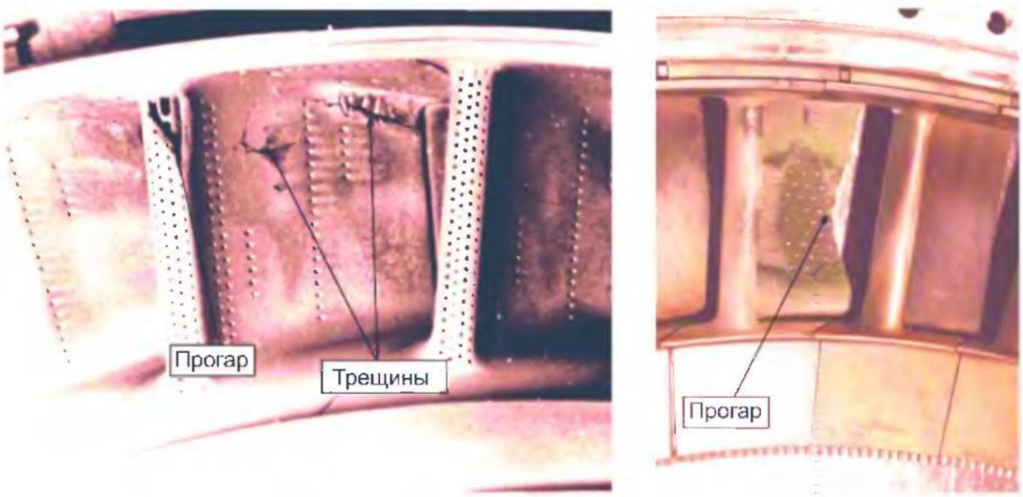


Figure 2.3 - Cracks and burnouts on the profile of GTE nozzle blades [10]

Local increase in the cooling air flow (mainly due to additional holes for cooling the film) is common in the finishing stage or in the first stages of operation, when it is necessary to adjust the blade cooling system according to the results of real tests.

The use of monocrystalline materials for HPT blades for civilian engines has been the general rule since the second half of the 1980s, when Pratt & Whitney monocrystalline blades were first used in the PW2000 (certified in 1983) and PW4000 (certified in 1980). Monocrystalline nozzle blades were first used by GE Aircraft Engines in the CFM56-5A TVD (certified 1988) and significantly increased blade durability over the CFM56-3Cl.

Also, characteristic defects of the operating blade are corrosion and erosion of the blade face (on all blades), damage to the inlet metal edge and the top of the chute.

Periodically varying forces act on the turbine blades during the engine operation. If the frequency of excitatory vibrations coincides with the frequency of vibration of the blades, a resonance occurs. In this case, the stresses in the blades increase dramatically, and the blades may break at the handle or at the lock part.

The periodicity of forces, causing the forced oscillations of the blades, is explained by the heterogeneity of the flow in the flowing part, associated with a finite number of stators, having aerodynamic and thermal traces. These are nozzles and fire tubes of the combustion chamber, nozzle blades, stators in the intermediate or rear support, rectifier blades on the turbine, etc. Blade fracture is boring. A failure of one blade usually leads to damage and destruction of other blades in the impeller and even blades of subsequent stages (Fig. 2.4).

The possibility of resonance is usually analyzed and eliminated in the design by:

- changing the number of nozzles, racks, number of nozzle blades, and other elements of the flow section that can disturb the vibration of the blades;
- optimization of sections and moments of inertia of the main sections of the profile part and the blade leg both by changing the profile shape and by introducing special ribs and other elements into the inner cavity of the blade;
- use of flaps under the lower flanges of the blades;
- use of banded flanges with zigzag edges with tensioning fixation on impeller contact surfaces.

Figure 2.4 - Breakage of the upper half of the profile part of the working blade of the second stage of the turbine and indirect damage to the bandage flanges of other blades [2]

The following methods are used to ensure the tensile strength of blades in production:

- increasing the tiring strength of the herringbone paddle lock by treating (strengthening) it with microspheres;
- control of natural vibration frequency of bladeless blades;
- control of tiring strength of blades during their manufacture.

However, the reliability of analytical methods is not yet high enough to completely eliminate the possibility of resonance or to ensure a safe level of vibration. Therefore, when finishing the turbine, the strain gauge (measurement of vibration stresses on the blades) is carried out on the basis of a preliminary calculated analysis of the most dangerous places. In case of detection of an unacceptably high level of stresses or the need to reduce the risk of their increase, the choice of the most reliable, cost-effective and time-saving measures to combat the potential defect is made.

Reducing resonant stresses without a costly and long-term redesign of the operating blade is possible by increasing the level of damping or by reducing the level of excitatory forces.

Any turbine (or degree of it) converts the energy of the flow entering it into mechanical work. However, in a real unit, this process can have different efficiencies. Part of the provided energy is necessarily spent "in vain", i.e. it turns into losses, which must be taken into account and measures taken to minimize them to improve the efficiency of the turbine, i.e. to increase its efficiency.

Losses consist of hydraulic and initial velocity losses. Hydraulic losses include the profile and the end. The profile is actually friction losses as the gas, having a certain viscosity, interacts with the surfaces of the turbine. Usually such losses in the impeller are about 2-3% and in the nozzle apparatus about 3-4%. Measures to reduce losses consist in "improvement" of the flow part by calculation and experiment, as well as correct calculation of velocity triangles for flow in degree of turbine, more precisely selection of the most favorable circumferential velocity U at a given speed C1. These actions are usually characterized by the parameter U/C1. The circumferential velocity in the average radius in a turbojet engine is 270 - 370 m/s.

Hydraulic parameters of the turbine cascade flow part take into account such parameter as adiabatic efficiency. Sometimes it is also called blade efficiency because it takes into account the friction loss in the blade degree (CA and RL).

There is another efficiency for a turbine that characterizes it as a unit of power, that is, the degree to which energy is used to create work on the shaft. This is what is called energy efficiency (or efficiency). It is equal to the ratio of work on the shaft to the temperature difference. This efficiency takes into account losses with the initial velocity. Typically, they are about 10-12% turbocharger (in modern turbojet engines C0=100-180 m/s, C1=500-600 m/s, C2=200-360 m/s). For turbines of modern gas turbine engines, the adiabatic efficiency value is about 0.9 to 0.92 for uncooled turbines. If the turbine is cooled, this efficiency can be 3-4% lower. The energy efficiency is typically 0.78 to 0.83. It is less than adiabatic in terms of initial velocity losses.

As for the final losses, these are the so-called "overflow losses". The flow section cannot be completely isolated from the rest of the motor due to the rotating units combined with the stationary units (housing + rotor). Consequently, a gas from high pressure areas tends to flow into the low-pressure area. In particular, for example, from the area in front of the working blade to the area along it through the radial gap between the blade and the turbine casing.

Such gas is not involved in the process of transformation of the flow energy into mechanical one, because it does not interact with blades in this regard, i.e. there are final losses (or losses in radial clearance). They are about 2-3% and have a negative impact on both adiabatic and energy efficiency, reduce the efficiency of gas turbine engine and very significantly.

For example, it is known that increasing radial clearance from 1 mm to 5 mm in a 1 m turbine can increase specific engine fuel consumption by more than 10%.

Radial clearances degrade engine efficiency. Therefore, both passive and active methods of radial clearance control have been developed to minimize them, which will be described below.

It is clear that it is impossible to get rid of radial clearance completely, but they try to minimize it. This is quite difficult, since an aircraft turbine is a heavily loaded unit. Accurately accounting for all the factors that affect the gap size is quite difficult.

Often the operating modes of the engine change, which means the magnitude of deformation of the blades, the disks on which they are mounted, the turbine housings as a result of changes in temperature, pressure and centrifugal forces.

Here it is necessary to take into account the magnitude of residual deformations during prolonged engine operation. In addition, evolutionary changes in the aircraft affect the deformation of the rotor, which also changes the clearance size.

Usually, the clearance is evaluated after stopping the warmed up engine. In this case, the thin outer casing cools faster than the massive discs and shaft and, decreasing in diameter, touches the blades. Sometimes the size of the radial clearance is simply chosen within 1.5-3% of the length of the feather blade. Here it is necessary to take into account the magnitude of residual deformations during prolonged motor operation. In addition, the evolution performed by the aircraft affects the deformation of the rotor, which also changes the size of the gaps.

Usually the clearance is evaluated after stopping the warmed up engine. In this case, the thin outer casing cools faster than the massive discs and shaft and, decreasing in diameter, touches the blades. Sometimes the radial clearance size is simply chosen within 1.5-3% of the feather blade length.

To avoid damage to the blades if they touch the turbine housing, it often has special inserts made of a material softer than the material of the blades (e.g. metal-ceramic). In addition, non-contact seals are used. These are usually labyrinth and honeycomb seals.

In this case, the blades are bandaged at the ends of the pen and seals or wedges (for honeycombs) are already placed on the shelves of the bandages. In compacted honeycomb, due to the thickness of the honeycomb walls, the contact area is very small (10 times smaller than a conventional labyrinth), so the assembly of the node is carried out without a gap. After work, the gap size is supplied about 0.2 mm.

Similar gap sealing methods are used to reduce gas leakage from the flow part (e.g. into the inter-disk space).

Axial clearances are also adjusted in the turbine. For example, there is usually a gap between the output edges of the CA and the inlet RL in the range of 0.1-0.4 of the RL chord per average blade radius. The smaller this gap, the less flow energy loss for the CA (friction and velocity field equalization for the CA). But at the same time, the vibration of the RL increases because of the alternating hitting from the areas behind the CA blade cases to the inter-blade area.

The so-called passive radial clearance control methods were discussed above. Many GTEs developed (and under development) since the late 1980s have so-called "active radial clearance systems" (SAURZ - active method). These are automatic systems, and the essence of their operation is to control the thermal inertia of the aircraft turbine housing (stator).

The rotor and the stator (outer casing) of a turbine differ from each other by material and "massiveness". Therefore, they expand differently in transient modes. For example, when the engine switches from low to high, high temperature, the thin-walled housing (rather massive rotor with discs) heats up and expands, increasing the radial clearance between itself and the blades. In addition, there is the change in pressure in the path and the evolution of the airplane.

To avoid this, an automatic system (usually a FADEC-type main controller) arranges to supply cooling air to the turbine housing in the necessary quantities. Thus, the heating of the housing is stabilized within the necessary limits, which means that the linear expansion of the housing and correspondingly the size of the radial clearances is changed. All this saves fuel, which is very important for civil aviation. Effective SAURZ systems are used in low-pressure turbines on jet engines such as the GE90, CFM56, Trent 900, PS-90A and several others.

Even rarer, but excellent enough to synchronize the heating rate of the rotor and stator, forced blowing of the turbine discs is used. Such systems are used on CF6-80 and PW4000 engines.

2.2 Methods of protection of GTE turbine blades during operation and methods of increasing their durability

Two types of coatings are used for turbine blades.

1. Metallic. Metallic coatings (consisting of several components) protect the blade base metal from oxidation and corrosion and are the main means of blade efficiency in aggressive working environment.

2. Ceramic. Thermal protection ceramic (TZP) coatings provide reduced heat flux into the part due to the reduced thermal conductivity of the protective layer. The reduction of the highest alloy temperature ranges from 30 to 90°C (depending on the coating thickness, its parameters and the temperature gradient between the gas and the blade wall). [5]

Oxidation and corrosion processes are particularly strong in turbines. In marine engines, protective coatings are practically the only means of ensuring durability of turbine blades. According to GE's experience, the first tests of the LM2500 engine (the land-based version of the TF39 aircraft engine) in marine applications showed an unprecedented deterioration of performance due to salt corrosion of the turbine blades. The problem was solved only by the urgent development of a new coating. The solution was a thermal protective ceramic coating.

The widespread use of coatings for HPT blades began in the 1990s on high-temperature engines with high thrust (PW4084, GE90), and then advances in technology began to justify their use on more mainstream engines with lower temperatures (PW2000, V2500, PW600).

The use of TZP significantly reduces cooling air clogging, increasing turbine efficiency and blade durability.

A typical example of a thermal barrier coating or TBC (Thermal Barrier Coating) is shown in Figure 2.5 below. The designation in the figure is: TGO (thermally grown oxide) - thermally grown oxide; Lining - basic blade material; Coat Bond - transition layer.

The figure (microphoto) shows the heat shield layer on the high-pressure turbine blade of a modern turbojet engine. TVS now includes such metals as: nickel, chromium, aluminum, yttrium and others. Experimental work is underway on the use of ceramic coatings based on zirconium oxide as a stabilizer (VIAM development).

However, without cooling, almost all heat-resistant metallic materials can only withstand temperatures of approximately 1050-1070°C.

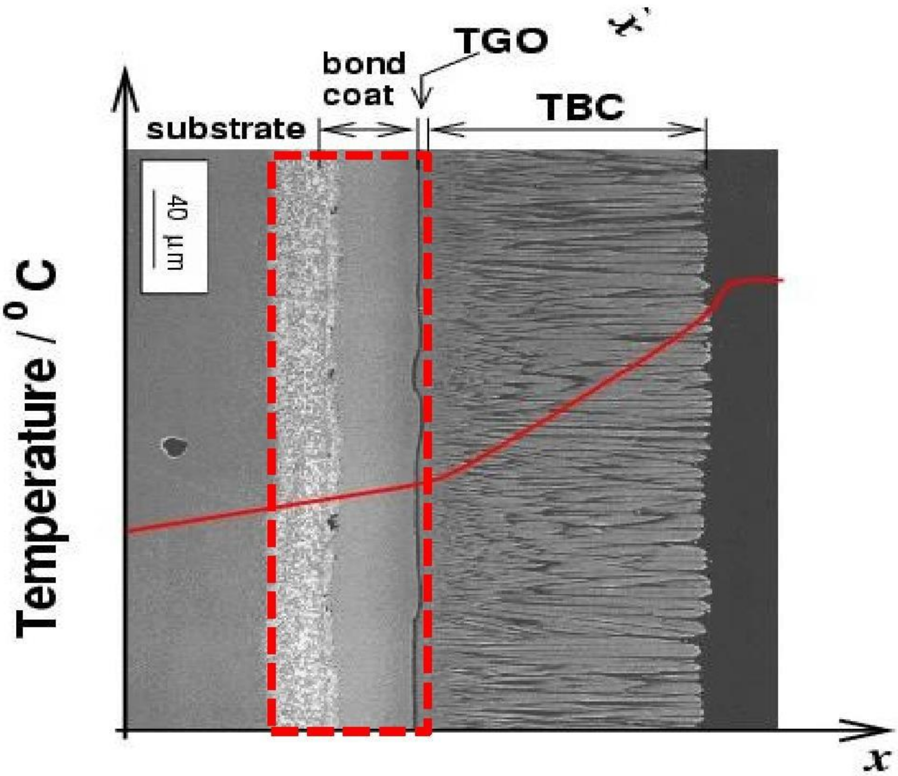


Figure 2.5 - Example of thermal protection coating. The nature of the temperature change of the blade crossing [8]

Therefore, the second, widely used measure is the use of various cooling systems of blades and other structural elements of aircraft turbines. It is still impossible to do without cooling in modern gas turbine engines, despite the use of new high-temperature heat-resistant alloys and special methods of manufacturing elements.

In terms of cooling systems, a distinction is made between open and closed systems.

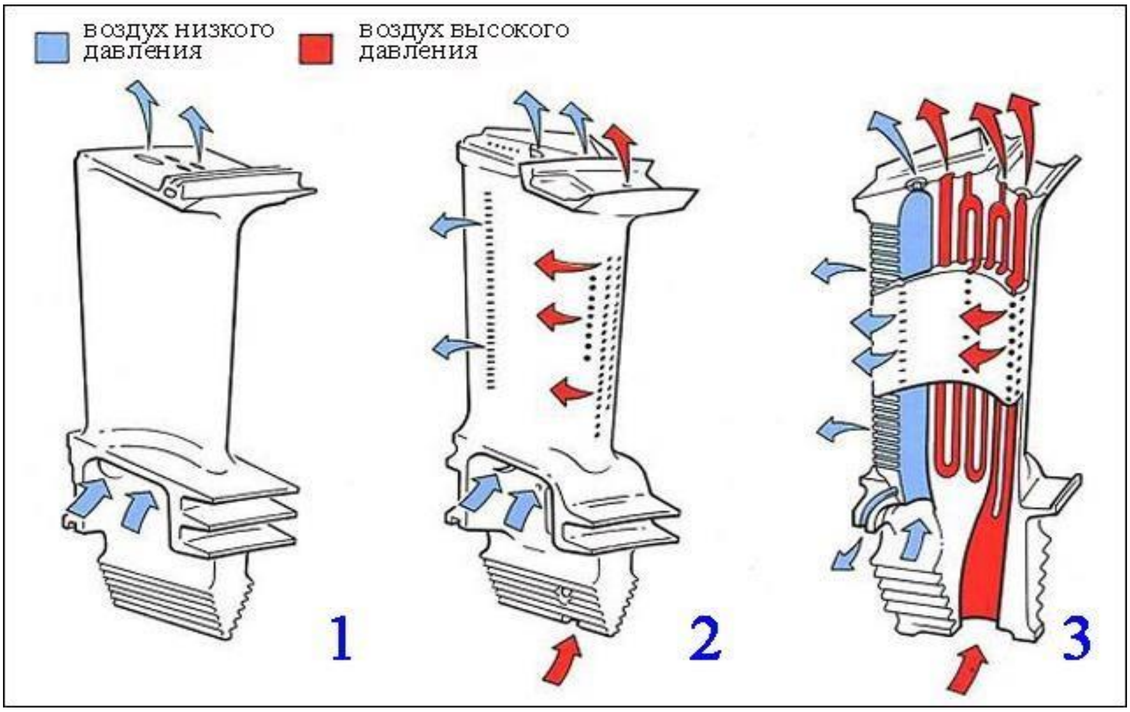
Closed systems can use forced circulation of a liquid coolant in the vane-cooler system or use the "thermosyphon effect" principle, in which the coolant moves under the action of gravitational forces, when warm layers are displacing colder ones. Sodium or an alloy of sodium and potassium, for example, can be used as the coolant here.

