МІНІСТЕРСТВО ОСВІТИ І НАУКИ УКРАЇНИ
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V. M. PATLAICHUK, M. V. VASCHILENKO

DESIGN OF GAS TURBINE UNITS

Recommendations for individual task

МЕТОДИЧНІ ВКАЗІВКИ
для виконання індивідуального завдання
з дисципліни
"ПРОЄКТУВАННЯ ГАЗОТУРБІННИХ АГРЕГАТІВ"

Рекомендовано Методичною радою НУК

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Автори: В. М. Патлайчук, кандидат технічних наук, доцент;
М. В. Ващиленко, кандидат технічних наук, доцент

Рецензент О. К. Чередніченко, кандидат технічних наук, доцент

Рекомендовано Методичною радою НУК

Патлайчук В. М.

П20 Design of gas turbine units : Recommendations for individual task : методичні вказівки для виконання індивідуального завдання з дисципліни "Проєктування газотурбінних агрегатів" / В. М. Патлайчук, М. В. Ващиленко. – Миколаїв : НУК, 2020. – 88 с.

Наведені рекомендації щодо виконання індивідуального завдання з дисципліни "Проєктування газотурбінних агрегатів". Послідовно розглянуті його етапи: аналіз конструкції та технічних параметрів базового газотурбінного двигуна, термодинамічне моделювання газотурбінного агрегату, оптимізація параметрів термодинамічного циклу, детальний розрахунок агрегату на номінальному режимі роботи, габаритні розрахунки основних частин газотурбінного агрегату – компресора, камери згоряння і турбіни. Пояснена побудова компоновочного ескізу агрегату.

Призначено для студентів спеціальності 142 "Енергетичне машинобудування", підготовка яких ведеться англійською мовою, а також для тих, хто бажає поглибити власні знання з англійської мови.

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INTRODUCTION

Student performance of an individual task is one of the stages in the study of the discipline "Design of gas turbine units". Its purpose is to teach the student to fulfill calculations of thermodynamic cycles and thermal schemes of gas turbine units (GTU's), as well as teach how to evaluate the dimensions of their main parts – a compressor, a combustion chamber and a turbine.

After completing an individual task, the student:

– must know the thermal processes that occur in compressors, combustion chambers and turbines of gas turbine units;

– must be able to develop the design and fulfill the calculations of gas turbine units;

– must understand the methods of GTU's thermal modeling, understand the methods of thermal calculation, and must know how to evaluate the dimensions of the GTU's main parts.

The theme of an individual task is **designing a simple cycle gas turbine unit for marine application**. As the initial data usually specified:

– output power of gas turbine unit;

– high pressure turbine inlet temperature;

– ambient temperature;

– basic gas turbine engine.

The student must choose all other parameters required for the design by himself, while focusing on the structural scheme of the gas turbine unit, its application and relevant recommendations.

Conditionally, the process of performing an individual task can be divided into the following stages:

– analysis of the design and technical parameters of the basic gas turbine engine;

– thermodynamic modeling of gas turbine unit and the working-out of a calculation program;

– optimization of GTU's thermodynamic cycle parameters;

– thermal calculation of gas turbine unit at nominal operating mode;

– calculation of the compressor dimensions;

– calculation of the combustion chamber dimensions;

– calculation of the turbine dimensions;

– sketching of GTU's layout.

The individual task is carried out sequentially at the indicated stages under the direct supervision of the teacher.

The performed results are formatted as a text document (report) and publicly discussed in the audience during the defense.

1. ANALYSIS OF THE DESIGN AND TECHNICAL PARAMETERS OF THE BASIC GAS TURBINE ENGINE

Gas turbine engines can have a different structural scheme. They can include a different number of compressors (from one to three) and a different number of turbines (from one to four). These compressors and turbines can have different designs and are connected in different ways to each other. The design of the combustion chamber, as well as other components of the gas turbine unit, may also be different.

Among the initial data of an individual task from the discipline "Design of gas turbine units", the basic gas turbine engine is indicated, the structural scheme and design of which must be guided when designing your own gas turbine unit. For this reason, it is necessary to begin the fulfillment of an individual task by searching for relevant information and analyzing it. The most important results of the study are summarized in the first part of the report.

It should be noted that students will be offered as basic engines the marine gas turbine engines of the leading companies in this area: "General Electric Co." (USA), "Rolls-Royce plc" (Great Britain) and "Zorya-Mashproekt" (Ukraine).

The fulfillment of this part of an individual task is presented in Example 1.

Example 1

Perform the analysis of the design and technical parameters of the basic gas turbine engine: UGT 6000 "Zorya-Mashproekt".

The UGT 6000 (Fig. 1) became the first of the third generation engines created by "Zorya-Mashproekt". It was developed in accordance with the decision of the Council of Ministers of the Soviet

Union, adopted in 1971, to create a 10,000 h.p. common engine for hovercraft, hydrofoil and displacement ships.