However, closed systems are not used in aviation practice due to a large number of complex problems and are in the experimental research stage.

Open cooling systems, on the other hand, are used more frequently. The refrigerant in such systems is air, which is usually supplied at different pressures through different compressor stages inside the turbine blades. Depending on the maximum gas temperature at which it is appropriate to use these systems, they can be divided into three types: convective, convective-film (or barrier) and porous.

During convective cooling, air is supplied to the vane through special channels and, flushing the most heated areas inside it, exits into the flow in the zone with less pressure. Different schemes of air flow organization in the blades can be used, depending on the shape of channels for it: longitudinal, transverse or loop-shaped (mixed or complicated).

The simplest scheme with longitudinal channels along the feather. Here the air outlet is usually organized in the upper part of the blade through the bandage shelf. In this scheme, there is a rather large temperature irregularity along the feather of the blade - 150-250˚C, which negatively affects the strength properties of the blade. The scheme is used on engines with a gas temperature T 1130ºC.



1 - convective, 2 - convective-foam, 3 - convective-foam with complicated loop channels in the blade

Fig. 2.6 - Examples of possible methods of blade cooling [2]

The next method of convective cooling involves a special deflector (thin-walled shell - inserted inside the pen) inside the pen, which helps to supply cooling air first to the hottest zones. The deflector forms a kind of nozzle that blows air into the front of the blade. This produces a jet cooling of the hottest part. The air then washes out the other surfaces through the longitudinal narrow holes in the handle.

In this scheme, the temperature irregularity is much lower; in addition, the deflector itself, which is inserted into the vane under tension along several centering transverse grooves, serves as a dampener and dampens vane oscillations due to its elasticity. This scheme is used at the maximum gas temperature T ≈ 1230°C.

The following so-called semi-circuit convective cooling scheme makes it possible to achieve a relatively uniform temperature field in the blade. This is achieved by experimentally selecting the location of the different fins and pins that direct the air flow inside the blade body. This arrangement allows for a maximum gas temperature of up to 1330C.

The nozzle blades are convectively cooled similarly to the working blades. They are usually made with two cavities with additional fins and pins to intensify the cooling process. Higher-pressure air is supplied to the front cavity at the leading edge than to the rear cavity (through different compressor stages) and vented to different areas of the path to maintain the minimum required pressure differential to provide the necessary air velocity to the cooling ducts.

The second method is convective film cooling. Convective film cooling is used at even higher gas temperatures, T 1380°C. In this method, part of the cooling air is released through special holes in the blade on its outer surface, thus creating a kind of barrier film that protects the blade from contact with the flow. hot gas. This method is used for both working and blade nozzles.

The third method is porous cooling. In this method, the power core of the blade with longitudinal channels is covered with a special honeycomb material, which ensures uniform and metered release of coolant to the entire surface of the blade, is flushed with a gas flow.

This is still a promising method, in mass practice the use of AMD is not applied because of technological and operational difficulties, difficulties with selection of porous material and high probability of rather rapid clogging of pores. They also have a relatively low strength, high production cost. However, when these problems are solved, the probably possible gas temperature with this type of cooling can reach 1650 °C. [10]

A large amount of air is removed from the compressor to cool the aircraft turbine. On different engines - up to 15-20%. This significantly increases the losses taken into account in the thermogasodynamic calculations of the engine. Some engines have systems that reduce the air supply for cooling (or even close it) at low engine speeds, which has a positive effect on efficiency.

2.3 Methods of damageability assessment of GTE parts

When forecasting the service life of critical structural elements of gas turbine engines at the design stage and when developing methods of calculation evaluation of equivalent operating time and residual service life during operation, the concept of material damageability is widely used [7, 33]. Damageability is understood as a process of loss of bearing capacity or reduction of service life of a material which occurs as a result of irreversible changes in the material caused by mechanical, thermal and chemical effects and finally leading to its destruction. The measure of damageability is damageability - a quantitative characteristic reflecting the part of the serviceability lost by the SE due to operation under the action of loads.

From the point of view of process physics the manifestation of damageability are irreversible changes of material structure (shear processes inside grains, twin formation, crushing of grains, loosening and void formation processes, change of strengthening phases, deformation on the grain boundaries and formation of submicroscopic breaks) aging and depletion, damage of surface layer due to the action of operating factors [34]. Structural changes form damages which cause violation of material continuity (macrocracks, form changes, warping, etc.), due to the nature of the acting load (static, long-term static, multicyclic mechanical, thermocyclic action).

The existing models of damage accumulation are mainly based on phenomenological notions. The available experimental data point to the complexity of physical processes at macro- and microlevel in the material structure and to their insufficiently stable correlation with the processes of force and temperature action. Damageability of the material, accumulated by a certain moment of the material's development, is manifested in the integral form. In particular, the values of basic characteristics of short-term and long-term strength of resistance to high and low cycle fatigue, creep and plasticity decrease, and also the physical characteristics of material change (specific electric resistance, acoustic emission, etc.), which can serve as an indirect measure of damageability.

The damageability of a material is also evaluated using parameters that describe the behavior of the material on the basis of continuum mechanics methods.

The degree of damage to the material of the part is usually estimated by the relative value P, which varies within 0+1, whose value in the initial, undamaged state is zero (P=0), and at the moment of reaching the limit state - unity (P=l).

The main parameters characterizing the degree of impact of static, multi-cycle mechanical, low-cycle isothermal and non-isothermal loads on damageability are temperature *T*, level of static stresses $\sigma_a$duration of mode *t*, amplitude of cyclic loads σ_a, number of cycles *N*, amplitude of elastic-plastic $\varepsilon^{(k)}$, plastic deformation per cycle, one-sided accumulated strain ${\varepsilon_p}^{(k)}$, maximum temperature of heating cycles T_{max}, duration of stress and temperature cycles.

The conditional principle of summing up the linear damage capacity in the form of relative durability is widely used to estimate the fraction of damage capacity from the action of a long-term static load:

$$\Pi_c = \int_0^t \frac{dt}{\tau_{mp}(\sigma_{cm}, T)},$$

Where $\tau(\cdot)$is the long-term strength characteristics.

Similarly, the fatigue damage fraction is estimated by the value

$$\Pi_ц = \int_1^N \frac{dN}{N(\sigma_a, T)},$$

Where *N(·)* - fatigue resistance characteristics.

In case of multicomponent loading, the criteria of force, time, strain and energy types, based on both linear and nonlinear method of damage summation, are used to calculate the damageability of a part.

Criteria of force type are based on construction of durability model, which is a function of acting static $\sigma_{ст}$, variable mechanical $\sigma_а$, thermocyclic $\sigma_т$and other types of loads, as well as temperature T

$$\tau_k = \tau_k(\sigma_{cm}, \sigma_a, \sigma_m, T, \ldots)$$

Where $\tau_k(\cdot)$ is the generalized strength characteristics of the material.

In this case, the damage capacity will be determined by the total effect of interaction of all the marked types of loads for the whole variety of working conditions of a given part.

$$\Pi \quad = \int_0^t \frac{dt}{\tau_k(\sigma_{cm}, \sigma_a, \sigma_m, T, \dots)},$$

Models of multicomponent loading, based on the construction of surfaces of ultimate stresses by individual types of loading, represent in the case of three-component loading a plane, elliptic paraboloid or other types of approximating surfaces or straight line, ellipse and other curves of the second order at two-component loading.

In the work of Kuznetsov M.D. i Tseptlin V.I. [35] suggest a force criterion in the case of a combined action of cyclic loads with stress amplitude σ_a and repeated-static loads σ_m. The generalized connection diagram between σ_a and σ_m at a given service life is described by a parabolic dependence satisfactorily agreeing with the experimental data

$$\sigma_a = \sigma_{-1}\left(1 - \left(\frac{\sigma_m}{\sigma_{B\tau}}\right)^a\right),$$

where $\sigma(-1)$, $\sigma_{B\tau}$ are endurance limits for high-frequency and low-frequency fatigue; a is a coefficient depending on material and loading conditions ($a \approx 2$).

Criteria of mixed type are usually of purely phenomenological nature, where the relative time or number of cycles to failure is used as the degree of damageability. They include a number of criteria for determining the durability of parts operating under low-frequency mechanical and static loading at elevated temperatures, in particular, the criteria of Manson [36], Getsov B.G. [37], Dulnev R.A. [39].

Currently, deformation and energy criteria based on the inelasticity of metals are widely used to describe the processes of damage accumulation and fracture of materials under cyclic mechanical and thermal loads.

Under uniaxial symmetric loading, an empirical power relationship between the inelastic strain per cycle and the number of cycles to failure was proposed based on experimental studies by Manson and Coffin.

$$\Delta\varepsilon_{\text{пл}} N_p^k = C$$

where $\Delta\varepsilon_{\text{пл}}$ is the plastic strain per cycle; N_p is the number of cycles to failure; k, C are constant parameters of the equation. This dependence is based on the assumption of additivity of strain accumulation in the material.

Other deformation-type criteria proposed by Manson, S.V. Serensen, R.M. Schneiderovich, A.P. Gustnkov, N.A. Makhutov, L.B. Getsov, R.A. Dulnev, and others cited in [38, 39].

Analysis of energy criteria proposed in works by Troshchen V.T. [43], Pisarenko F.C., Mozharovsky M.S. [42], Muratov L.V. the previously considered authors show that these criteria, as well as deformation criteria, have their advantages and disadvantages, as well as a limited area of application. In particular, the energy criteria are more justified from a physical point of view, but the mathematical expressions for their description are rather complicated. The noted incidents require the search for more appropriate characteristics of damageability of materials for engineering calculations.

In case of multicomponent loading, when the failure can occur due to several causes, in case of using a linear model the ultimate total damageability can be written in the form of the sum of relative damages:

$$\sum_{i=1}^{K_1}\frac{\tau_i}{\tau_{pi}}+\sum_{i=1}^{K_2}\frac{n_i}{N_{pi}}+\sum_{i=1}^{K_3}\frac{l_i}{l_{pi}}=1$$

where the first term takes into account the damageability from static loading, the second term - from cyclic loading, the third term - from cyclic creep.

In the general case of the action of S simultaneous factors, both independent and factors affecting each other, the limit state can be determined from the following conditions:

$\sum_{j=1}^{S}\Pi_j = 1$ - linear damage summation hypothesis;

$\max\limits_{S} \left(\Pi_j\right) = 1$- for independent factors;

$\sum_{j=1}^{S} g\left(\Pi_j\right) = 1$ - nonlinear damage summation hypothesis.

This method of damage estimation is especially convenient in engineering practice, since in this case the strength characteristics under one-component load are used as the limit characteristics.

A more reasonable and accurate method of ultimate damage assessment under complex thermal cycling loading is based on the analysis of elastic-strain kinetics of structural elements. Approximate approaches to such calculation using cyclic deformation diagrams were considered in [5]. However, the calculation of cyclic elasto-plastic deformations in dynamics is rather laborious and is possible only when the stabilization of the deformation process occurs after a small number of load cycles.

Kiyalbaev D.A. and Chudnovskiy A.I. [44] use the notion of the entropic nature of material failure. Ivanova V.S. [16, 34] considers the process of fracture as a result of breaking of interatomic bonds due to absorption by the crystal lattice of the ultimate energy. Bolotin V.V. [7], analyzing the available experience, developed mathematical models of the generalized theory of damage accumulation, describing dependences of durability on loading history, as well as multi-stage theories, where the fracture process is represented as two stages, for example, preparatory and crack development and deepening stages.

Let us analyze some peculiarities of the process in damage accumulation. Let's consider generalization of damage models and related criteria of limiting state. The distribution of stresses, deformations and temperatures in parts is, as a rule, non-uniform. Therefore, the damage processes occurring at different points of the part are not uniform, which determines the dependence of damageability not only on time, but also on coordinates. This leads to continuum models of material damage, examples of which are models of plasticity and creep theories or damage fields characterizing the density of microcracks, dislocations, slip lines or other crystal lattice defects. In practice, the use of such damage measures is evaluated in the most stressed zones.

Parts of GTE work in conditions, when the occurrence of limiting state is possible due to several reasons, both independent and dependent on each other (formation of cracks, inadmissible deformations, limiting abrasion or corrosion wear). In this case, it is advisable to introduce several measures of damageability, characterizing different processes with different types of limit state. Such interrelated processes running in parallel can be described by a vector-function P, and their limit state can be represented as a certain norm, the typical expression for which has the form:

$$|\Pi| = \left(\sum_{i=1}^{k} \Pi_i^{\nu_i} \right)^{\nu_0}$$

where $v_i(i = \overline{0, k})$ - are the parameters characterizing the relationship between the components; *k* is the number of factors.

The process of damageability of parts material consists of several successive stages, characterized by their structural components: the laws of initiation, development and consolidation of damage. The rate of damage accumulation at each stage of the process depends on both the level of stresses and temperatures, and the order of loading. This is reflected in the fact that the failure time will depend on the load prehistory.

Multistage models can describe the effects of "far" load prehistory. Mathematical models based on convolution-type integral equations are used to account for the "near" history, in particular the effects of aftereffects and lagging that appear, for example, in the creep of metals.

Thus, when building generalized models of damage processes, their dependence on time and coordinates and other peculiarities of damageability of material, multistage, multicomponent nature should be taken into account.

Consequently, all the suggested hypotheses of damage summation have a limited scope of application. Each hypothesis is based on assumptions, which cannot be considered to be quite true in the general case. There is obviously not enough statistical data to reliably prove one or the other theory. Materials of different nature differ in their damage accumulation patterns. In this regard, of all theories of summation, the theories based on the linear hypothesis or its modifications and on the separated accounting of the influence of different load components of structural elements on the accumulated damage are the simplest and most convenient for practical use. Linear models are also convenient to investigate from a probabilistic point of view.

Conclusions to chapter 2

In this section the method of estimation of influence of GTE operation conditions and technical condition of damage accumulation in its turbine parts has been studied.

Accumulation of damage in turbine parts, which manifests itself during operation of aviation GTEs, has been investigated.

Methods of GTE turbine blades protection during operation and methods of their durability improvement have been analyzed.

Methods of damageability estimation of GTE parts are described. It is determined that the construction of generalized models of damage processes, their dependence on time and coordinates and other peculiarities of damageability of material should take into account multistage, multicomponent nature.

CHAPTER 3.
ASSESSMENT OF THE INFLUENCE OF GTE OPERATION CONDITIONS AND ITS TECHNICAL CONDITION ON THE DAMAGE ACCUMULATION IN ITS TURBINE PARTS

3.1 General information about the PS-90A turbofan engine

PS-90A is a turbofan engine with a maximum thrust of 16,000 kgf. According to the scheme it is a double-circuit turbojet engine with mixing of flows (inner and outer circuits) (Fig.3.1).

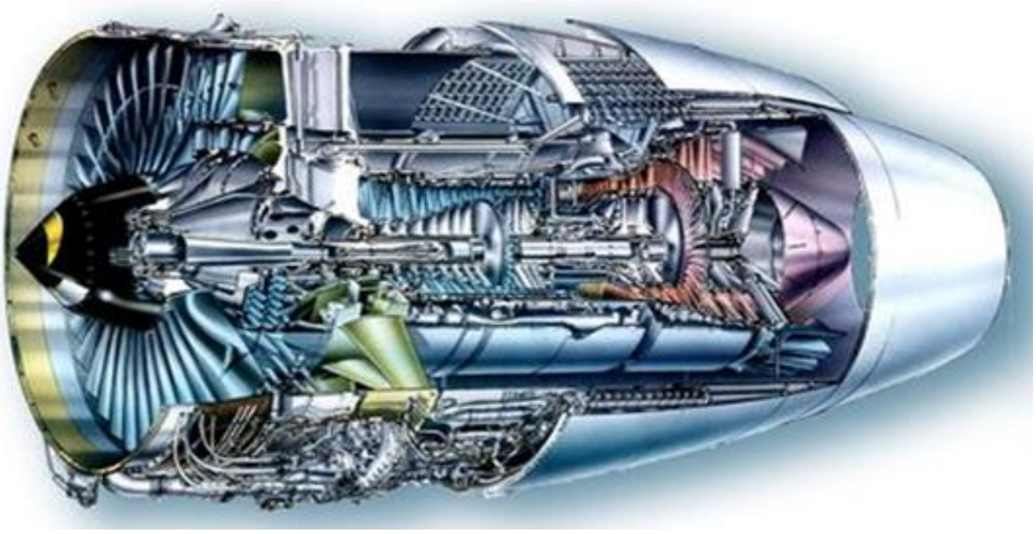


Fig. 3.1 - Scheme of the PS-90A engine [12]

It is designed to be installed on passenger aircraft of Il-96 (Il-96-300, Il-96-400), Tu-204 (Tu-204-100, Tu-204-300, Tu-214) and family of Il-76 (Il-76MD-90, Il-76TD-90, A-50EI, Il-76MF). The last development of aircraft designer P.A. Solovyov, after whom it is named: PS - Pavlo Solovyov. It is manufactured by OJSC “Perm Engine Plant”.

The PS-90 engine was certified in 1992 and has been in service ever since. The engine is operated according to the technical condition within the assigned resources (cycles) of the main parts. The maximum operating time without removal from the wing is 12198 hours (Z.M. 3949043102040), which is twice as long as the overhaul interval of the previous generation engines, and the lead engine has worked 35503 hours (Z.M. 3949042001017).

The PS-90A is certified to meet ICAO emissions standards in 2008 and ensures all aircraft installed meet ICAO standards for aircraft noise.

The PS-90A engine module is shown in Fig. 3.2. The modular design in combination with a developed diagnostics system and testability allows the engine to be operated according to its technical condition. All the modules, except for the basic one, can be replaced in operation:

1 - fan impeller; 2 - straightener; 3 - basic;

4 - reversing device; 5 - low-pressure turbine; 6 - nozzle

7 - rear support; 8 - high pressure turbine; 9 - drive box;

10 - low pressure compressor; 11 - inlet guide vanes of a low pressure compressor

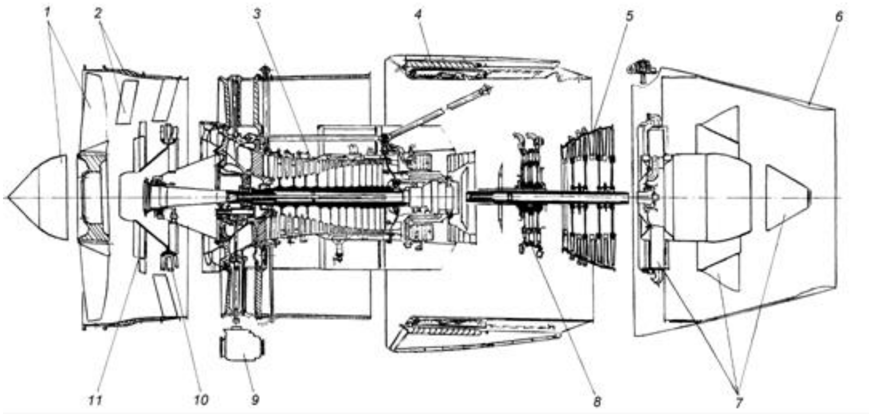


Figure 3.2 - Engine module PS-90A [10]

At PS-90A engine it is possible to replace some components of the modules, as well as the most damaged parts, such as flame tubes and nozzles of KS, grids and flaps of RF, etc. In addition, it is possible to replace all installed units and equipment, as well as to perform visual and optical inspection of the entire flow part. To reduce the noise level, sound-absorbing structures are installed in the engine case.

3.2 Mathematical model of the engine operating process

3.2.1 Thermodynamic calculation of the turbofan engine

Input data:
- engine type - TFE;
- thrust F=70000 N;
- degree of two circuits m=5,5;
- degree of air pressure increase in the compressor $\pi^*K\sum=30$;
- degree of air pressure increase in coefficient of efficiency LPS $\pi^*_{LPS}=3$;
- degree of air pressure increase in the HPS $\pi^*_{HPS}=13$;
- degree of air pressure increase in the fan $\pi^*_{ВЛ\ II}=1,5$;
- gas temperature before the turbine $T^*_Г=1600$ K;
- design conditions $H=0$, $V=0$.

Determination of working medium parameters before the engine (section H-H).

Using ISA(MCA) tables, we find pressure P_H and temperature T_H for a given height and determine the parameters of the retarded flow P^*_H and T^*_H by formulas:

1. determining the parameters of the working body before the engine (section H-H)

$$T^*_H = T_H + \frac{V^2}{2\frac{K}{K-1}R} = 288{,}15K$$;

$$P^*_H = P_H \left(\frac{T^*_H}{T_H}\right)^{\frac{K}{K-1}}$$; $$P^*_H = 101325\,Па$$

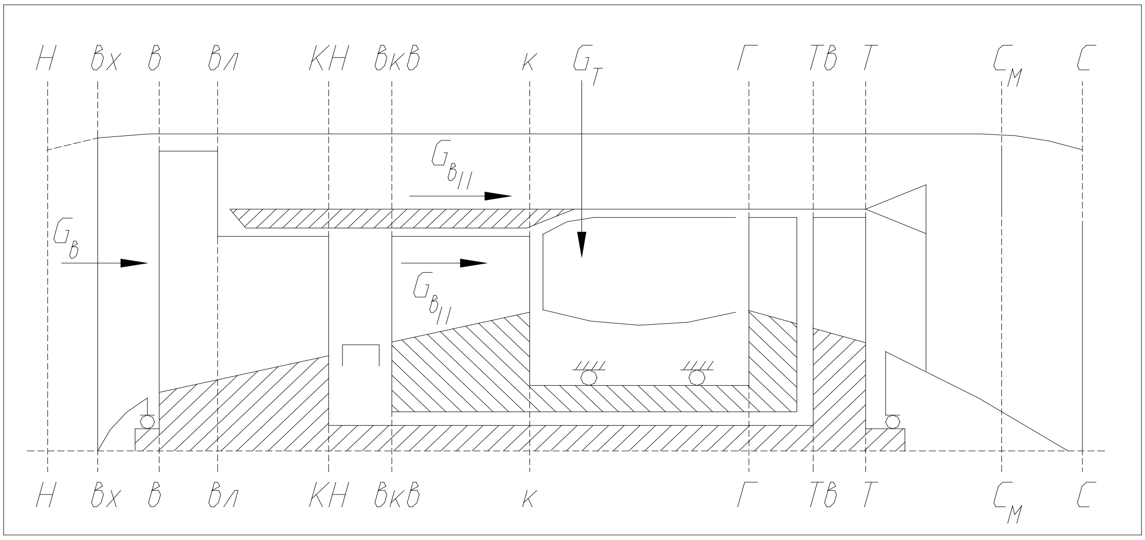


Fig. 3.3. Scheme of turbofan engine with separate exhaust [12].

Determination of working body parameters upstream the engine (section H-H).

According to the MCA tables, we find the pressure P_H and temperature T_H for a given altitude and determine the parameters of the retarded flow P^*_H and T^*_H by the formulas:

1. determining the parameters of the working body before the engine (section H-H)

$$T^*_H = T_H + \frac{V^2}{2\frac{K}{K-1}R} = 288{,}15K\,;$$

$$P^*_H = P_H\left(\frac{T^*_H}{T_H}\right)^{\frac{K}{K-1}};\; P^*_H = 101325\,Па$$

2.Determination of air parameters at fan inlet (section B-B). According to energy equation we find:

$$T^*_B = T^*_H = 288{,}15 \text{ K}.$$

The coefficient of recovery of full pressure in the inlet device σ_{BX} is taken equal to 1 and we find P^*_B by the formula:

$$P^*_B = P^*_H\,\sigma_{BX}\,;\; P^*_B = 101325\cdot 1 = 101325\,Па$$

2. Determination of working body parameters by the fan in the external circuit (section Vl-Vl (Вл-Вл)). We select the fan efficiency $\eta^*_{ВЛ\,II}$ of 0.88 and find the work of air compression in the external circuit according to the equation:

$$L_{ВЛ\,II} = \frac{K}{K-1} RT^*_{В}\left(\pi^{*\frac{K-1}{K}}_{ВЛ\,II} - 1\right)\frac{1}{\eta^*_{ВЛ\,II}};$$

$$L_{ВЛ\,II} = \frac{1,4}{1,4-1} 287,3 \cdot 288,15\left(1,5^{0,286} - 1\right)\frac{1}{0,85} = 44314,59\ \frac{Дж}{кг}.$$

The fan pressure $P^*_{ВЛ\,II}$ and temperature $T^*_{ВЛ\,II}$ are determined by the formulas:

$$P^*_{ВЛ\,II} = P^*_{В}\,\pi^*_{ВЛ\,II};\ P^*_{ВЛ\,II} = 101325 \cdot 1,5 = 151987,5\,Па;$$

$$T_{ВЛII} = T_{В} + \frac{K-1}{K} \cdot \frac{L_{ВЛII}}{R};\ T_{ВЛII} = 288,15 + \frac{1,4-1}{1,4} \cdot \frac{44314,59}{287,3} = 332,26K$$

4. Determining the air parameters by compressor (K-K section).

We determine the efficiency of the HPS and the LPS using an approximate formula, specifying the compressor stage efficiency $\eta^*_{CT} = 0,9$:

$$\eta^*_{КНД} = \frac{\pi^{*\frac{K-1}{K}}_{КНД} - 1}{\pi^{*\frac{K-1}{K\eta^*_{CT}}}_{КНД} - 1};\ \eta^*_{КВД} = \frac{\pi^{*\frac{K-1}{K}}_{КВД} - 1}{\pi^{*\frac{K-1}{K\eta^*_{CT}}}_{КВД} - 1};$$

$$\eta^*_{КНД} = \frac{3^{\frac{1,4-1}{1,4}} - 1}{3^{\frac{1,4-1}{1,4\cdot 0,9}} - 1} = 0,88;\ \eta^*_{КВД} = \frac{10^{\frac{1,4-1}{1,4}} - 1}{10^{\frac{1,4-1}{1,4\cdot 0,9}} - 1} = 0,86.$$

The effective work of compressing air in the compressors is found from equation:

$$L_{КНД} = \frac{K}{K-1} RT^*_{В}\left(\pi^{*\frac{K-1}{K}}_{КНД} - 1\right)\frac{1}{\eta^*_{КНД}};$$

$$L_{КНД} = 3,5 \cdot 287,3 \cdot 288,15\left(3^{0,286} - 1\right)\frac{1}{0,88} = 121458\,\frac{Дж}{кг};$$

$$T^*_{КНД} = T^*_{В} + \frac{L_{КНД}}{\frac{K}{K-1}R};\ T^*_{КНД} = 288,15 + \frac{121458}{3,5 \cdot 287,3} = 409,06\,K;$$

$$L_{КВД} = \frac{K}{K-1} RT^*_{КНД}\left(\pi^{*\frac{K-1}{K}}_{КВД} - 1\right)\frac{1}{\eta^*_{КВД}};$$

$$L_{КВД} = 3,5 \cdot 287,3 \cdot 409,06\left(13^{0,286} - 1\right)\frac{1}{0,86} = 516203,052\,\frac{Дж}{кг};$$

$$L_{K} = L_{КНД} + L_{КВД};\ L_{K} = 516203,052 + 121458 = 637661,052\,\frac{Дж}{кг}.$$

Compressor temperature T^*_{K} and pressure P^*_{K} are determined by formulas:

$$P^*_{K} = P^*_{В}\,\pi^*_{K\Sigma};\ P^*_{K} = 101325 \cdot 30 = 3039750\,Па;$$

$$T^*_{K} = T^*_{КНД} + \frac{k-1}{k} \cdot \frac{L_{КВД}}{R};\ T^*_{K} = 409,06 + \frac{1,4-1}{1,4} \cdot \frac{516203,052}{287,3} = 922,93K.$$

5. Determination of the operating body parameters at the combustion chamber outlet (section G-G (Г-Г)). Setting the coefficient of recovery of full pressure in the combustion chamber $\sigma_{KC} = 0,97$, we find the pressure before the turbine:

$P^*_Г = P^*_К\,\sigma_{КС}$; $P^*_Г = 3039750 \cdot 0,97 = 2948557,5\,Па$.

The gas temperature before the turbine $T^*_Г$ is given in the initial data:

$T^*_Г = 1600К$

The average heat capacity of the gas in the combustion chamber is calculated by the formula:

$$C_{РСР} = 878 + 0,208\left(T^*_Г + 0,48T^*_К\right);$$

$$CPCP = 878 + 0,208\left(1600 + 0,48 \cdot 922,93\right) = 1302,95\frac{Дж}{кг \cdot К}.$$

Setting the combustion completeness factor $\eta^*_T = 0,98$, and taking the value of fuel thermal conductivity $H_U = 42500\frac{кДж}{кг}$, we find the relative fuel flow rate:

$$q_T = \frac{C_{РСР}\left(T^*_Г - T^*_К\right)}{\eta_Г H_U};\; q_T = \frac{1302,95\left(1600 - 922,93\right)}{0,98 \cdot 42,5 \cdot 10^6} = 0.0212.$$

6. Determination of gas parameters along the turbine (T-T section).

Taking into account the gas temperature behind the combustion chamber relative to the amount of air taken for cooling the turbine parts $q_{ОХЛ} = 0,1$, and taking the value of mechanical efficiency $\eta_M = 0,99$, we determine the effective operation of all turbine stages of the turbine of the turbofan engine by the equation:

$$L_T = \frac{mL_{ВЛ\,II} + L_К}{\left(1 + q_T\right)\left(1 - q_{ОХЛ}\right)\eta_M};\; L_T = \frac{5,5 \cdot 44314,59 + 637661,052}{\left(1 + 0,0212\right)\left(1 - 0,1\right)0,99} = 968561,87\frac{Дж}{кг}.$$

Taking the turbine efficiency $\eta^*_T = 0,9$, $K_Г = 1,333$, $R_Г = 288$, we calculate the turbine temperature $T^*_Г$ and pressure P^*_T by the formula:

$$T^*_T = T^*_Г - \frac{K_Г - 1}{K} \cdot \frac{L_T}{R_Г};\; T^*_T = 1600 - \frac{1,333 - 1}{1,333}\frac{968561,87}{288} = 759,23К;$$

$$P^*_T = P^*_Г\left[1 - \frac{T^*_Г - T^*_T}{T^*_Г\eta^*_T}\right]^{\frac{K_Г}{K_Г - 1}};$$

$$P^*_T = 2948557,5\left[1 - \frac{1600 - 759,23}{1600 \cdot 0,9}\right]^4 = 91750,146\,Па.$$

7. Determination of the working fluid parameters at the outlet of the mixing chamber (S_M-S_M (C_M-C_M) section).

The temperature of the working body at the outlet of the mixing chamber is calculated by equation:

$$T^*_{СМ} = \frac{mC_{PB}T^*_{ВЛII} + C_{PГ}T^*_T}{\left(1 + m\right)C_{PСМ}},$$

where C_{PB} - is the average heat capacity in the interval from T^*_T to $T^*_{СМ}$;

$C_{PСМ}$ - average heat capacity of the mixture, which is calculated by the formula:

$$C_{PСМ} = \frac{mC_{PB} + C_{PГ}}{1 + m}.$$

In approximate calculations we can assume:

$C_{PB} = 1010 \frac{Дж}{кг \cdot К}$; $C_{PГ} = 1100 \frac{Дж}{кг \cdot К}$;

$$C_{P\,CM} = \frac{5,5 \cdot 1010 + 1100}{1 + 5,5} = 1023,846 \frac{Дж}{кг \cdot К}.$$

Then $T^*_{CM} = \frac{5,5 \cdot 1010 \cdot 332,26 + 1100 \cdot 759,23}{(1+5,5)1023,846} = 402,83К$.

The value of the adiabatic index $К_{CM}$ is between $К_Г$ and $К$ and can be calculated by the formula:

$$К_{CM} = \frac{C_{P\,CM}}{C_{V\,CM}} = \frac{C_{P\,CM}}{C_{P\,CM} - R_{CM}}.$$

The gas constant of the mixture R_{CM} has a value between

$R_Г = 288 \frac{Дж}{кг \cdot К}$ and

$R_Г = 288 \frac{Дж}{кг \cdot К}$, which is determined by the formula:

$$R_{CM} = \frac{mR_B + R_Г}{1+m}; \; R_{CM} = \frac{5,5 \cdot 287,3 + 288}{1+5,5} = 287,408.$$

Then $К_{CM} = \frac{1023,846}{1023,846 - 287,408} = 1,39$.

The pressure of the operating body at the outlet of the mixing chamber is determined by the approximate formula:

$$P^*_{CM} = \sigma_{CM} \left(P^*_T \cdot P^{*m}_{BCM}\right)^{\frac{1}{1+m}},$$

where σ_{CM} - the coefficient of recovery of total pressure at mixing of streams;

$$\sigma_{CM} = 0,98$$

$$P_{BCM} = P^*_{вл} \cdot \sigma_{II} = 151987,5 \cdot 0,99 = 150467,13.$$

Then $P^*_{CM} = 0,98\left(136358,14 \cdot 150467,13^{5,5}\right)^{\frac{1}{1+5,5}} = 107849,36Па$.

8. Determination of the operating body parameters at the jet nozzle outlet (section C-C).

The critical pressure drop is calculated by the equation:

$$\pi_{C\,KP} = \left(\frac{К_{CM}+1}{2}\right)^{\frac{К_{CM}}{К_{CM}-1}}; \; \pi_{C\,KP} = \left(\frac{1,39+1}{2}\right)^{\frac{1,39}{1,39-1}} = 1,89.$$

For expansion $\frac{P^*_{CM}}{P_H} = 1,064 < \pi_{C\,KP}$ in the jet nozzle, the total and the parameters of the working body are determined by the formulas:

$P_C = P_H = 101325Па$;

$$C_C = \varphi_C \sqrt{2\frac{К_{CM}}{К_{CM}-1} R_{CM} T^*_{CM} \left[1 - \left(\frac{P_H}{P^*_{CM}}\right)^{\frac{К_{CM}-1}{К_{CM}}}\right]};$$

$\phi_C = 0{,}98$;

$$C_C = 0{,}98\sqrt{2\frac{1{,}39}{1{,}39-1}287{,}408\cdot 402{,}83\left[1-\left(\frac{101325}{107849}\right)^{\frac{1{,}39-1}{1{,}39}}\right]} = 125{,}9;$$

$$T_C = T^*_{CM} - \frac{C^2{}_C}{2\frac{K_{CM}}{K_{CM}-1}R_{CM}};\ T_C = 402{,}83 - \frac{125{,}9^2}{2\frac{1{,}39}{1{,}39-1}287{,}408} = 395{,}09K.$$

9. Determination of basic specific parameters of turbojet two-circuit engines with flow mixing.

Specific thrust of turbojet engines with flow mixing is calculated by the equation

$$P_{УД\Sigma} = \left(1+\frac{g_T}{1+m}\right)C_C - V + \frac{R_{CM}T_C}{C_C}\left(1-\frac{P_H}{P_C}\right).$$

Since $P_C = P_H$, and V=0, so

$$P_{УД\Sigma} = \left(1+\frac{g_T}{1+m}\right)C_C;\ P_{УД\Sigma} = \left(1+\frac{0{,}0212}{1+5{,}5}\right)125{,}9 = 127{,}08.$$

Air flow rate, specific fuel flow rate and internal efficiency are determined by the formulas:

$$G_B = \frac{P}{P_{УД\Sigma}};\ G_B = \frac{70000}{127{,}08} = 550{,}83;$$

$$G_{BI} = \frac{G_B}{1+m} = \frac{550{,}83}{1+5{,}5} = 84{,}74;\ G_{BII} = \frac{m}{1+m}G_B = \frac{5{,}5}{6{,}5}550{,}83 = 466{,}09;$$

$$C_{УД\Sigma} = \frac{3600\, g_T\left(1-g_{ОХЛ}\right)}{P_{УД\Sigma}(1+m)};\ C_{УД\Sigma} = \frac{3600\cdot 0{,}0212(1-0{,}1)}{127{,}08(1+5{,}5)} = 0{,}083;$$

$$\eta_\ell = \frac{L_Ц}{Q_0};\ L_Ц = \frac{C^2{}_C}{2};\ Q_0 = \frac{C_{УД\Sigma}\cdot H_U\cdot P_{УД\Sigma}}{3600}$$

$$\eta_\ell = \frac{P^2{}_{УД\Sigma}}{2g_T\cdot H_U\cdot(1-g_{охл})} = \frac{127{,}08^2}{2\cdot 0{,}0212\cdot 42{,}5\cdot 10^6\cdot 0{,}9} = 0{,}1.$$

3.2.2 Gas dynamic calculation of the turbofan engine TFE

Input data:

- engine type - TFE;
- thrust P=70000 N
- air consumption G=550.83 kg/sec;
- double-circuit degree m=5.5.