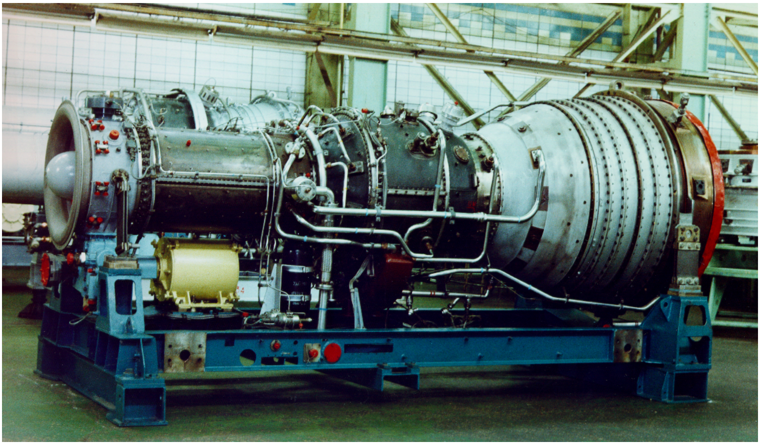

Fig. 1. UGT 6000 (general view)

Since 1972, the UGT 6000 has been widely used on Soviet Union and Russian Federation Navy ships. It is used on landing and anti-submarine hovercraft and hydrofoil ships. It is also used as a marching or accelerator engine in the combined power plants of displacement warships and some commercial vessels.

The industrial use of UGT 6000 began in 1993. It is often used in gas pumping units of compressor pipeline stations. UGT 6000 is also used to drive electric generators.

The UGT 6000 (Fig. 2) consists of a two-spool gas generator and a free power turbine.

The gas generator includes two axial-flow multistage compressors (respectively, low pressure compressor 2 and high pressure compressor 3), which are rotated by two axial-flow turbines (low pressure turbine 6 and high pressure turbine 5), as well as a tubo-annular combustion chamber 4.

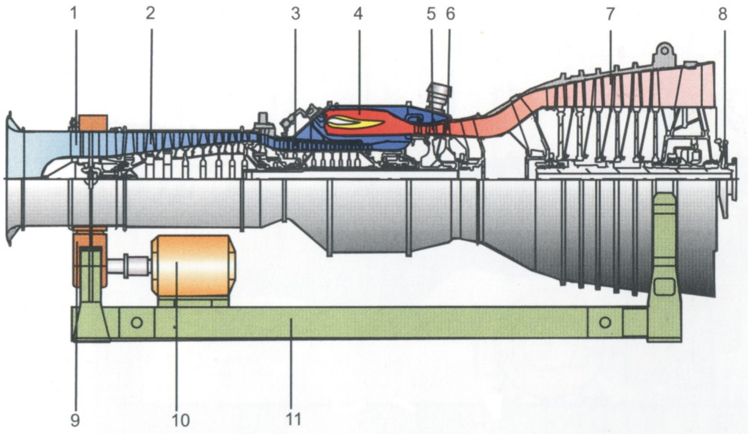

Fig. 2. UGT 6000 (cross-section):
1 – inlet guide vanes; 2 – low pressure compressor; 3 – high pressure compressor; 4 – combustion chamber; 5 – high pressure turbine; 6 – low pressure turbine; 7 – power turbine; 8 – output flange; 9 – gearbox; 10 – electric starter; 11 – frame

The low pressure and high pressure compressors has 8 and 9 stages, respectively. The reverse-flow combustion chamber has 10 flame tubes.

The high pressure turbine and the low pressure turbine are single stage ones. The power turbine, depending on the modification, can have 2, 3, 4 or 6 stages. The rotation of the power turbine is clockwise or counterclockwise.

The lubrication system is made circulating, under pressure (forced).

The engine is started by an 70 kW electric starter. Engine control is remote, automatic.

The time between overhaul is 20,000 hours. The total service life is 60,000 hours.

The main technical parameters of UGT 6000's different modifications are shown in Table 1, service parameters – in Fig. 3.

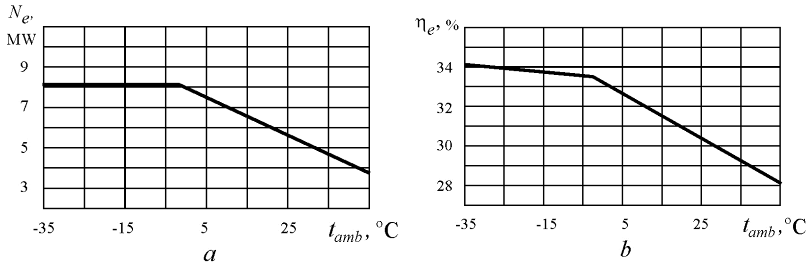

Fig. 3. The output power (*a*) and efficiency (*b*)
of UGT 6000 versus ambient temperature

Table 1

The main parameters of UGT 6000's modifications

Parameter	**Modifications[1]**				
	DT71	**DV71**	**DP71**	**DM71**	**DC71**
Output power, kW	6500	6360	7350	7350	7350
Efficiency, %	31.5	31.5	32.0	32.0	30.0
Air mass flow, kg/s	31	31	32	32	32
Pressure ratio	16.5	16.5	16.5	16.5	16.5
High pressure turbine inlet temperature, K	1290	1290	1290	1290	1290
Exhaust gas temperature, K	703	698	713	713	743
Power turbine rotational speed, rpm	8200[2]	3000	7000	10000	4750
Weight, ton	3.5	4.5	3.5	3.1	3.5
Dimensions, m:					
Length	3.8	4.6	3.2	2.8	3.4
Width	1.3	1.8	1.7	1.7	1.7
Height	1.6	1.7	1.8	1.8	1.8

[1] DT71 – engine modification for gas pumping units; DV71 – engine modification for electricity generation; DP71 and DM71 – marine non-reversible engines; DC71 – marine reversible engine.

[2] Engine modifications for gas pumping units can also have power turbine speed of 5200, 5700 or 10000 rpm.

2. THERMODYNAMIC MODELING OF GAS TURBINE UNIT AND THE WORKING-OUT OF A CALCULATION PROGRAM

2.1. Features of GTU's thermal calculation

2.1.1. Structural schemes of marine gas turbine units

For use on ships, two structural schemes of GTU have become most popular (Fig. 4).

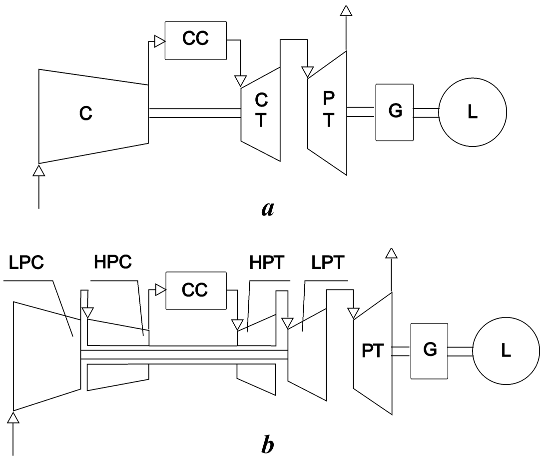

Fig. 4. The structural schemes of marine GTUs:
C – compressor; CC – combustion chamber; CT – compressor turbine; PT – power turbine; G – reduction gear; L – load (consumer); LPC – low pressure compressor; HPC – high pressure compressor; HPT – high pressure turbine; LPT – low pressure turbine

The first structural scheme realised in such marine GTUs as:

– LM500, LM1500, LM2500, LM2500+ and LM2500G4 of "General Electric Co.";

– TF40 and TF50 of "Vericor Power Systems Inc.".

The second structural scheme realised in such marine GTUs as:

– "Olympus", "Tyne", "Spey" and MT30 of "Rolls-Royce plc";

– UGT 3000, UGT 6000, UGT 15000, UGT 16000 and UGT 25000 of "Zorya-Mashproekt".

For further modeling, **we choose the second structural scheme**, because that scheme is more complex.

For this structural scheme, we will use some indexes: $k1$ – low pressure compressor; $k2$ – high pressure compressor; $t1$ – high pressure turbine; $t2$ – low pressure turbine; $t3$ – power turbine.

The cycle in the thermal diagram, that corresponds to this structural scheme, is shown in the Fig. 5.

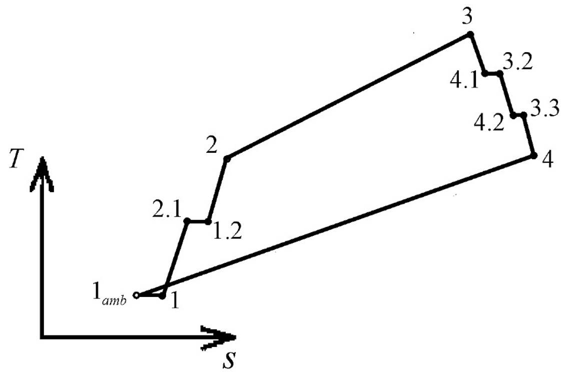

Fig. 5. Cycle of GTU with two-spool gas generator and free power turbine

This cycle consists of the following processes:

process (1_{amb}–1) take place in GTU's inlet casing;

process (1–2.1) – in low pressure compressor;

process (2.1–1.2) – in intersection between LPC and HPC;

process (1.2–2) – in high pressure compressor;

process (2–3) – in combustion chamber;

process (3–4.1) – in high pressure turbine;

process (4.1–3.2) – in intersection between HPT and LPT;
process (3.2–4.2) – in low pressure turbine;
process (4.2–3.3) – in intersection between LPT and PT;
process (3.3–4) – in power turbine;
process ($4–1_{amb}$) – so-called "cooling process".

2.1.2. Determination of air and gas pressure in GTU

We will determine the pressure in different parts of the GTU using **pressure coefficients**. These coefficients consider the pressure losses due to friction in the transitional parts of the GTU.