Parameters of the operating body in the characteristic cross sections of the flow part should be taken according to the results of thermodynamic calculations.

1. Determination of cross-section sizes at the fan inlet.

Taking into account comparatively big air flow (550,83 kg/s), to reduce diametrical dimensions of the engine we set

- axial air velocity $C_{1\alpha}$ = 200 m/s;
- circumferential speed of fan blades in peripheral section U_{1K} = 500 m/s;

- relative diameter of the hub of the first fan stage $\bar{d}_1 = 0.5$;

We calculate the induced velocity $\lambda_{1\alpha}$ and relative current density $q(\lambda_{1\alpha})$ by the formulas:

$$\lambda_{1\alpha} = \frac{C_{1\alpha}}{C_{KP}};\ \lambda 1\alpha = \frac{200}{18,3\sqrt{288,15}} = 0,644;$$

$$q(\lambda_1) = 0,877.$$

We find the crossing area at the fan inlet from the equation:

$$F_B = \frac{G\sqrt{T^*{}_B}}{m_B\,P^*{}_B\,q(\lambda_1)};\ F_B = \frac{550,83\sqrt{288,15}}{0,040348\cdot 101325\cdot 0,877} = 2,6\text{м}^2.$$

The diameter of the LC (PK) at the periphery is calculated by the formula:

$$D_{1K} = \sqrt{\frac{4F_B}{\pi\left(1-\bar{d}^2{}_1\right)}};\ D_{1K} = \sqrt{\frac{4\cdot 2,6}{\pi\left(1-0,5^2\right)}} = 2,1\text{м};$$

$$D_{1BT} = \sqrt{D^2{}_{1K}\frac{4}{\pi}F_B};\ D_{1BT} = \sqrt{2,1^2 - \frac{4}{\pi}2,6} = 1,05\text{м}.$$

The diameter of the conditional section separating the flows of I and II circuits, we find by the formula:

$$D_I = \sqrt{D^2{}_{1K} - \frac{4}{\pi}F_{II}};\ D_I = \sqrt{2,1^2 - \frac{4}{\pi}2,2} = 1,27\text{м};$$

$$F_{II} = F_B\frac{G_{BII}}{G_B} = 2,6\cdot\frac{466,09}{550,83} = 2,2.$$

where G_B = 550.83 kg/s, G_{BII} = 466.09 kg/s - air flow rate (total and through the outer loop, determined by thermodynamic calculation).

2. Determination of number of fan stages of two-circuit turbojet engine. Circumferential velocity of blades at diameter we calculate by formula:

$$U_I = U_{1K}\frac{D_I}{D_{1K}};\ U_I = 500\cdot\frac{1,27}{2,1} = 302,38\,\text{м}/\text{с}.$$

Lattice density in the LC(PK) sleeve is taken $\left(\frac{B}{t}\right)_{BT} = 2.$. We calculate the density of the lattice, the twisting of the air and the work provided to the air by the working blades of the fan, on the diameter, according to the equations:

$$\left(\frac{B}{t}\right)_I = \left(\frac{B}{t}\right)_{BT}\cdot\frac{D_{1BT}}{D_I};\ \left(\frac{B}{t}\right)_I = 2\cdot\frac{1,05}{1,27} = 1,65.;$$

$$\Delta W_{UI} = C_{1\alpha}\frac{1,55}{1+1,5\left(\frac{t}{B}\right)_I};\ \Delta W_{UI} = 200\frac{1,55}{1+1,5\frac{1}{1,65}} = 162,39\,\text{м}/\text{с};$$

$$L_{UI} = U_I\,\Delta W_{UI};\ L_{UI} = 302,38\cdot 162,39 = 49103,49\frac{\text{Дж}}{\text{кг}}.$$

The average value of the fan's work in the zone of the internal circuit is determined by the formulas:

$$\Delta W_{UBT} = C_{1\alpha}\frac{1,55}{1+1,5\left(\frac{t}{B}\right)_{BT}} = 200\frac{1,55}{1+1,5\frac{1}{2}} = 177,14\,\text{м}/\text{с};$$

$$U_{1BT} = U_{1K}\frac{D_{1BT}}{D_{1K}} = 500\frac{1,05}{2,1} = 250 м/с;$$

$$L_{UBT} = U_{BT}\Delta W_{UBT};\ L_{UBT} = 250\cdot 177,14 = 44285\frac{Дж}{кг};$$

$$L_{ВЛI} = \frac{1}{2}(L_{UBT} + L_{UI} = \frac{1}{2}(49103.49 + 44285) = 46694.245\frac{Дж}{кг}.$$

Thus, finally, we accept a single-stage fan, the dimensions at the inlet $D_{1K} = 2,1 м.;\ D_{1BT} = 1,05 м;\ D_I = 1,27 м.;\ U_{1K} = 500 м/с.$

Operation in the zone of the external contour $L_{UI} = 49103,49\frac{Дж}{кг}$, in the zone of the internal contour $L_{UBT} = 44285\frac{Дж}{кг}$.

3. Distribution of compression work between compressor cascades and determination of the number of high-pressure turbine stages. In order to check whether it is possible to execute the fan without connected stages, we determine the work which in this case falls on the HPS:

$$L^I{}_{КВД} = \frac{k}{k-1}\cdot R\cdot T^*_{КНД}(\pi_{КВД}^{\frac{к-1}{к}} - 1)\cdot\frac{1}{\eta_{КВД}};$$

$$L^I{}_{КВД} = \frac{1,4}{1,4-1}\cdot 287,3\cdot 409,6\cdot(13^{\frac{1,4-1}{1,4}} - 1)\cdot\frac{1}{0,86} = 516203,052\frac{Дж}{кг}.$$

We find the work of the high-pressure turbine by the formula:

$$L^I{}_{ТВД} = \frac{L^I{}_{КВД}}{(1+q_T)(1-q_{ОХЛ})\eta_M};$$

$$L^I{}_{ТВД} = \frac{516203,052}{(1+0,0212)(1-0,1)0,99} = 567256,1\frac{Дж}{кг}.$$

Approximately determine at what circumferential speed of the high-pressure turbine the work can be obtained $L_{ТВД} = 465\frac{кДж}{кг}$.

Let $Y^* = 0,5$. Then in case of two-stage turbine ($Z = 2$, $\eta_T = 0,89$) we obtain:

$$U_{T\,CP} = Y^*\sqrt{\frac{Z\,L_T}{Z\,\eta_{ТВД}}};\ U_{T\,CP} = 0,5\sqrt{\frac{2\cdot 968561,87}{2\cdot 0,89}} = 521,6 м/с.$$

Let's take $U_{T\,CP} = 521,6 м/с.$

Consequently, we can accept engine scheme with two-stage high-pressure turbine and one-stage fan. Let's determine the number of connected stages:

$$L_{ПР} = L_K - L_{ВЛ} - L_{КВД} = 637661,052 - 46694,245 - 516140,199 = 74826,608\frac{Дж}{кг}$$

We distribute the turbine operation between the two stages:

$$L_{CT1} = 340353,6\frac{Дж}{кг};\ L_{CT2} = 226902,4\frac{Дж}{кг}.$$

4. Determination of air parameters and diametric dimensions at fan outlet.

We define the degree of air pressure increase in the fan in the zone of internal contour, taking - as in thermodynamic calculation:

$$\pi^*_{КНД} = \left[1 + \frac{L_{КНД}\eta^*_{ВЛ\,I}}{\frac{K}{K-1}RT^*_B}\right]^{\frac{K}{K-1}};\ \pi^*_{КНД} = \left[1 + \frac{121458 \cdot 0,88}{1005 \cdot 288,15}\right]^{3,5} = 3.$$

The pressure and temperature of the air at the LPS outlet are determined by the formulas:

$$P^*_{КНД} = P^*_B \cdot \pi^*_{КНД};\ P^*_{КНД} = 101325 \cdot 3 = 304177\ Па.$$

$$T^*_{КНД} = T^*_B + \frac{K-1}{K \cdot R}L_{КНД};\ T^*_T = 288,15 + \frac{1,4-1}{1,4 \cdot 287,3}121458 = 408,88\,K.$$

Since the fan is a one-stage fan, and having a velocity drop of the degree of about 10 m/s, we take $C_{\alpha ВЛ\,I} = 200 м/с$, and $C_{\alpha ВЛ\,II} = 190 м/с$ taking into account the two connected stages. We calculate λ, $q(\lambda)$, areas of connections $F_{ВЛ\,I}$ and $F_{ВЛ\,II}$ by formulas:

$$\lambda_{1\alpha ВЛ\,I} = \frac{C_{1\alpha}}{18,3\sqrt{T^*_{ВЛ\,I}}};\ \lambda_{1\alpha ВЛ\,I} = \frac{200}{18,3\sqrt{334,59}} = 0,59.;$$

$$\left.\begin{array}{l} q(\lambda_\alpha)_{ВЛ\,I} = 0,8015 \\ q(\lambda_\alpha)_{ВЛ\,II} = 0,7823 \end{array}\right\}$$ according to the tables of gas dynamic functions.

$$F_{ВЛ\,I} = \frac{G_{BI}\sqrt{T^*_{ВЛ\,I}}}{m_B\,P^*_{ВЛ\,I}\,q(\lambda_\alpha)_{ВЛ\,I}};\ F_{ВЛ\,I} = \frac{84,74\sqrt{334,59}}{0,040348 \cdot 151987,5 \cdot 0,8015} = 0,315 м^2.;$$

$$F_{ВЛ\,II} = \frac{G_{BII}\sqrt{T^*_{ВЛ\,II}}}{m_B\,P^*_{ВЛ\,II}\,q(\lambda_\alpha)_{ВЛ\,II}};\ F_{ВЛ\,II} = \frac{466,09\sqrt{332,26}}{0,040348 \cdot 151987,5 \cdot 0,7823} = 1,77 м^2.$$

The outside diameter of the fan $D_{ВЛ\,II}$ is taken equal to $0.9D_{1K}$,

$$D_{ВЛ\,II} = 0,9 \cdot 2,1 = 1,89 м.$$

The diameter of conditional section separating flows of external and internal circuits at the fan outlet is determined by the formula:

$$D_{II} = \sqrt{D^2_{ВЛ\,II} - \frac{4}{\pi}F_{ВЛ\,II}};\ D_{II} = \sqrt{1,89^2 - \frac{4}{\pi}1,78} = 1,148 м.$$

We accept the thickness of the separating case between I and II circuits to be 20 mm. Then external diameter of I contour is equal to 1.148 - 0.020 = 1.128 m. The diameter of the bushing is determined by the formula:

$$D_{ВТ\,ВЛ} = \sqrt{D^2_{ВТ\,ВЛ} - \frac{4}{\pi}F_{ВЛ\,I}};\ D_{ВТ\,ВЛ} = \sqrt{1,128^2 - \frac{4}{\pi}0,346} = 0,91 м.$$

So, at the outlet of the fan we have:

$$\left.\begin{array}{l} D_{ВЛ\,II} = 1,89 м \\ D_{II} = 1,148 м \end{array}\right\}$$ - the outer circuit;

$$\left.\begin{array}{l} D_{ВЛI} = 1,128 м \\ D_{ВТ\,ВЛ} = 0,91 м \end{array}\right\}$$ - the inner circuit;

5.Determination of diametrical dimensions at the inlet of the high-pressure compressor.

Parameters of air at the inlet of the HPS:

- air temperature: $T^*_{В\,КВД} = T_{ВЛ\,I} = 334,59\,K$;

- air pressure including losses in the transition case between the fan and HPS:

$P^*_{В\,КВД} = P^*_{ВЛ}\,\sigma_{ПЕР}$; $P^*_{В\,КВД} = 151987,5 \cdot 0,99 = 150467,6\,Па.$

We set the air velocity $C_{\alpha\,КВД} = 185\,м/с.$ at the inlet to HPS. We find:

$$\lambda_{В\,КВД} = \frac{C_{\alpha\,КВД}}{18{,}3\sqrt{T^*_{ВЛ\,I}}};\ \lambda_{В\,КВД} = \frac{185}{18{,}3\sqrt{334{,}59}} = 0{,}55\,м.;$$

$$F_{В\,КВД} = \frac{G_B\sqrt{T^*_{ВЛ\,I}}}{m_B\,P^*_{В\,КВД}\,q\left(\lambda^B\right)_{КВД}};$$

The relative diameter of LC (PK) is taken at the inlet to HPS $d_{ВТ} = 0{,}5$ and we find the outer diameter $D_{1КВД}$; D1квд:

$$D_{1КВД} = \sqrt{\frac{4F_{В\,КВД}}{\pi\left(1-d^2_{ВТ}\right)}};\ D_{1КВД} = \sqrt{\frac{4\cdot 0{,}335}{\pi\left(1-0{,}5^2\right)}} = 0{,}754\,м.$$

Diameter of RK bushing of the first stage of HPS we find by the formula:

$$D_{ВТ\,КВД} = \sqrt{D^2_{1КВД} - \frac{4}{\pi}F_{В\,КВД}};\ D_{ВТ\,КВД} = \sqrt{0{,}754^2 - \frac{4}{\pi}0{,}335} = 0{,}377\,м.$$

The height of the blades at the inlet to HPS is determined by the formula:

$$h_Л = \frac{D_{1КВД} - D_{ВТ\,КВД}}{2};\ h_Л = \frac{0{,}754 - 0{,}377}{2} = 0{,}19\,м.$$

6. Determination of diametrical dimensions at the high-pressure compressor outlet. First, we specify the air parameters at the HPS outlet. According to the thermodynamic calculation, the pressure $P^*_K = 3039750\,Па.$ Work $L_{КВД} = 516203{,}052\,\frac{Дж}{кг}$. Air temperature at HPS $T^*_K = 922{,}93\,K.$

The degree of air pressure rise in HPS shall be determined by the formula

$$\pi^*_{КВД} = \frac{P^*_K}{P^*_{КНД}\cdot\sigma_{ПЕР}};\ \pi^*_{КВД} = \frac{3039750}{304177{,}65\cdot 0{,}99} = 10{,}094.$$

We set the air velocity at the HPS outlet: $C_{\alpha КВД} = 120\,м/с.$

We calculate $\lambda_{\alpha K}$ and $q\left(\lambda_\alpha\right)_K$ by the formulas:

$$\lambda_{\alpha K} = \frac{C_K}{18{,}3\sqrt{T^*_K}};\ \lambda_{\alpha K} = \frac{120}{18{,}3\sqrt{922{,}93}} = 0{,}217$$

$q\left(\lambda_\alpha\right)_K = 0{,}331$ - according to the tables of gasdynamic functions. We find the crossing area at the HPS outlet by the formula:

$$F_K = \frac{G_{BI}\sqrt{T^*_K}}{m_B\,P^*_K\,q\left(\lambda_B\right)_K};\ F_K = \frac{84{,}74\sqrt{922{,}93}}{0{,}040348\cdot 3039750\cdot 0{,}331} = 0{,}0632\,м^2.$$

Assuming that $D_{1КВД} = 0{,}754\,м$ =const, we find:

$$D_{ВТ\,K} = \sqrt{D^2_{1КВД} - \frac{4}{\pi}F_K};\ D_{ВТ\,K} = \sqrt{0{,}754^2 - \frac{4}{\pi}0{,}0632} = 0{,}6989\,м;$$

$$h_Л = \frac{D_{1КВД} - D_{ВТ\,К}}{2};\ h_Л = \frac{0,754-0,6989}{2} = 0,0276 м;$$

$$d_{ВТ\,К} = \frac{D_{ВТ\,К}}{D_{1КВД}};\ d_{ВТ\,К} = \frac{0,6989}{0,754} = 0,927.$$

7. Determination of diameter dimensions at the high-pressure turbine inlet. When determining the number of stages of the high-pressure turbine, we obtained $U_{Т\,СР} = 521,6 м/с$ the value adopted for the first stage work $L_{СТ1} = 340353,6 \frac{Дж}{кг}$.