Pressure coefficient generally defined as

$$\nu = \frac{p_{out}}{p_{in}}.$$

Consider the pressure change along the flowing path of gas turbine unit.

1) We have ambient pressure p_{amb}
(for example, p_{amb}= 0.101 MPa).

2) Pressure at point 1 (at the LPC inlet)

$$p_1 = p_{amb}\nu_{inl},$$

where ν_{inl} – inlet casing pressure coefficient (0.98–0.99).

3) Pressure at point 2.1 (at the LPC outlet)

$$p_{2.1} = p_1\pi_{k1} = p_{amb}\nu_{inl}\pi_{k1},$$

where π_{k1} – LPC pressure ratio.

4) Pressure at point 1.2 (at the HPC inlet)

$$p_{1.2} = p_{2.1}\nu_k = p_{amb}\nu_{inl}\nu_k\pi_{k1},$$

where ν_k – compressor intersection pressure coefficient (0.99–1.00).

5) Pressure at point 2 (at the HPC outlet)

$$p_2 = p_{1.2}\pi_{k2} = p_{amb}\nu_{inl}\nu_k\pi_{k1}\pi_{k2},$$

where π_{k2} – HPC pressure ratio.

6) Pressure at point 3 (at the combustion chamber outlet)

$$p_3 = p_2 \nu_{cc} = p_{amb} \nu_{inl} \nu_k \nu_{cc} \pi_{k1} \pi_{k2},$$

where ν_{cc} – combustion chamber pressure coefficient (0.96–0.98).

7) Pressure at point 4.1 (at the HPT outlet)

$$p_{4.1} = \frac{p_3}{\pi_{t1}} = p_{amb} \nu_{inl} \nu_k \nu_{cc} \frac{\pi_{k1} \pi_{k2}}{\pi_{t1}},$$

where π_{t1} – HPT pressure ratio.

8) Pressure at point 3.2 (at the LPT inlet)

$$p_{3.2} = p_{4.1} \nu_{t1} = p_{amb} \nu_{inl} \nu_k \nu_{cc} \nu_{t1} \frac{\pi_{k1} \pi_{k2}}{\pi_{t1}},$$

where ν_{t1} and ν_{t2} – turbine intersection pressure coefficients (0.99–1.00).

9) Pressure at point 4.2 (at the LPT outlet)

$$p_{4.2} = \frac{p_{3.2}}{\pi_{t2}} = p_{amb} \nu_{inl} \nu_k \nu_{cc} \nu_{t1} \frac{\pi_{k1} \pi_{k2}}{\pi_{t1} \pi_{t2}},$$

where π_{t2} – LPT pressure ratio.

10) Pressure at point 3.3 (at the PT inlet)

$$p_{3.3} = p_{4.2} \nu_{t2} = p_{amb} \nu_{inl} \nu_k \nu_{cc} \nu_{t1} \nu_{t2} \frac{\pi_{k1} \pi_{k2}}{\pi_{t1} \pi_{t2}}.$$

11) Pressure at point 4 (at the PT inlet)

$$p_4 = \frac{p_{3.3}}{\pi_{t3}} = p_{amb} \nu_{inl} \nu_k \nu_{cc} \nu_{t1} \nu_{t2} \frac{\pi_{k1} \pi_{k2}}{\pi_{t1} \pi_{t2} \pi_{t3}}$$

or

$$p_4 = \frac{p_{amb}}{\nu_{ex}},$$

where π_{t3} – PT pressure ratio; ν_{ex} – exhaust casing pressure coefficient (0.98–0.99).

From the last equations it follows that

$$p_{amb}\nu_{inl}\nu_k\nu_{cc}\nu_{t1}\nu_{t2}\frac{\pi_{k1}\pi_{k2}}{\pi_{t1}\pi_{t2}\pi_{t3}} = \frac{p_{amb}}{\nu_{ex}}$$

and

$$\nu_{inl}\nu_k\nu_{cc}\nu_{t1}\nu_{t2}\nu_{ex}\pi_{k1}\pi_{k2} = \pi_{t1}\pi_{t2}\pi_{t3}\,.$$

2.1.3. Determination of compressor and turbine efficiency

If the number of stages of the axial-flow compressor z_k is more than 5, the efficiency of compressor can be estimated by the formula (Fig .6)

$$\eta_k = \frac{\pi_k^m - 1}{\pi_k^{\frac{m}{\eta_{st.k}}} - 1},$$

where $m = \dfrac{k-1}{k}$; k – specific heat ratio; $\eta_{st.k}$ – efficiency of single compressor stage.

The efficiency of single axial-flow compressor stage is 0.89–0.91.

If the number of stages of the axial-flow turbine is more than 5, the efficiency of turbine can be estimated by the formula (Fig. 7)

$$\eta_t = \frac{1-\pi_t^{-m\,\eta_{st.t}}}{1-\pi_t^{-m}},$$

where $m = \dfrac{k-1}{k}$; k – specific heat ratio; $\eta_{st.t}$ – efficiency of single turbine stage.