We set the flow exit angle from CA $\alpha_1 = 20°$ and find the gas flow rate from CA:

$$C_1 = \frac{L_{СТ1}}{U_{Т\,СР}\cos\alpha_1};\ C_1 = \frac{340353,6}{521,6\cdot 0,94} = 694,174 м/с;$$

$$\lambda_1 = \frac{C_1}{C_{КР}};\ \lambda 1 = \frac{694,174}{18,5\sqrt{1600}} = 0,94.$$

$q(\lambda_1)_{C\alpha} = 0,9957$ - from the tables of gas dynamic functions. The gas flow rate at the CA outlet is calculated by the formula:

$$G_Г = G_{ВI}(1+q_T)(1-q_{ОХЛ});\ G_Г = 84,74(1+0,0212)(1-0,103) = 77,62\frac{кг}{с}$$

Gas pressure by thermodynamic calculation $P^*{}_Г = 2948557,5\ Па$. The crossing area at the CA outlet is determined by the formula:

$$F_{C\alpha} = \frac{G_Г\sqrt{T^*{}_Г}}{m_Г P^*{}_Г \sigma_{CA} q(\lambda)_{C\alpha}\sin\alpha_1};$$

$$F_{C\alpha} = \frac{77,62\sqrt{1600}}{0,0396\cdot 2948557,5\cdot 0,8\cdot 0,9957\cdot 0,342} = 0,097 м^2.$$

We take $D_{Т\,СР} = 0,98 м$, then:

$$h_Л = \frac{F_{C\alpha}}{\pi D_{Т\,СР}};\ h_Л = \frac{0,097}{\pi 0,98} = 0,032 м;$$

$$D_Т = D_{Т\,СР} + h_Л;\ D_Т = 0,98 + 0,032 = 1,01 м.$$

We assume $D_Т = 1,01 м$ and find $D_{ВТ}$:

$$D_{ВТ} = \sqrt{D^2{}_Т - \frac{4}{\pi}F_{1C\alpha}};\ D_{ВТ} = \sqrt{1,01^2 \frac{4}{\pi}0,097} = 0,95 м.$$

The axial gas velocity at the LC(PK) inlet is determined by the formula:

$$C_{1\alpha} = C_1\sin\alpha;\ C_{1\alpha} = 694,174\cdot 0,34 = 237,42 м/с.$$

8. Determination of diameter dimensions at the high-pressure turbine outlet. Parameters of gas at the high-pressure turbine outlet are determined by formulas:

$$T^*{}_{ТВД} = T^*{}_Г - \frac{L_{ТВД}}{\frac{K_Г}{K_Г - 1}R_Г};\ T^*_{ТВД} = 1600 - \frac{567256,1}{4\cdot 288} = 1106,39 К;$$

$$P^*{}_{ТВД} = 2948557,5\left[1 - \frac{1600-1106,39}{1600\cdot 0,89}\right]^4 = 536637,465 Па.$$

We set the reduced velocity $\lambda_{2\alpha} = 0{,}55$ corresponding to the axial component of gas velocity at the high-pressure turbine outlet:

$$C_{2\alpha} = 0,55 \cdot 18,15\sqrt{1106,39} \cong 332 м/с.$$

According to the tables of gas dynamic functions we find $q(\lambda_{2\alpha}) = 0{,}7651$.

Taking into account that part of the cooled air will enter the gas flow and mix with it, we assume $q_{ОХЛ} = 0{,}1$ and find the flow rates at the high-pressure turbine outlet:

$$G_Г = G_{ВІ}(1+q_T)(1-q_{ОХЛ});\; G_Г = 84,74(1+0,0212)(1-0,96) = 78,23 кг/с.$$

The crossing area at the high-pressure turbine outlet is determined by the formula:

$$F_{ТВД} = \frac{G_Г\sqrt{T^*_{ТВД}}}{m_Г\, P^*_{ТВД}\, q(\lambda_{2\alpha})};\; F_{ТВД} = \frac{78,23\sqrt{1106,39}}{0,0396 \cdot 536637,5 \cdot 0,7651} = 0,157 м^2;$$

We assume $D_{T\,CP} = 0{,}98$ and find the height of the blade of the second stage of the high-pressure turbine (at the outlet edge):

$$h_Л = \frac{F_{ТВД}}{\pi D_{T\,CP}};\; h_Л = \frac{0,157}{\pi 0,98} = 0,051 м.$$

Then

$$D_{ТВД} = D_{T\,CP} + h_Л;\; D_{ТВД} = 0,98 + 0,051 = 1,031.$$

We assume $D_{ТВД} = 1{,}031$ and find $D_{ВТ\,ТВД}$:

$$D_{ВТ\,ТВД} = \sqrt{D^2_{ТВД} - \frac{4}{\pi}F_{ТВД}};\; D_{ВТ\,ТВД} = \sqrt{1,031^2 - \frac{4}{\pi}0,157} = 0,929 м.$$

In order to make sure that the dimensions obtained are acceptable, we draw the flow section of the two-stage high-pressure turbine and find that the angle of expansion of the flow section does not exceed 25° (up to 30° is allowed). Consequently, the diametrical dimensions of the high pressure can be taken as final.

9. Determining the number of high-pressure compressor stages. The work of the first stage of HPS is calculated by taking the density of the lattice $\left(\frac{B}{t}\right) = 2$;

$$\Delta W_{UВТ} = C_{1\alpha}\frac{1,55}{1+1,5\frac{t}{B}};\; \Delta W_{UВТ} = 185\frac{1,55}{1+1,5\frac{1}{2}} = 163,86 м/с;$$

$$U_{1ВТ} = U_{TCP}\frac{D_{1ВТТВД}}{D_{T\,CP}};\; U_{1ВТ} = 521,6\frac{0,929}{0,98} = 494,5 м/с;$$

$$U_{1K} = U_{T\,CP}\frac{D_{1КВД}}{D_{T\,CP}};\; U_{1K} = 521,6\frac{0,754}{0,98} = 401,3 м/с.$$

$$L_{CT1} = U_{ВТ}\Delta W_{UВТ};\; L_{CT1} = 494,5 \cdot 163,86 = 81028,77\frac{Дж}{кг}.$$

The work of the last stage of the HPS is determined by taking the density of the lattice in the sleeve $\left(\frac{B}{t}\right)_{ВТ} = 1{,}8$;

$$\Delta W_{UBT\,Z} = C_{1\alpha K}\frac{1{,}55}{1+1{,}5\dfrac{t}{B}};\ \Delta W_{UBTZ} = 130\frac{1{,}55}{1+1{,}5\dfrac{1}{1{,}8}} = 109{,}96\,м/с;$$

$$U_{1BT\,Z} = U_{T\,CP}\frac{D_{BT\,K}}{D_{T\,CP}};\ U_{1BT\,Z} = 521{,}6\frac{0{,}699}{0{,}98} = 372{,}04\,м/с;$$

$$L_{CT\,Z} = U_{BT\,K}\Delta W_{UBT\,K};\ L_{CT\,Z} = 372{,}04\cdot 109{,}96 = 40909{,}5\frac{Дж}{кг}.$$

The average work of the stage

$$L_{CP} = \frac{81028{,}77+40909{,}5}{2} = 60969{,}1\frac{Дж}{кг};$$

is the number of stages of HPS $Z = \frac{L_{КВД}}{L_{CP}} = \frac{516203{,}052}{60969{,}1} \approx 8{,}46$.

Distribution of work $L_{КВД}$ by stages and change in the axial air velocity in the HPS are shown in Table 1. The sum of works of all stages must be equal to the compressor operation, i.e. $\sum L_{CT1} = L_K$

Table 3.1

Distribution of compression operation and axial velocity of air on the compressor stages:

ТАБЛИЦА 3.1

We take the number of stages of HPS to be 8. Power balance of HPS and high-pressure turbine is checked by formulas:

$$N_{КВД} = G_{BI}L_{КВД};\ N_{КВД} = 84{,}74\cdot 516203{,}052 = 43743046{,}63\,Вт;$$

$$N_{ТВД} = G_{Г}L_{ТВД};\ N_{ТВД} = 78{,}23\cdot 567256{,}1 = 44376444{,}7\,Вт;$$

$$\eta_M = \frac{N_{КВД}}{N_{ТВД}};\ \eta_M = \frac{43743046{,}63}{44376444{,}7} = 0{,}99.$$

High-pressure rotor speed is determined separately for compressor and turbine by the equations:

$$n_{КВД} = \frac{60U_K}{\pi D_K};\ n_{КВД} = \frac{60\cdot 401{,}3}{\pi\cdot 0{,}754} = 10159{,}5\,об/мин;$$

$$n_{ТВД} = \frac{60U_{T\,CP}}{60D_{T\,CP}};\ n_{ТВД} = \frac{60\cdot 521{,}6}{\pi\cdot 0{,}98} = 10095{,}5\,об/мин.$$

10. Determination of number of stages and distribution of work by stages of low-pressure turbine.

Taking into account that the gas temperature at the inlet to the HPS $T^*_{ТВД} = 1106{,}39\,K$ and therefore the LPS may not be cooled, and all the air cooling the high-pressure turbine elements is mixed with the gas flow, we obtain:

$$G_{Г\,ТНД} = G_{BI}(1+q_T);\ G_{Г\,ТНД} = 84{,}74(1+0{,}0212) = 86{,}5\,кг/с;$$

$$L_{ТНД} = \frac{mL_{ВЛ\,II}+L_{КНД}}{(1+q_T)\eta_M};\ L_{ТНД} = \frac{5{,}5\cdot 44314{,}59+121458}{(1+0{,}0212)0{,}99} = 361572{,}5\frac{Дж}{кг};$$

From the scale drawing of the flow part we find that approximately $D_{ТНД\,СР} = 1м$. Then:

$$U_{ТНД\,СР} = U_{1К}\frac{D_{ТНД\,СР}}{D_{1К}};\; U_{ТНД\,СР} = 500\frac{1}{2,1} = 238 м/с.$$

We determine the load parameter y^* at $Z = 4$:

$$y^* = U_{ТНД\,СР}\sqrt{\frac{Z\,\eta^*_{ТНД}}{2L_{ТНД}}};\; y* = 238\sqrt{\frac{4\cdot 1}{2\cdot 361572,5}} \approx 0,56.$$

Hence, it is possible to use four-stage turbine.

$$L_{CT1} = 144629\frac{Дж}{кг};\; L_{CT2} = 108471,8\frac{Дж}{кг};$$

$$L_{CT3} = 72314,5\frac{Дж}{кг};\; L_{CT4} = 36157,25\frac{Дж}{кг}.$$

Determination of diametrical dimensions at the outlet of the first nozzle apparatus of the low-pressure turbine. Let's determine the critical gas velocity in the CA of low-pressure turbine by the formula:

$$C_{КР} = 18,15\sqrt{T^*_{ТВД}};\; C_{КР} = 18,15\sqrt{1106,39} = 603,7 м/с.$$

Taking the angle $\alpha_1 = 20°$, we find the gas outflow velocity from the CA, considering $C_{2U} = 0$:

$$C_1 = \frac{L_{CT1}}{U_{ТНД\,СР}\cos\alpha_1};\; C_1 = \frac{144629}{238\cdot 0.94} = 646,5 м/с;$$

$$\lambda_1 = \frac{C_1}{C_{КР}};\; \lambda_1 = \frac{646,5}{603,7} = 1,07.$$

Consequently, the pressure drop in CA is less than the critical one and it is possible to design the turbine degree with axial gas outlet from LC(PK). In this case, to reduce the pressure drop in CA, we reduce the speed by giving the gas at the LC (PK) outlet a twist in the opposite direction to the rotation.

We set it λ_1 =0.9.

Then $C_1 = \lambda_1 \cdot C_{КР} = 543,33$;

$$C_{2U} = \frac{L_{CT1}}{U_{ТНДСР}} - C_1\cos\alpha = \frac{144629}{238} - 543,33\cdot 0,94 = 96,98 м/с;$$

$$q(\lambda_1) = 0,9883.$$

We find the crossing area at the outlet of the CA of the LPT by the formula:

$$F_{1C\alpha\,ТНД} = \frac{G_Г\sqrt{T^*_{ТВД}}}{m_Г\,P^*_{ТВД}\,\sigma_{ПЕР}\sigma_{C\alpha}\,q(\lambda_1)\sin\alpha_1};$$

$$F_{1C\alpha ТНД} = \frac{86,5\sqrt{1106,39}}{0,0396\cdot 536637,465\cdot 0,975\cdot 0,975\cdot 0,9883\cdot 0,34} = 0,424 м^2;$$

At $D_{ТНД\,СР} = 1м$.

Then the outer diameter at the outlet of the CA LPT:

$$D_T = D_{ТНД\,СР} + h_Л;\; D_T = 1 + 0,15 = 1,15 м.$$

We assume $D_T = 1,15$ and find $D_{ВТ}$:

$$D_{ВТ} = \sqrt{D^2{}_Т - \frac{4}{\pi} F_{1CαТНД}}\ ;\ D_{ВТ} = \sqrt{1{,}15^2 - \frac{4}{\pi} 0{,}46} = 0{,}86 м.$$

The obtained dimensions are put on the flow part drawing and we find that taking into account the transition case between high-pressure turbine and low-pressure turbine the obtained dimensions can be considered acceptable.

Determination of diametrical dimensions at the low-pressure turbine outlet. The gas parameters at the LPT outlet are calculated by the formulas:

$$T^*{}_Т = T^*{}_{ТВД} - \frac{L_{ТНД}}{\frac{К_Г}{К_Г - 1} R_Г}\ ;\ T^*{}_Т = 1106{,}39 - \frac{361572{,}5}{4 \cdot 288} = 816{,}67 К\ ;$$

$$P^*{}_Т = P^*{}_{ТВД}\, \sigma_{ПЕР} \left[1 - \frac{T^*{}_{ТВД} - T^*{}_Т}{T^*{}_{ТВД}\, \eta^*{}_{ТНД}}\right]^{\frac{К_Г}{К_Г - 1}};$$

$$P^*{}_Т = 536637{,}465 \cdot 0{,}975 \left[1 - \frac{1106{,}39 - 816{,}67}{1106{,}39 \cdot 0{,}89}\right]^4 = 132959{,}4\, Па\,.$$

At the LPT outlet the axial component of gas velocity is equal to

$C_{1\alpha} = C_1 \sin\alpha_1$; $C_{1\alpha} = 543{,}33 \cdot 0{,}34 = 184{,}73 м/с$.

We set the reduced velocity at the LPT outlet:

$$\lambda_{\alpha Т} = 0{,}65$$

According to the table of gas dynamic functions we find

$$q(\lambda_{\alpha Т}) = 0{,}8564$$

The crossing area at the LPT outlet is calculated according to the formula:

$$F_Т = \frac{G_Г \sqrt{T^*{}_Т}}{m_Г\, P^*{}_Т\, q(\lambda_{\alpha Т})}\ ;\ F_Т = \frac{86{,}5\sqrt{816{,}67}}{0{,}0396 \cdot 132959{,}4 \cdot 0{,}8564} = 0{,}548 м^2\ ;$$

We assume $D_{ТНДСР} = 0{,}96 м$ and find:

$$h_Л = \frac{F_Т}{\pi D_{Т\,СР}}\ ;\ h_Л = \frac{0{,}548}{\pi \cdot 1} = 0{,}17 м\ ;$$

$$D_Т = D_{ТСР} + h_Л\ ;\ D_Т = 1 + 0{,}17 = 1{,}17 м$$

We assume $D_Т = 1{,}17 м$ and find:

$$D_{ВТ} = \sqrt{D^2{}_Т - \frac{4}{\pi} F_Т}\ ;\ D_{ВТ} = \sqrt{1{,}17^2 - \frac{4}{\pi} 0{,}548} = 0{,}82 м\ .$$

The flowing part of LPT is scaled and we find that the geometrical dimensions are acceptable. Consequently, it is possible to realize LPT with four stages.

We check the balance of turbine and fan power according to the equations:

$$N_{ТНД} = G_Г\, L_{ТНД}\ ;\ N_{ТНД} = 86{,}5 \cdot 361572{,}5 = 31276 кВт\ ;$$

$$N_{ВЛI} + N_{ВЛ} = G_B L_{КНД} + G_{II} L_{ВЛ\,II} = 30087 кВт\ ;$$

$$\eta_М = \frac{N_{ВЛI} + N_{ВЛ\,II}}{N_{ТНД}}\ ;\ \eta_М = \frac{30087}{30330} = 0{,}992\,.$$

The low pressure rotor speeds are determined separately for the fan and turbine by the formulas:

$$n_{КНД} = \frac{60U_{1K}}{\pi D_{1K}};\; n_{КНД} = \frac{60 \cdot 500}{\pi \cdot 2,1} = 4549,5 об/мин$$

$$n_{ТНД} = \frac{60U_{T\,CP}}{\pi D_{T\,CP}};\; n_{ТНД} = \frac{60 \cdot 238}{\pi \cdot 1} = 4547,7 об/мин.$$

13. Determination of the section diameter at the nozzle outlet of the turbofan engine. Nozzle outflow velocity $C_C = 125,9 м/с$

$$\lambda_{C1} = \frac{C_C}{18,3\sqrt{T_C}};\; \lambda_{C1} = \frac{125,9}{18,3\sqrt{395,09}} = 0,35;\; q(\lambda_1) = 0,5273$$

Nozzle area and diameter are determined by the equations:

$$F_C = \frac{G_Г\sqrt{T^*{}_{CM}}}{m_Г P^*{}_{CM}\, q(\lambda_{CI})};$$

$$F_C = \frac{86,5\sqrt{402,83}}{0,0396 \cdot 107849,36 \cdot 0,5273} = 0,77 м^2;$$

$$D_C = \sqrt{\frac{4F_C}{\pi}};\; D_C = \sqrt{\frac{4 \cdot 0,77}{\pi}} = 0,99 м.$$

3.3 Analysis of influence of operational and technological factors on damage

Since the durability of materials is a function of stresses and temperatures acting in the structural elements (SE), then the action of the transition from the GTE workflow parameters to the loading parameters is necessary to have an appropriate mathematical model of the KE load, it is necessary to have an appropriate mathematical model of the SE loading. As such a model we use the method [51].

We determine the stress by varying the circumferential speed U_i, depending on the relative speed $\bar{n}$:

$$U_i = \frac{n_i \cdot 3.14 \cdot D_{TBT}}{60},$$

Where $n_i = \bar{n} \cdot n_p$ is the rotor speed of BT, *rpm*.

The relative radius of dangerous section is found by formula:

$$\bar{r}_{руйн} = \frac{r_{руйн}}{R_к}$$

According to the method presented in [51], we find tensile, bending and total stresses.

Results of calculations in Mathcad are summarized in Table 3.2.

Table 3.2.

Stress and temperature values at throttling

Relative friction of rotation, n_i	Angular velocity, U_i, м/с	Tension, stretching $\sigma_р$, Па	Bending stress, $\sigma_{зг}$, Па	Combined stresses, σ , Па	Blade temperature T, К

		10^7	10^7	10^7	
0,9	293,592	3,916	0,8615	4,778	1102
0,92	306,249	4,092	0,9003	4,992	1120
0,94	312,907	4,272	0,9398	5,212	1138
0,96	319,565	4,456	0,9802	5,436	1158
0,95	326,222	4,643	1,022	5,665	1178
1,0	332,880	4,835	l,064	5,898	1199
1,02	335,537	5,03	1,107	6,137	1221

Using the results of Table 3.2 we make a diagram of total stresses change in a dangerous section of the blade at throttling (Fig.3.4).

From the diagram it is seen, that at throttling from 0,9 to 1,02 the total stresses increase practically in direct proportion.

Similarly to the preliminary calculations, we determine in Mathcad durability and damage for the dangerous intersection according to the formulas given in the method [51]. The relative damage capacity is calculated by the formula

$$\overline{\Psi}_i = \frac{\Psi_i}{\Psi_{руйн}}$$

where Ψ_destruction = 0.0141 is the damageability of failure in the critical cross-section.

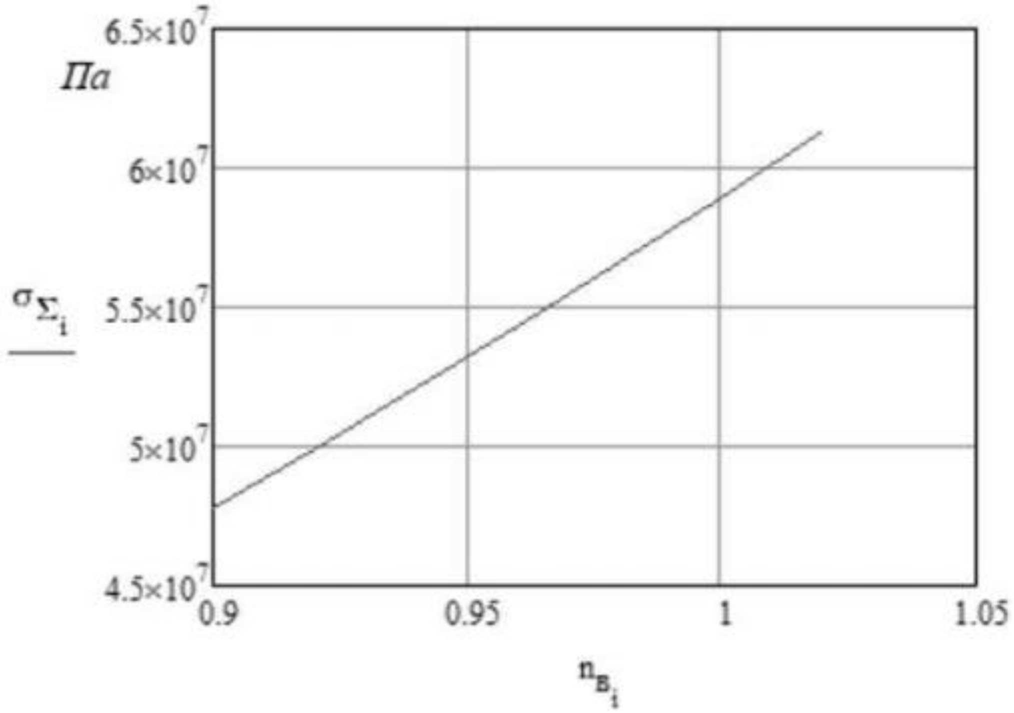


Figure 3.4 - Dependence plot of total stresses at radius $r_{руйн}$ ($r_{destruction}$) at throttling.

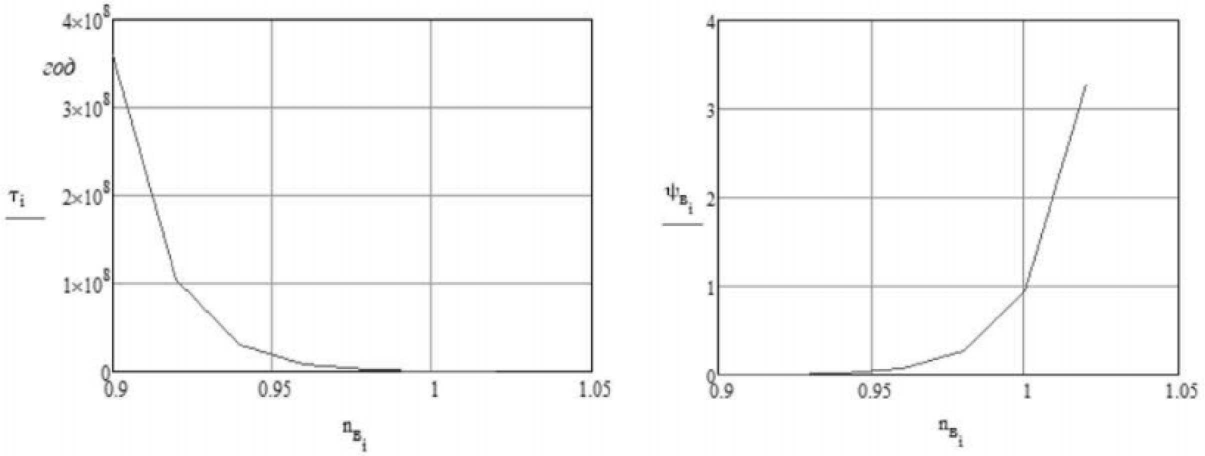
Let's sum up the results of calculations in table 3.3.

Table 3.3.

Stress, durability and damage parameter value in the dangerous section at throttling

Relative friction of rotation,, n_i	Combined stresses, σ , Па 10^7	Stress parameter, P_{σ} 10^4	Durability, τ 10^6	Damage, Ψ_i,	Relative damage, $\overline{\Psi}_i$
0,9	4,778	3,140	359,8	$2{,}773 \cdot 10^{-5}$	$1{,}985 \cdot 10^{-3}$
0,92	4,992	3,137	1104,8	$9{,}541 \cdot 10^{-5}$	$6{,}815 \cdot 10^{-3}$
0,94	5,212	3,129	30,64	$3{,}263 \cdot 10^{-4}$	0,023
0,96	5,436	3,12	8,959	$1{,}116 \cdot 10^{-3}$	0,08
0,98	5,665	3,112	2,611	$3{,}83 \cdot 10^{-3}$	0,274
1,0	5,898	3,103	0,7567	0,013	0,945
1,02	6,137	3,095	0,2176	0,046	3,283

According to the results of calculations let's make diagrams of dependences of durability and relative damage at throttling (Fig.3.5).



а) б)

Figure 3.5 - Dependence of durability (a) and relative damage (b) at throttling

According to the chart we see that during throttling the durability decreases close to the smooth downward curve and tends to zero with increasing relative speed. Throttling damage increases near the rising curve. At relative frequency close to unity and above, the value of damage increases sharply.

We accept outside air temperature in a range of three points: $T_в$ = 278 K; 288 K (design mode); 298 K.

First, we calculate all GTE parameters according to the method stated earlier.

We perform calculations in Mathcad. According to the calculation results, the change of GTE parameters is visible.

For more truthful and objective conducted modeling we achieve the same values of GTE power in the end of calculation by mathematical model for definition of loading parameters equal to power value on calculation at $T_в$ = 278 K by selection of relative speed.

The further calculation of total stresses, durability and damage is carried out by the methodology stated above.

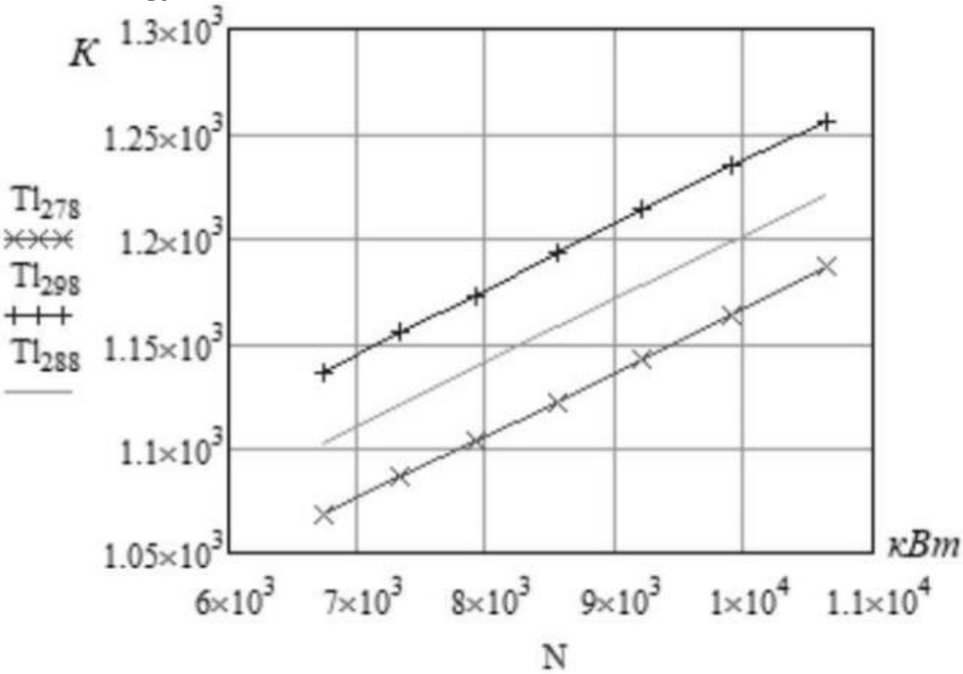


Fig. 3.6 - Dependence of blade temperature at different inlet temperatures (278; 288; 298 K) on GTE capacity

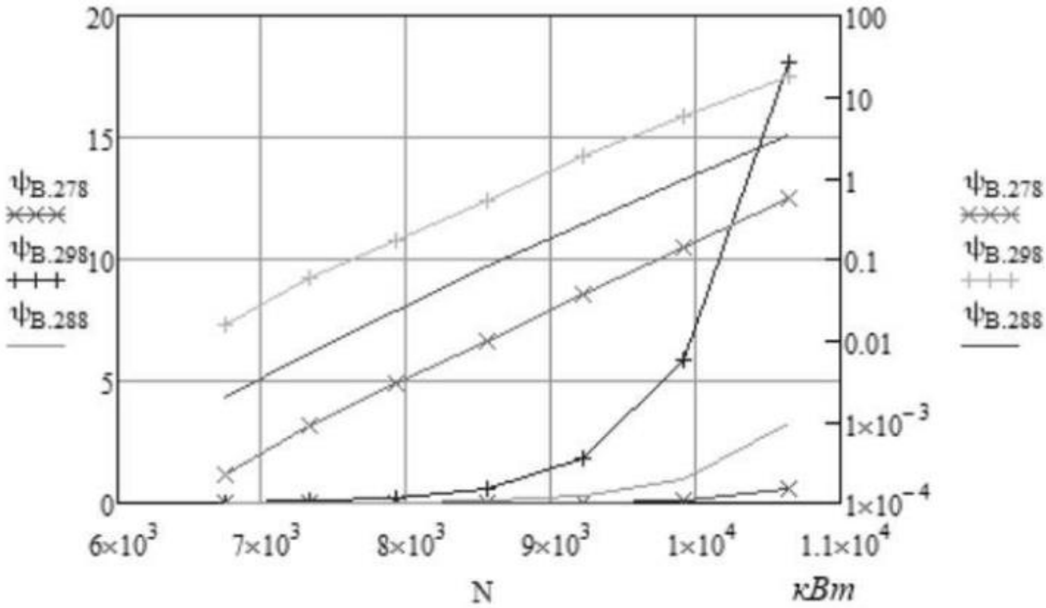


Fig. 3.7 - Dependence of relative blade damage and decimal logarithm of relative damage at different inlet pressure $T_в$ on GTE capacity.

After the analysis of the above figures (Fig.3.6-3.7) it is distinctly seen that at lowering the inlet temperature the values of temperatures and of the blade, of the total stress, as well as of the relative damage are markedly reduced, at increasing the inlet temperature all the listed parameters are expected to increase. The values of the parameters themselves also increase as the power increases along a clearly expressed rising straight line. Value of relative damage at growth of power values to values, where relative speed of high pressure rotor and higher, sharply increases.

We carry out the change in compressor efficiency i of the total compressor pressure increase [46] (one and two percent less than the calculated parameters).

In this way we modulate the change in the technical condition of the GTE due to contamination of the flow part of the engine over time.

Similarly, we make all calculations on both mathematical models, and by selecting the relative speed of rotation, we achieve the equation of capacities when changing η_k^*i π_k^*, on the input.

$$\pi_{к}^{*} = \pi_{кр}^{*} \cdot 0{,}99 = 19 \cdot 0{,}99 = 18{,}81;$$

$$\pi_{к}^{*} = \pi_{кр}^{*} \cdot 0{,}98 = 19 \cdot 0{,}98 = 18{,}62;$$
$$\eta_{к}^{*} = \eta_{кр}^{*} \cdot 0{,}99 = 19 \cdot 0{,}99 = 18{,}81;$$
$$\eta_{к}^{*} = \eta_{кр}^{*} \cdot 0{,}98 = 19 \cdot 0{,}98 = 18{,}62.$$

Similarly, we make all calculations for both mathematical models, and by selecting the relative speed, we achieve the equation of powers when changing $\eta_{к}^{*}$i and $\pi_{к}^{*}$at the input.

All calculation results are presented in the form of graphs (fig.3.8 -3.9).

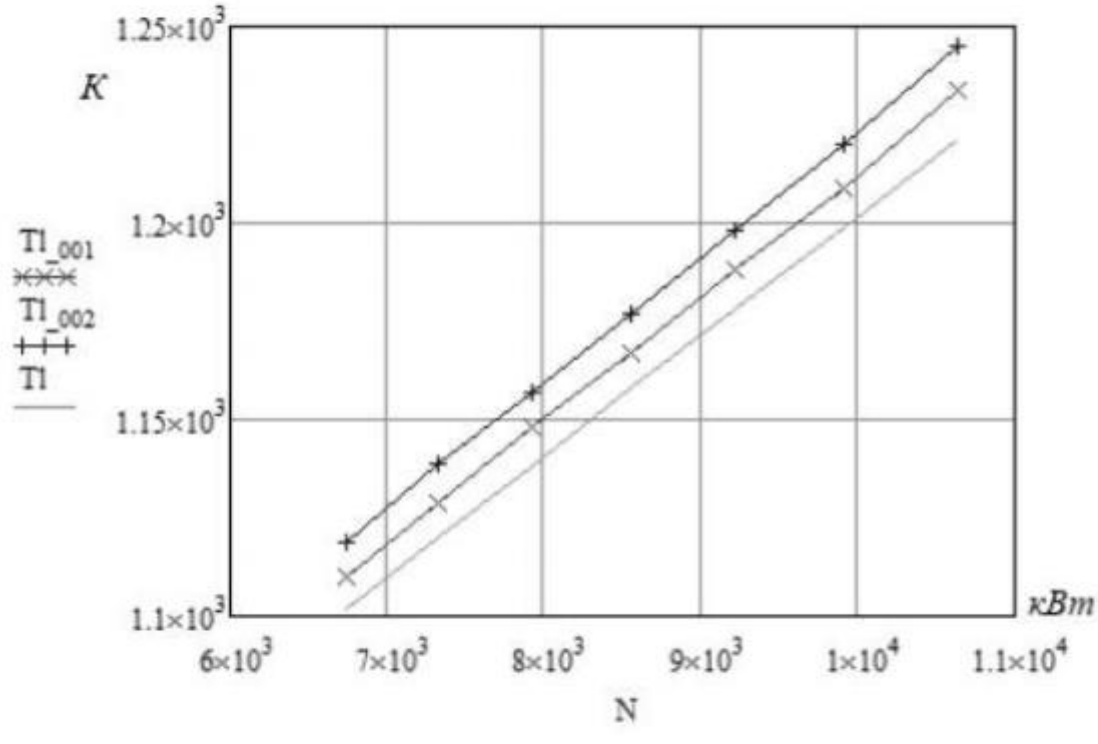


Figure 3.8 - Dependence of blade temperature at lowering $\eta_{к}^{*}$ and $\pi_{к}^{*}$ on GTE power

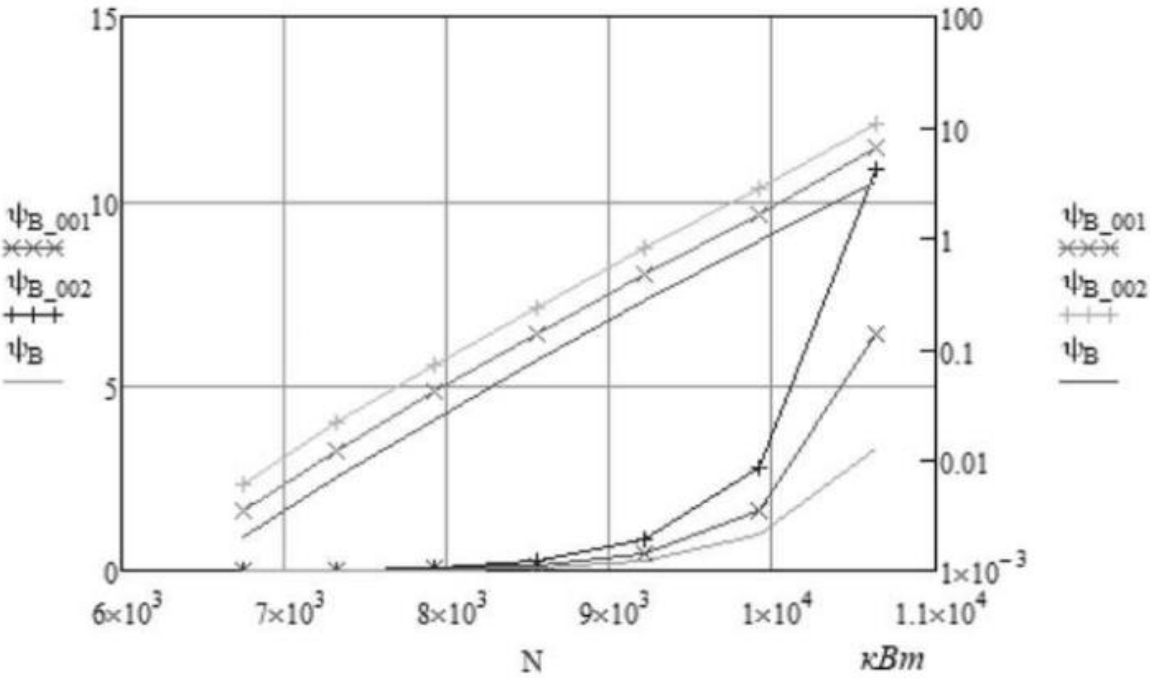


Fig. 3.9 - Dependence of relative blade damage at change from power GTE

After the analysis of the above figures (Fig.3.8-3.9) it is clearly seen that when changing (reducing) the compressor efficiency $\eta_{к}^{*}$as well as reducing the compressor pressure rise $\pi_{к}^{*}$, the blade temperature, the total stress, as well as the relative damage increases, and the more this decrease, the more increase of the observed parameters. The values of the parameters themselves also increase with increasing power along a clearly expressed rising straight line. The value of the relative damage when the power values increase to values where the relative speed of the high pressure rotor $n = 1$ or higher increases sharply, especially when the reduction of $\eta_{к}^{*}$ and $\pi_{к}^{*}$ is increased.