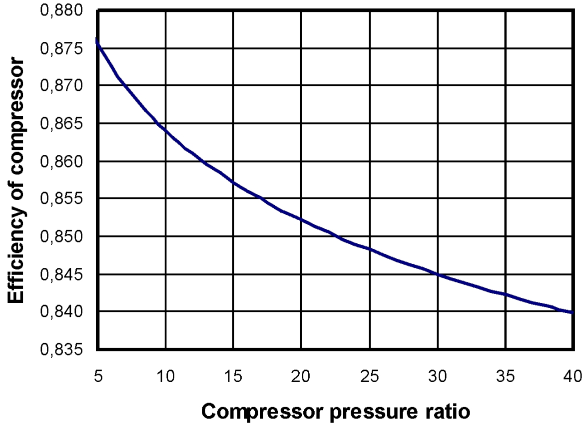

Fig. 6. The efficiency of compressor versus compressor pressure ratio ($\eta_{st.k}$ = 0,90; k = 1,40)

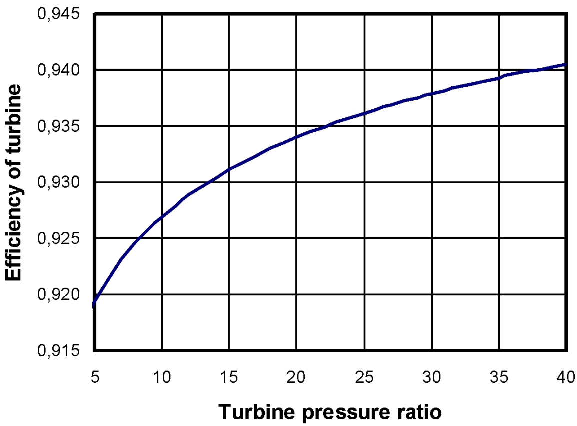

Fig. 7. The efficiency of turbine versus turbine pressure ratio ($\eta_{st.t}$ = 0,90; k = 1,40)

The efficiency of single axial-flow turbine stage usually is 0.86–0.90.

In real conditions, turbines of the GTUs have a number of stages from 1 to 3 and the previous formula cannot be used.

During the thermal calculation of GTU you must take such values of turbines efficiency:

for HPT = 0.86–0.88;
for LPT = 0.88–0.90;
for PT = 0.90–0.91.

2.1.4. Determination of specific heat

The working fluid in the GTU's compressor is air, in the GTU's turbine – gas.

2.1.4.1. Determination of specific heat for air

During the thermal calculation of GTU we need to determine specific heat of constant pressure for air.

Specific heat depends on temperature. During the calculation we will use parameter $c_{pa}\Big|_{T_{in}}^{T_{out}} = f(T_m)$. That is the specific heat of air for process that goes from temperature T_{in} to temperature T_{out}. The specific heat is determined for medium temperature of this process:

$$T_m = \frac{T_{in} + T_{out}}{2}.$$

Next determination of specific heat can go in several ways.

The 1st way: by using Table 2.

(You need to interpolate values of that table.)

Table 2

Specific heat of air and gas (when using liquid fuel[1])

T_m, K	Air	Gas		
		α = 1	α = 3	α = 5
	c_{pa}, kJ/kg/K	c_{pg}, kJ/kg/K	c_{pg}, kJ/kg/K	c_{pg}, kJ/kg/K
273	1.0035	1.055	1.0215	1.0173
373	1.0102	1.0801	1.0383	1.0299
473	1.0245	1.1095	1.0550	1.0467
573	1.0446	1.1388	1.0801	1.0718
673	1.0684	1.1723	1.1053	1.0969
773	1.0923	1.1974	1.1304	1.1220
873	1.1149	1.2309	1.1597	1.1471
973	1.1354	1.2560	1.1806	1.1681
1073	1.1538	1.2853	1.2057	1.1890
1173	1.1702	1.3062	1.2225	1.2057
1273	1.1844	1.3272	1.2351	1.2225
1373	1.1970	1.3439	1.2518	1.2351
1473	1.2083	1.3607	1.2644	1.2476
1573	1.2179	1.3732	1.2769	1.2560
1673	1.2267	1.3855	1.2853	1.2644
1773	1.2346	1.3983	1.2937	1.2769

[1] Table 2 is calculated for fuel whose chemical composition was determined by averaging the parameters of the most common types of liquid fuel used in gas turbine units.

The 2nd way: by using graph.

Using the data from table 2, we can draw the graph of specific heat. Using the graph, we determine the specific heat

$$c_{pa}\Big|_{T_{in}}^{T_{out}} = f(T_m).$$

The 3rd way: by using approximating formula.