3.4 Decision making on control actions based on the results of damage parameters monitoring

The operating conditions and individual features of GTEs cause a high intensity of damage accumulation during the monitoring period of operation (the average damage is much higher than the average damage for the fleet). Also, the damage level is greatly influenced by the air parameters at the engine inlet, which differ significantly by seasons of the year. In the process of GTE operation, the contamination of the engine flow part is possible, as a consequence of which also the level of accumulated damage significantly increases.

To reduce the intensity of damage accumulation control actions are necessary:

- adjustment of the automatic control system;
- flushing and cleaning of the flow part;
- change of GTE operation modes;
- transfer to the reserve etc.

For acceptance of adequate control action it is necessary to carry out additional researches directed on definition of the factor which has caused the increased intensity of damage accumulation.

At the moment when the engine works out the appointed service life, the parts under control have a significant residual service life by the damage criterion. This fact can be one of the reasons for extending the individual assigned service life.

Conclusions to Chapter 3

The section assesses the influence of GTE operation conditions and its technical condition on the damage accumulation in its turbine parts by the example of PS-90A turbine propulsion jet engine.

The general information about PS-90A turbofans is described.

The thermodynamic and gasdynamic calculation of TFE is performed.

The influence of operating and technological factors on the damageability is analyzed.

Solutions for control actions based on the results of damage parameters monitoring have been proposed, namely: regulation of the automatic control system; flushing and cleaning of the flow part; changes in the GTE operation modes; transfer to the reserve, etc.

The analysis of the influence of the operational and technological factors on the damage, in particular: the GTU operation mode on the damage; the temperature and pressure at the GTU inlet on the damage; the parameters of the GTD nodes technical state on the damage. As a result, the dependences of the blade temperature, the total stress and the relative damage of the high pressure rotor speed during the change of operational and technological factors have been obtained.

SECTION 4
LABOUR SAFETY AND HEALTH

4.1 Hazardous and harmful industrial factors, arising at GTE operation and repair

During GTE operation and repair the following dangerous and harmful production factors can operate on the workers:

- during loading and unloading operations during installation and dismantling of the unit (movable items, workpieces and materials);
- unprotected movable elements of GTE, lifting mechanisms and production equipment;
- vehicles for delivery of equipment units;
- destruction and dispersal of fragments, components, parts of the production equipment;
- falling tools and materials during maintenance and repair work;
- increased slipperiness (as a result of ice, moistening and oiling of the installation surface;
- an increased noise level (reduces productivity, quickly causes a feeling of fatigue, can be a consequence of occupational diseases)
- increased level of vibration;
- increased level of ultrasound;
- increased infra-red radiation from heated drive parts;
- increased ultraviolet and thermal radiation;
- increased dust and gas content in the GTE area;
- increased or decreased temperature of the GTE surface of the equipment and materials;
- increased or decreased temperature and humidity of the air in the GTE zone;
- dangerous voltage level in the electric circuit;
- absence or lack of sunlight;
- reduced contrast of the resolution objects with the background;
- toxic substances (gas used as fuel and as a compression object);

- physical overload (static and dynamic);
- neuropsychological (emotional).

4.2 Organizational, structural and technological measures to reduce the level of hazardous and harmful production factors

In order to ensure the safety of the operating personnel in the development of GTE there are measures to increase the level of safety during operation and maintenance of the unit. These measures may include the calculation of operating loads that may arise in case of possible accidents.

The control panel is located in a certain block and is behind the kill zone at probable engine and supercharger destruction, which allows to protect the personnel from a number of harmful and dangerous factors of drive and supercharger parts affection at their destruction and dispersal. In addition, the turbine and supercharger compartments of the GTE are separated by a concrete wall and locked in a protective casing, which also plays the role of a thermal screen and noise-absorbing design element.

The coupling design, which connects the engine and the supercharger, allows to compensate vibration and dampens resonance vibrations, arising during GTE operation.

An air conditioning system is included in the design to create a comfortable working environment for the personnel. This system can operate both in normal and emergency modes.

Special silencers are used to reduce the noise level of the air intake and exhaust device of the GTE. Container walls of all units of the unit are made of special panels filled with sound-absorbing material.

During operation of the plant, the probability of loss of transported gas is not excluded, which leads to an explosive and emergency situation, as well as harms the health of the operating personnel. To prevent such cases any GTE is equipped with gas analyzers and gas loss alarms. Personal respiratory protection kits are kept at the plant, and personnel are trained to learn how to use personal protective equipment.

4.3 Calculation of lightning protection for the heat exchanger

The object (heat exchanger) with radius r_x=10 m and height h_x=5 m.

Calculation. Let's substitute the value of the object (heat exchanger) radius r_x and height h_x in the formula. Then we get:

$$h = \frac{r_x + 1{,}63 h_x}{1{,}5} = \frac{10 + 1{,}63 \cdot 5}{1{,}5} = 12{,}1 \text{ м}$$

Lightning rod supports are made of wood, reinforced concrete and metal. At DOS warehouses, the most accepted supports are made of reinforced concrete and metal. Reinforced concrete supports for freestanding lightning rods have a number of advantages.

Rod lightning arrestors are made of a steel rod with a cross section of at least 100 mm^2 or water pipes. The pipe at the top must be caulked. A collector must be resistant to mechanical stress. The length of the rod lightning collector should not exceed 2-2.5 m.

Rolled steel bars of various profiles are used as current conductors. For leads, laid on the walls of buildings and structures, most often used strip steel flat cross section of 48 mm^2. They are laid at the shortest distance between an earth electrode and an earth electrode and they are reinforced with steel staples, nails and dowels. Connections between an earth electrode and an earth electrode are made by welding. If the support is metal, it is used as an earth electrode.

The design of earthing chosen depending on the normalized impulse resistance. A distinction is made between the impulse current flow resistance R, the direct lightning strike current and the earthing impedance of the earth leakage current of the industrial part. The impedance of the grounding device at the outflow of industrial current is determined fairly accurately if the short-circuit current, the size of the earth electrode and the soil resistivity is known. Determining the earth resistance for pulsed lightning currents is very difficult, which depends on a number of reasons. Impulse lightning current around the ground causes the emergence of a spark zone, which covers the earth electrode around it, increases its geometric dimensions and thereby reduces the value of the resistance from the earth electrode.

4.4 Fire and explosion safety during GTE maintenance

The main source of fire hazard at the GCU is the gas turbine engine, which uses natural gas as fuel. In addition to this, the turboblock engine block also contains the heat-stressed radial diffuser (volute) and the engine connection to the volute and torsion shaft shroud, which are dangerous in terms of fire. The blower is also a source of explosion hazard as its working medium is natural gas (methane).

Possible gas losses in case of leakage of joints or for other reasons create an explosive mixture of air and gas at a concentration of the latter from 4% to 8% (by volume).

The automatic fire-extinguishing system is developed on the basis of analysis of possible fire situations and provides fire protection of engine and blower compartments by timely detection of a fire point and its subsequent suppression by automatic extinguishing agent supply (both during unit operation and when it is in reserve or under repair). The extinguishing agent can be triggered by the fire detectors, remotely from the fire starter located in the control room or in the automation compartment, or by means of the manual starter in the fire extinguishing compartment.

The fire-extinguishing system includes an aggregate part, main pipelines or sprinklers with discharge nozzles. The aggregate part includes two batteries of Bage - 4 - 1, a battery of Bage - 2 - 1, a universal pressure alarm of each protected compartment and electric contact pressure gauges, located in an isolated compartment with non-combustible walls and overlapping with the fire resistance limit of 0.75.

Due to the small size of the fire-fighting compartment, batteries of units are mounted on a wheelbarrow to provide the necessary passageways for installation and removal of cylinders.

When the system operates in automatic mode, the fire signal in the compartment comes from the sensors to the signal-starting fire device, which gives an impulse to explode the squibs of the main battery charging section electric release head and to the automatic control system for emergency shutdown of the unit.

Through the gate-heads, opened by the cable of the electric release head, the extinguishing agent "chladone" falls from the cylinders into the pipeline and through the return valve into the sprayer with nozzles.

When the unit is in operation, the impulse to detonate the squibs of the engine compartment battery is given with a delay of 15...20 s. This is due to the fact that the engine compartment is equipped with a forced ventilation exhaust and to exclude the release of extinguishing agent into the exhaust shaft due to ejection, it is necessary to first of all turn off the fans.

When the system is remotely activated by pressing the button in the automation compartment or on the signal-starting fire device, an impulse is given to the automated control system to emergency stop the drive engine and to explode the squibs of the electric release head of the standby charge section. When the fire extinguishing agent "refrigerant" enters the pipeline, the alarm is activated, which gives a control signal to the signal-starting firefighting device. By electric contact pressure gauges is remotely control the pressure in each of the cylinders.

Table 4.1.

List of primary firefighting equipment and the fire-fighting equipment

Name	Unit of measure	Quantity
Sand box	pcs.	12
A bucket	pcs.	12
Shovel	pcs.	12
Felted cloth	pcs.	12
Fire extinguisher ОУ- 8	pcs.	4
Fire extinguisher ОУ- 25	pcs.	5
Fire extinguisher ОП –50	pcs.	5
Emergency fire pump	pcs.	1
Fire pond 2×250 $м^3$	pcs.	1
Fire hydrant	pcs.	2

4.5 Instruction on labour safety of the machine operator of process compressors

Fire protection is achieved by using one or a combination of the following methods:

- the use of fire extinguishing equipment and appropriate types of fire equipment;
- the use of automatic fire alarms and fire extinguishing equipment;
- the use of basic building structures and materials, including those used for cladding structures, with normalized fire safety indicators;
- the use of impregnation of structures with fire retardants and application of fire retardant paints to their surfaces;
- the timely notification of the evacuation of people by technical means, including automatic means;
- the use of smoke protection equipment.

Fire spread limitation shall be achieved by:

- establishment of fire barriers;
- establishment of maximum permissible, according to technical and economic calculations, areas of fire compartments and sections, and floors of buildings and structures;
- installation of emergency shutdown and switching of communication installations.

4.6 Basic requirements for the observance of occupational safety rules during the operation of the designed motor

Maintenance of the equipment, including its start-up, shutdown and routine issues, must be carried out in accordance with the requirements of the technical instructions of the manufacturer. Operation of GTE with parameters deviating from the values specified in the operating instructions is not allowed.

After the installation of the main and auxiliary equipment during commissioning works, when the construction, installation, operating and commissioning organizations are concentrated at the CS and the only management is heavy, attention should be paid to compliance with safety regulations. Before the supply of gas to the CS, all personnel of construction, installation, commissioning and other organizations involved in the site must be instructed in safety, which should be documented.

When installing the GTE, the following should be paid attention to:

- lifting of the turboblock should be performed with the help of a special traverse;
- lifting of other units (suction chamber, oil cooler unit, exhaust unit, silencer, etc.) shall be performed in accordance with the insurance schemes and recommendations stated in the technical documentation on GTE.

It is necessary to eliminate all leaks and slots in the connections of GTE units, oil tank covers should be installed hermetically. On a non-operating unit, the louvers of air-cleaning device (ACD), air intake compartment of the engine must be closed, inlet diffusers of fans of oil cooler unit must be shrouded.

Work on commissioning, repair and operation of GTE should be carried out in accordance with the requirements of "Safety rules for installation of equipment of compressor stations of main gas pipelines".

The maintenance personnel who have passed special training, passed examinations and are admitted in the established order for their maintenance and operation are allowed to operation and repair of gas-turbine installations.

Before starting the GTE, it is necessary to make sure that the audible signal activates when the Start button is pressed. Starting the unit without guards and covers on rotating parts and nodes located at a height of not more than 2 m from the floor (oil cooling unit fans, starting pumps coupling) or removing them during the unit operation is not allowed.

When operating the GTE, it is prohibited:

- to enter the engine compartment when starting and running the engine;
- to work on the engine when the GTE system is under current;
- to perform work in the suction chamber and exhaust shaft of the unit when starting or running the engine;

- to work with the doors of the engine compartment, supercharger, BOP and suction chamber open.

The air in the oil tank must be checked daily for combustible gas content and recorded in the logbook. If the combustible gas content in the oil tank is more than 1%, the GCU operation is not allowed.

It is not allowed for the operating personnel to stay near the unit without individual noise protection equipment for more than 1 hour during one work shift.

The permissible vibration level of GTE measured by standard equipment must not exceed 30 mm/s.

The airtight partition between the engine compartment and the supercharger compartment must be maintained so that air from the supercharger compartment does not penetrate into the engine compartment. When hot air from the engine is supplied to heat the machine compartments, the personnel working in the compartments must be notified. Protective gloves must be worn when handling hot air fittings.

In case of power failure, stationary portable 12V explosion-proof lanterns must be used.

In winter time the sites for maintenance of GTE should be periodically cleared of snow.

Emergency shutdown of the unit shall be performed in the following cases:

- if the safety of the operating personnel is endangered;
- at breakage of the unit;
- if there is metal knocking and banging;

Severe loss of oil or gas;

- if there is an ignition of oil or gas;
- pumping phenomena in the unit.

Before inspecting or repairing the unit, there must be cooking gas around the blower circuit and "Do Not Turn On! People working."

All engine adjustment work may only be performed when the unit is at a standstill. Maintenance and repair work on the engine must be carried out only after the outer surfaces have cooled down to a temperature of 45 C. During assembly and disassembly of the unit, you must use serviceable special tools and devices to ensure safe operation.

Do not:

- use defective lifting mechanisms and devices to lift the engine, supercharger cover, rotor, and other unit assemblies;

- leave any parts suspended on the hoisting devices;

- operate hoisting devices at temperatures below -20 C.

Lifting devices working in pairs should be loaded evenly, in order to avoid their breakage and personnel injury.

Fire-safe detergents must be used when dismantling and washing the parts.

It is forbidden to store kerosene, gasoline and other flammable materials in the GTE shelters or near them.

Operation of the fire extinguishing unit is not allowed if the service life of the cylinders has expired, as well as defects are found that exclude the guarantee of safe operation of the unit. It is forbidden to transport the unit with fire-extinguishing agent in the cylinders. Enter the compartments of the engine and blower after activation of the fire extinguishing system without a gas mask is allowed only after thorough ventilation and taking samples of gassiness in the room.

To determine the content of harmful substances in the working area air, control measurements must be taken by sampling at least once a year.

The doors of the engine compartment, suction chamber and air purifier must be marked with forbidding safety signs with the inscription "Do not enter during GCU operation".

Fulfillment of the above requirements is necessary for safe and reliable operation of all GCU equipment.

Conclusions on section 4

This section of the diploma paper identifies dangerous and harmful factors when working in the compressor shop, the calculation of lightning protection for the heat exchanger. Measures for fire and explosion safety at maintenance of gas turbine unit are defined.

SECTION 5.
ENVIRONMENTAL PROTECTION

5.1 Impact of GTE and CS on the environment

Natural gas is an environmentally friendly fuel, which allows to radically reduce the pollution of the atmosphere with acid gases at the modern technological and technical level. It is known that during the same power generation, carbon dioxide emissions (one of the most active greenhouse gases) from natural gas combustion are about 25-30 % lower than those from fuel oil combustion and 40-50 % lower than those from natural gas combustion, when burning coal[21].

It is obvious that widespread use of natural gas in the national economy helps to stabilize greenhouse gas emissions, prevent global climate change, environmental and social disruption.

This has been one of the reasons why the evolution of natural gas consumption over the last decade has become one of the most important elements in the development of the global energy industry. In most industrially developed countries, the development of the natural gas industry is a necessary component of the general pattern of development of the energy saving system.

In the last 20 years, global consumption of natural gas has increased by 65%, accounting for 21% of the world's energy balance. Leading experts predict that by the middle of the 21st century this percentage will reach 30%, significantly pushing oil back.