Using the data from table 2, we can get approximating formula. Using the formula, we determine the specific heat for air:

$$c_{pa}\Big|_{T_{in}}^{T_{out}} = f(T_m) =$$

$$=0.917747497558594 + 0.000268046831479296 \cdot T_m -$$
$$- 4.88794533737291 \cdot 10^{-8} \cdot T_m T_m.$$

2.1.4.2. Determination of specific heat for gas

Marine gas turbine units operates on liquid fuel (diesel fuel, etc.). Gas for GTU's turbine is the product of combustion of that fuel.

Specific heat of that gas depends on **excess air coefficient in the combustion chamber** α. Therefore, for gas:

$$c_{pg}\Big|_{T_{in}}^{T_{out}} = f(T_m, \alpha).$$

Next steps of determination of specific heat are similar to the previous one.

Approximating formula for specific heat of gas:

$$c_{pg}\Big|_{T_{in}}^{T_{out}} = f(T_m, \alpha) =$$

$$= 1.00549757480621 + 0.000384173501515761 \cdot T_m -$$
$$- 0.0556742288172245 \cdot \alpha -$$
$$- 6.81171030691985 \cdot 10^{-8} \cdot T_m \cdot T_m +$$
$$+ 0.00784802157431841 \cdot \alpha \cdot \alpha -$$
$$- 0.000013345797924557 \cdot T_m \cdot \alpha.$$

2.1.5. Determination of gas constant

Gas constant **for air**

$$R_{air} = 0.287 \text{ kJ/kg/K}.$$

Gas constant **for turbine gas** R_{gas} = 0.288–0.290 kJ/kg/K and depends on excess air coefficient . Approximating formula for it:

$$R = f(\alpha) =$$

$$= 0.291471034288406 - 0.00253063812851906 \cdot \alpha +$$

$$+ 0.000602589745540172 \cdot \alpha \cdot \alpha -$$

$$- 0.0000495026179123 67 \cdot \alpha \cdot \alpha \cdot \alpha.$$

2.1.6. Determination of specific heat ratio

Specific heat ratio both for air and gas determined as

$$k = \frac{c_p}{c_p - R}.$$

2.1.7. Determination of cooling air consumption

To increase the service life, the blades and disks of turbines of modern GTUs are air-cooled. Cooling air is taken from the compressor of the same GTU and is supplied to the turbine through special channels and tubes. For cooling, the turbine blades have internal cavities. After cooling the blades, the air enters the flowing path and mixes with the gas stream.

Without cooling, the turbine blades can withstand the maximum temperature

$$T_{perm} = 1073\text{–}1123 \text{ K}.$$

If the gas temperature before the turbine is less than permissible ($T_3 \le T_{perm}$), then the blades do not need to be cooled.

If $T_3 > T_{perm}$, then number of cooled rows of turbine blades:

$$n_{cool} = \frac{2\lg \dfrac{T_{perm}}{T_3}}{\lg\left[1-(0.11...0.15)\right]}.$$

Number of cooled rows of turbine blades n_{cool} cannot be bigger than the total number of turbine blade rows. HPT and LPT of many GTUs (UGT 6000, UGT 10000, UGT 15000, UGT 25000) has single stage, then the total number of blade rows in that turbines is 2 (one row of stationary nozzle vanes and one row of rotating blades).

Specific cooling air consumption can be estimated by formula

$$g_{cool} = \frac{G_{cool}}{G_k} = a\,k_b\,g_m\,\beta_t\,\frac{(n_{cool}+1)}{2}\times\frac{(T_3 - T_{perm})}{(T_{perm}-T_{cool})}\times\left(\frac{T_3}{T_{cool}}\right)^{0.25},$$

where G_{cool} – cooling air consumption, kg/s; G_k – air mass flow through the compressor, kg/s; T_3 – gas temperature before turbine, K; T_{perm} – permissible blade metal temperature, K; T_{cool} – cooling air temperature (consider equal to the air temperature after the compressor T_2), K; a – coefficient (if $T_3 - T_{perm} \le 100$, then a = 1.7; if $T_3 - T_{perm} > 100$, then a = 1.4–1.3); k_b = = 1.05–1.10 – heat transfer coefficient at the ends of the blades; g_m = 0.020–0.025 – model air consumption; $\beta_t = \dfrac{G_t}{G_k}$ – turbine consumption coefficient; G_t – gas mass flow through the turbine, kg/s.

2.2. Stages of GTU's thermal calculation

2.2.1. Calculation of GTU's compressors

1) Choose the ambient parameters. For standard conditions:
– ambient pressure p_{amb} = 0.101 MPa,
– ambient temperature T_{amb} = 288 K.
2) Pressure at point 1 (at the LPC inlet)

$$p_1 = p_{amb} \nu_{inl},$$

where ν_{inl} – inlet casing pressure coefficient (0.98–0.99).
Temperature at point 1 (at the LPC inlet)

$$T_1 = T_{amb}.$$

3) We **accept with next clarification** the value of total compressor pressure ratio

$$\pi_{k\Sigma}.$$

4) Choose the HPC pressure ratio:

$$\pi_{k2} = \sqrt{\pi_{k\Sigma}},$$

but

$$\pi_{k2} \leq 4.4\text{–}4.6.$$

5) LPC pressure ratio:

$$\pi_{k1} = \frac{\pi_{k\Sigma}}{\pi_{k1}}.$$

6) We **accept with next clarification** the value of specific heat ratio for process in LPC:

$$k'_{a1} = 1.40.$$

7) Parameter

$$m_{a1} = \frac{k'_{a1} - 1}{k'_{a1}}.$$

8) Efficiency of LPC

$$\eta_{k1} = \frac{\pi_{k1}^{m_{a1}} - 1}{\pi_{k1}^{\frac{m_{a1}}{\eta_{st.k}}} - 1},$$

where $\eta_{st.k}$ – efficiency of single compressor stage (0.89–0.91).

9) Actual temperature rise in LPC

$$\Delta T_{k1} = T_1 \left(\pi_{k1}^{m_{a1}} - 1\right) \frac{1}{\eta_{k1}}.$$

10) Temperature at point 2.1 (at the LPC outlet)

$$T_{2.1} = T_1 + \Delta T_{k1}.$$

11) Medium temperature in LPC

$$T_{m1} = \frac{1}{2}\left(T_1 + T_{2.1}\right).$$

12) Determine the specific heat of air in LPC

$$c_{pa1} = f(T_{m1}).$$

and specific heat ratio for process in LPC

$$k_{a1} = \frac{c_{pa1}}{c_{pa1} - R_{air}},$$

where R_{air} = 0.287 kJ/kg/K – gas constant for air.

After this, we **repeat the calculation from point 7 to point 12 using new value of specific heat ratio** (k_{a1} instead of k'_{a1}).

13) Pressure at point 2.1 (at the LPC outlet)

$$p_{2.1} = p_1 \pi_{k1}.$$

14) Pressure at point 1.2 (at the HPC inlet)

$$p_{1.2} = p_{2.1} \nu_k ,$$

where ν_k – compressor intersection pressure coefficient (0.99–1.00).

Temperature at point 1.2 (at the HPC inlet)

$$T_{1.2} = T_{2.1}.$$

15) We **accept with next clarification** the value of specific heat ratio for process in HPC:

$$k'_{a2} = k_{a1}.$$

16) Parameter

$$m_{a2} = \frac{k'_{a2} - 1}{k'_{a2}}.$$

17) Efficiency of HPC

$$\eta_{k2} = \frac{\pi_{k2}^{m_{a2}} - 1}{\pi_{k2}^{\frac{m_{a2}}{\eta_{st.k}}} - 1}.$$

18) Actual temperature rise in HPC

$$\Delta T_{k2} = T_{1.2}\left(\pi_{k2}^{m_{a2}} - 1\right)\frac{1}{\eta_{k2}}.$$

19) Temperature at point 2 (at the HPC oulet)

$$T_2 = T_{1.2} + \Delta T_{k2}.$$

20) Medium temperature in HPC

$$T_{m2} = \frac{1}{2}\left(T_{1.2} + T_2\right).$$

21) Determine the specific heat of air in HPC

$$c_{pa2} = f(T_{m2})$$

and specific heat ratio for process in HPC

$$k_{a2} = \frac{c_{pa2}}{c_{pa2} - R_{air}}.$$

After this, we **repeat the calculation from point 16 to point 21 using new value of specific heat ratio** (k_{a2} instead of k'_{a2}).

22) Pressure at point 2 (at the HPC outlet)

$$p_2 = p_{1.2}\pi_{k2}.$$

2.2.2. Calculation of GTU's combustion chamber

23) Pressure at point 3 (at the combustion chamber outlet)

$$p_3 = p_2 \nu_{cc},$$

where ν_{cc} – combustion chamber pressure coefficient (0.96–0.98).

24) Determine the specific heat of air:

$$c_{pa}\Big|_{293}^{T_2} = f(T_m) \quad \text{and} \quad c_{pa}\Big|_{293}^{T_3} = f(T_m),$$

where T_3 – HPT inlet temperature (initial data for work).

25) Determine the specific heat of "pure" gas:

$$c_{pg}\Big|_{293}^{T_3} = f(T_m, \alpha = 1).$$

26) Choose the fuel parameters: calorific value of fuel Q_f and stoichiometric coefficient L_0.

For standard liquid fuel Q_f = 42915 kJ/kg and L_0 = 14.78 kg/kg;

for natural gas: Q_f = 50032 kJ/kg and L_0 = 17.18 kg/kg.

27) Relative fuel consumption

$$g_f = \frac{c_{pa}\Big|_{293}^{T_3}(T_3 - 293) - c_{pa}\Big|_{293}^{T_2}(T_2 - 293)}{Q_f \eta_{cc} - \left[c_{pg}\Big|_{293}^{T_3}(L_0 + 1) - c_{pa}\Big|_{293}^{T_3} L_0\right](T_3 - 293)},$$

where – combustion chamber efficiency (0.985–0.995).