The environmental program of the industry cannot be implemented without introducing industrial environmental monitoring (IEM) on the condition of air, water, soil and subsoil. The fundamental task of IEM is a system of planned repetitive observations that provide an assessment of the state and prediction of changes in natural environments in order to effectively influence these changes.

One of the main objects of the gas transportation system is compressor stations (CS) [14]. It is here that the largest amount of energy-intensive equipment, designed to ensure the technological process of gas transport, ramified systems of technological communications are concentrated, a large number of service personnel are involved. Special environmental services were established to solve current and prospective issues related to environmental protection and performance of control and measuring measures at compressor stations. The main task of the environmental services is to control the environmental impact of compressor station operations.

This control is carried out through chemical, metrological laboratories and various production services.

In addition, specialized organizations are engaged to measure emissions of combustion products, natural gas, discharges into open water bodies, etc.

At the same time, as a rule, the following activities are carried out:

- interaction with state, local environmental and sanitary control bodies on the organization of works;
- timely collection and transmission of statistical data and reporting;
- development and organization of implementation of ecological measures, both industry-specific and those recommended by controlling organizations, aimed at reducing the harmful effect on the environment;

- timely organization of the development of normative documents, regulating the norms of the enterprise's influence on the environment

- conducting expert reviews of projects that are being reconstructed and newly built facilities.

Operation of equipment and technological systems of compressor stations is associated with environmental impacts. Such impacts should include:

- emissions of harmful substances into the atmosphere;
- ischarge of contaminants into water bodies
- toxic wastes;
- impact on soil and subsoil;
- noise, etc.

5.2 Emissions of harmful substances into the atmosphere

Emissions of harmful substances at compressor stations can be divided into two main groups:

- emissions (emissions) of natural gas;
- emissions of combustion products (exhaust gases).

The distribution of the total value of natural gas emissions during its transportation can be presented as the following relations [20]:

- the total value of natural gas emissions at the CS - 100%;
- during starts and stops of GPA (turbo expander, supercharger circuit) - 73 %;
- leaks (diffusion emissions) - 17%;
- sealing of shutoff valves in the stem - 1.86 %;
- flange and threaded connections - 0.47 %;
- safety valves - 2,9 %;
- plug seal of candle shutoff valves - 7.67%;
- compressor seals - 2.81 %;
- other process equipment - 1.29 %;
- repair work, emergencies, etc. - 6 %;
- impact on soil and subsoil;
- noise.

The main types and sources of methane emissions (as the main component of natural gas) at compressor stations can be grouped into the following categories [22]:

(a) planned (design) emissions, i.e., gas emissions into the atmosphere associated with the daily, technologically necessary operation of equipment under standard operating conditions of technological installations. These are emissions from safety valves, triggered at a certain pressure, turbine expanders, degassers and similar technological equipment of the gas transportation system. They are called planned (design) because the values of such leaks are determined on the basis of technical parameters of equipment and can be checked by means of selective (selective) measurements or calculations.

The main value of emissions, connected with the implementation of technological operations at the compressor stations, accounts for the operations carried out at the moment of start-up and shutdown of GCU. The average value of these emissions is characterized by the data of Table 5.1.

Table 5.1.

Gas consumption by components of GCU start-stop operation

Name	Average value gas consumption	
	nm^3	%
Gas consumption for start-stop operation	5264,3	100,0 %
Gas consumption for turbine expander operation	4100	77,9 %
Gas consumption for blowing gas in the blower circuit	61	1,2 %
Gas volume flushed from the blower circuit	1053	20,0 %
Impulse gas consumption in start, stop mode	50	0,9 %

As can be seen from the data of Table 5.1, the greatest gas losses occur during turbo expander operation and during bleeding of gas from the supercharger circuit. These volumes make up about 95 - 97 % of all losses during technological operations;

b) emission during operation and repair works at gas transportation system facilities, connected with periodically conducted measures on maintenance of serviceability of these facilities.

Dust collectors account for relatively large gas emissions during the operation of compressor station process equipment. Annual losses of dust collector blowdowns at some compressor stations are as high as 10 million m3 per year.

The main factors determining the volume of gas losses in blowing dust collectors include:

- technological scheme of dust collector blowing (open, closed);
- type of blowing (manual, automatic);
- operating gas pressure in dust collectors;
- frequency and duration of blowing.

Theoretical calculation of gas losses (nm3) during blowing is recommended to be determined by the empirical formula:

$$q_{П} = Г_{ф} \cdot V_{K}^{CT} + 3{,}2 \cdot N \cdot n$$

where - gas factor of raw condensate, nm3/m3;

- total amount of stable condensate collected when blowing dust traps, m3;

n - number of dust traps.

N - number of blows of one unit during the period under consideration.

The greatest gas losses occur during manual blowing into an open tank, which leads not only to losses of gas dissolved in condensate, but also to direct consumption of natural gas itself. Application of automatic blowing into a closed tank allows to be limited only by losses of gas of condensate degassing, however quality of blowing itself worsens due to reduction of pressure drop of a drain line.

The blowing frequency depends on the conditionality of the transported gas and has a wide range: from once a week to 8 times a day [20]. The amount of emitted gas at blowing out dust collectors can be reduced if automatic closed systems are used;

c) diffusive emissions, i.e. constant and accidental leaks of natural gas due to equipment leaks. The magnitude of emissions of this type can be determined by direct measurements. Attempts to estimate them by calculation are associated with large inaccuracies in the calculations. Diffusive emissions are characterized by persistent and continuous gas leakages into the atmosphere due to various kinds of unsealed CS fittings and holes (fistholes) in the pipe walls or CS equipment. Although most diffusive leaks on pipeline elements are small emission points, the large number of such sources eventually leads to significant total losses of natural gas.

The occurrence of diffuse natural gas leaks is associated with the presence of leaks:

- in gland and other seals of shut-off valves;
- in butt joints (flanges, threaded joints);
- in areas affected by corrosion;
- in areas with hidden defects and other mechanical damage.

There are several valves on blowers which can be the source of large gas losses. Sources of diffusive gas releases can be various ball valves and gate valves that direct and regulate the gas flow as it passes through compressor station assemblies, gas leaks on safety valves of superchargers, dust collectors and air coolers can also occur. In addition, leakage can occur behind numerous flanges, small welded and threaded joints of pipes available at all nodes of the station. All these assemblies, as a rule, should be inspected during inspection of the compressor station not only by visual methods, but also by portable gas analyzers of methane in atmospheric air.

Work to eliminate and reduce the value of diffusion losses of gas is the most advantageous to perform after calculating the leakage rate for each type of compressor station equipment, which allows to determine where and what measures should be taken in the first place to optimally reduce natural gas losses during its transportation;

d) accidental emissions - losses of natural gas in case of emergency ruptures and other violations of tightness of CS equipment. The value of these losses is assessed on the basis of statistical data for each individual case separately.

Different measures are taken to reduce methane emissions into the atmosphere [22].

- development of new technologies of equipment operation;
- use of compressed air for starting up GCUs;
- application of wireless technologies;
- upkeep of the shutoff valves in a tight condition;
- observance of technological discipline;
- other design and technological solutions.

At that, it is necessary to pay special attention to diffuse type emission, because here are the main reserves of methane emission reduction. To do this, it is necessary to regularly carry out the control search, measurement and elimination of natural gas leaks due to leaks of various equipment of compressor stations.

5.3 Emissions from exhaust gases

In addition to emissions of natural gas (methane) at the compressor stations there are emissions of harmful substances from combustion of fuel at GPA and boiler plants [23]. They include: combustion products - nitrogen, water vapor, carbon dioxide; nitrogen oxides; carbon dioxide; sulfur oxides; hydrocarbons (including not completely burnt methane); soot.

When burning gases containing hydrogen sulfide, sulfur and sulfur dioxide anhydrides, unburned hydrogen sulfide are also emitted into the atmosphere. Quantity of emissions of harmful substances also depends on the type of gas compressor units (Table 5.2). Depending on the natural and climatic conditions of the region and the number of GCUs at compressor stations the harmful impact of emissions extends from 1 to 6 km.

The most harmful impact on the environment is caused by sulfur compounds, carbon monoxide, nitrogen oxides.

Table 5.2.

Value of nominal emissions of harmful substances for different types of GCUs

Turbofan engine	**NO_x nominal, g/m^3**	**CO nominal, г/м^3**
ГТ–700–5	15,00	3,40
ГТК–5	15,00	3,40
ГТ–750–6	23,40	4,0
ГТ–6–750	4,77	7,15
ГТН–6	4,53	6,80
ГПА–ц–6,3	3,87	8,30
ГПА–ц–8	5,03	6,86
ГПА–ц–16	4,44	17,70
ГТК–10	21,90	2,90
ГТНР–10	12,10	2,01
ГПУ–10	3,97	1,70
ГТН–16	7,00	7,79
ГПУ–16	4,60	2,30
ГТН–25	4,58	5,40
ГТН–10И	6,45	1,61
ГТН–25И	4,90	1,02

The main ways to reduce the value of emissions of harmful substances in the composition of exhaust gases are design and technological.

The design ones include, first of all:

- modernization of obsolete equipment, mainly combustion chambers;
- use of burners, providing a more complete combustion of fuel;
- use of filters;
- development of catalysts.

The technological directions include:

- optimization of the combustion process;

- optimization of GPA operation modes, etc.

In practice, design methods are mainly used, aimed mainly at optimizing the combustion process by changing and modernizing combustion chambers.

The main document, regulating and rationing the value of harmful emissions, is a draft MAE (maximum limit emissions). By MAE we mean the mass of a substance, the maximum allowable for emission (g/s, t/year). MPC is set taking into account and based on the maximum allowable concentrations (MAC) of substances. MAE (maximum allowable emissions) projects are developed by specialized design organizations that have the appropriate license and are approved by the local environmental authorities with the issuance of an emission permit.

For each substance polluting the atmospheric air, certain MAC are established in two values - single and average daily MAC. Single MAC is set to prevent reflex reactions in humans during short-term exposure to atmospheric pollution (up to 20 minutes), and the average daily MAC - to prevent their general toxic, mutagenic and other effects.

In accordance with the schedule, approved when the emission permit is issued, but at least once a year, the environmental control services must monitor the compliance of the chemical composition of exhaust gases with the values specified in the draft MAE. If the norms specified in the MAE are exceeded, penalties are imposed on the company in accordance with current legislation.

5.4 Soil Protection

One of the main environmental problems associated with the extraction, transport, storage and processing of natural gas is the disturbance of the soil cover due to mechanical impact and chemical pollution.

Measures are taken to protect the soil [22]:

- minimization of allotments for construction of facilities;
- protection of lands from wind erosion by seeding grasses, bushes and trees;
- reclamation of disturbed lands;
- prevention of land pollution by industrial wastes;
- application of bioindication to determine the level of technogenic pollution.

According to the requirements of environmental legislation, all lands disturbed during the construction cycle are subject to restoration, i.e. reclamation of a set of measures aimed at restoring the economic, medical and aesthetic value of the disturbed landscapes.

Recultivation works are performed in two stages:

- technical - preparation of lands for further target performance in the national economy;

- biological - restoration of fertility which is carried out after the technical stage and includes a set of agrotechnical and phytomeliorative measures aimed at restoration (microbiological composition of soil).

Recultivated lands shall be considered as lands brought to a condition suitable for use in the national economy (agriculture, forestry, water, etc.) and transferred to land users under acts in accordance with the current procedure for transferring recultivated lands to enterprises, organizations and institutions carrying out construction and other works related to disturbance of the soil cover.

Gas industry enterprises, reclamation works are carried out according to the classical scheme: technical stage, biological one. If there is no chemical or other contamination of the soil are limited to the technical stage. In most cases biological reclamation is required, which involves cleaning the soil from pollutants, the main of which are oil products.

In practice the following measures are taken at present in order to re-cultivate the lands contaminated with oil products [14]:

- earthing - filling contaminated areas of soil with earth;
- burning followed by dredging;
- raking and removal of the contaminated soil layer with subsequent burial.

It should be noted that these methods do not contribute to the restoration of soil and vegetation, but rather cause additional damage to nature itself.

The most advanced methods of cleaning soil from petroleum products nowadays are various microbiological methods.

5.5 Noise and other impacts

Noise is a form of physical (wave) pollution of the environment. It arises as a result of oscillating changes in air pressure. In general, it is a chaotic accumulation of sounds of different frequency, power (amplitude) and duration, which go beyond the limits of sound comfort. A distinction is made between noise that is constant, intermittent, periodic, intermittent, pulsed. Noise has a harmful effect on human health, reduces their performance, causes diseases of the hearing (deafness), endocrine, nervous, cardiovascular system (hypertension). Adaptation of an organism to noise is practically impossible, therefore regulation and restriction of noise pollution of environment is an important and obligatory measure [24].

The unit of noise measurement is the physical unit decibel (dB) - the ratio of the effective value of sound pressure to the minimum value perceivable by the human ear.

A suitable soundscape has always existed on Earth, and man has always used the properties of the environment as a conductor, a carrier of sounds. Noise in the natural environment is 30 to 60 decibels. In modern conditions industrial and transport noises are added to the natural background, and their level is often higher than 100 decibels. Sources of noise are all types of transport, industrial facilities, loudspeakers, elevators, televisions, radios, musical instruments, crowds of people, etc.

A hundred years ago, the noise level on the central highways of large cities did not exceed 60 dB.

As admissible norms establish such noise levels [24]. which action for a long time does not cause a decrease in acuity of hearing and provides satisfactory intelligibility of speech at a distance of 1.5 m from the interlocutor. The pain threshold is defined by the intensity of the sound, equal to 140 dB. Wave noise, for example, in a weak wind, is 8-10 dB, normal conversation - 40 dB, the noise near the car - 70-90 dB, the noise from the axial fans - 105 dB. Permissible limit of sound power depending on conditions is 45-85 dB.

At the level of noise background 70 dB disturbances of human endocrine system occur, the number of neuroses and psychoses increases significantly. Long-term exposure to 90 dB of noise can impair hearing, and prolonged exposure to 120 dB of noise causes physical pain and becomes unbearable. Central nervous system disorders, cardiovascular disease, hypertension, mental depression, etc. also occur.

Sound waves with a frequency lower than 16 Hz are perceived by a person not as sound, but as vibration. Vibration causes trembling or shaking of the whole body or its individual parts during various works (concrete laying, pneumatic electric crushing of rocks or track pavement, work with a jackhammer, spraying of materials, etc.). Prolonged vibrations cause great harm to health - from severe fatigue and minor changes in many body functions to concussion, tissue ruptures, cardiac disorders, nervous system disorders, muscle and bone deformities, skin sensitivity disorders, etc.

Noise pollution of the atmosphere is caused by the work of compressor units and the work of transport (road transport, helicopters, etc.), when noise levels exceed health standards, there are serious problems for service personnel [25].

Noise also affects the life of wild animals, birds, creating unfavorable conditions for their habitat. In human habitats noise affects when the distance from the CS to the development is less than 2 - 3 km.

The main way to combat the impact of noise is the use of modern GCUs with effective sound insulation, modernization of existing units in order to reduce noise, as well as the construction of sound reflective screens, forest planting, etc.

Conclusions on Section 5

This section considers the issues of reducing harmful emissions into the atmosphere, and also considers measures for controlling the noise of the gas turbine unit and protecting the grounds from harmful emissions.

Special environmental services have been established to solve current and prospective issues related to environmental protection and performance of control and measuring measures at compressor stations. The main task of the environmental services is to monitor the environmental impact of compressor station operations.

CONCLUSIONS

As a result of the study of methods to assess the influence of GTE operating conditions and its technical condition on the damage accumulation in its turbine parts, the following results have been obtained:

1. In the first chapter the peculiarities of GTE and its turbine operation were analyzed. Component parts and basic characteristics of typical GTE designs have been analyzed and described. The main requirements to the design and peculiarities of GTE turbine operation are described, namely: maximum efficiency; minimum cooling air flow rate; minimum production cost and maintenance cost; ensuring the resource (service life) of the main parts necessary for engine competitiveness.

Physical processes arising in the turbine blade during GTE operation are studied. Without changing the pressure, and hence the value of the relative velocity of the flow (tangibly changing only its direction - the turn of the flow) in the degree of the turbine. With a drop in pressure, an increase in the relative flow velocity and some change in its direction in the degree.

3. In the second section the method of estimation of influence of GTE operation conditions and its technical condition of damage accumulation in its turbine parts is studied. Accumulation of damage in turbine parts, which manifests itself during operation of aviation GTEs, is investigated. Methods of GTE turbine blades protection during operation and methods of their durability improvement are analyzed.

Methods of evaluation of damageability of GTE parts are described. It is defined, that at construction of generalized models of damage processes, their dependence on time and coordinates and other peculiarities of damageability of material should take into account multistage, multicomponent nature.

5. In the third chapter the influence of GTE operating conditions and its technical condition on damage accumulation in its turbine parts is evaluated. General information about PS-90A turbofan engine is described. The thermodynamic and gasdynamic calculation of TFE is performed.

6. The influence of operating and technological factors on damageability is analyzed. Solutions for control actions based on the results of damage parameters monitoring have been suggested, namely: regulation of the automatic control system; flushing and cleaning of the flow part; changes in the GTE operation modes; transfer to the reserve, etc.

7. The analysis of the influence of operational and technological factors on the damage, in particular: the GTU operation mode on the damage; temperature and pressure at the GTU inlet on the damage; parameters of the GTU nodes technical state on the damage As a result, the dependences of the blade temperature, total stress and relative damage of the high pressure rotor rotation speed at variation of operational and technological factors have been obtained.

8. In the section of occupational safety, hazardous and harmful factors were determined when working in the compressor shop, calculation of lightning protection for the heat exchanger was made. Measures on fire and explosion safety during maintenance of gas turbine unit were defined.

9. The environmental protection section considers the issues of reducing harmful emissions into the atmosphere, and also considers measures to control the noise of the gas turbine plant and protection of soils from harmful emissions.

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