28) Excess air coefficient in the combustion chamber

$$\alpha = \frac{1}{g_f L_0}.$$

2.2.3. Calculation of GTU's turbines

29) Specific cooling air consumption for HPT

$$g_{cool.1} = a\, k_b\, g_m\, \beta'_{t1} \frac{(n_{cool.1} + 1)}{2} \times \frac{(T_3 - T_{perm})}{(T_{perm} - T_{cool})} \times \left(\frac{T_3}{T_{cool}}\right)^{0.25},$$

where T_{perm} = 1073–1123 K – permissible blade metal temperature for HPT; $T_{cool} = T_2$ – cooling air temperature, K; a – coefficient (if $T_3 - T_{perm} \le 100$, then α = 1.7; if $T_3 - T_{perm} > 100$, then α = 1.4–1.3); k_b = 1.05–1.10 – heat transfer coefficient at the ends of the blades; g_m = 0.020–0.025 – model air consumption; $\beta'_{t1} = \frac{G_{t1}}{G_{k2}} \approx 0.90$ – simplified value of HPT consumption coefficient.

Number of cooled blade rows for HPT:

$$n_{cool.1} = \frac{2\lg \frac{T_{perm}}{T_3}}{\lg\left[1-(0.11...0.15)\right]}.$$

You must remember that:

– the maximum value of $n_{cool.1}$ for single-stage HPT is 2;

– if $T_3 \le T_{perm}$, then HPT blades do not need to be cooled. Then $n_{cool.1} = 0$ and $g_{cool.1} = 0$.

30) Simplified gas temperature at the LPT inlet

$$T'_{3.2} \approx T_3 - 0.9\frac{\Delta T_{k2}}{2}.$$

31) Specific cooling air consumption for LPT

$$g_{cool.2} = a\,k_b\,g_m\,\beta'_{t2}\frac{(n_{cool.2}+1)}{2}\times\frac{\left(T'_{3.2}-T_{perm}\right)}{\left(T_{perm}-T_{cool}\right)}\times\left(\frac{T'_{3.2}}{T_{cool}}\right)^{0.25},$$

where T_{perm} = 1073–1123 K – permissible blade metal temperature for LPT; $T_{cool} = T_2$ – cooling air temperature, K; a – coefficient (if $T'_{3.2} - T_{perm} \le 100$, then a = 1.7; if $T'_{3.2} - T_{perm} > 100$, then a = 1.4–1.3); k_b = 1.05–1.10 – heat transfer coefficient at the ends of the blades; g_m = 0.020–0.025 – model air consumption; $\beta_{t2} = \frac{G_{t2}}{G_{k1}} \approx 0.98$ – simplified value of LPT consumption coefficient.

Number of cooled blade rows for LPT:

$$n_{cool.2} = \frac{2\lg \dfrac{T_{perm}}{T'_{3.2}}}{\lg[1-(0.11...0.15)]}.$$

You must remember that:

– the maximum value of $n_{cool.2}$ for single-stage LPT is 2;

– if $T'_{3.2} \le T_{perm}$, then LPT blades do not need to be cooled. Then $n_{cool.2} = 0$ and $g_{cool.2} = 0$.

32) Combustion chamber consumption coefficient

$$\alpha_{cc} = \frac{G_{cc}}{G_{k1}} = \alpha_{k2} - g_{cool.1} - g_{cool.2} - 0.011,$$

where $\alpha_{k2} = \dfrac{G_{k2}}{G_{k1}} \approx 0.99–1.0$ – HPC consumption coefficient.

33) HPT consumption coefficient

$$\beta_{t1} = \frac{(1+g_f)\alpha_{cc}}{\alpha_{k2}}.$$

34) LPT consumption coefficient

$$\beta_{t2} = \beta_{t1}\alpha_{k2} + g_{cool.1} + 0.006.$$

35) PT consumption coefficient

$$\beta_{t3} = \beta_{t2} + g_{cool.2} + 0.004.$$

36) Exhaust casing consumption coefficient

$$\beta_{ex} = \beta_{t3} + 0.004.$$

37) Medium gas temperature in HPT

$$T_{m1} = T_3 - 0.9\frac{\Delta T_{k2}}{2\eta_{t1}},$$

where η_{t1} = 0.86–0.88 – HPT efficiency.

38) Determine the specific heat of gas for process in HPT:

$$c_{pg1} = f(T_{m1}, \alpha)$$

39) Specific heat ratio of gas for process in HPT

$$k_{g1} = \frac{c_{pg1}}{c_{pg1} - R_{gas}},$$

where R_{gas} = 0.288–0.290 kJ/kg/K – gas constant for turbine gas.

40) Power balance ratio for HPC and HPT

$$C_2 = \frac{c_{pa2}}{\beta_{t1} c_{pg1} \eta_{m1}},$$

where η_{m1} = 0.99–0.995 – mechanical coefficient for HPT.

41) Actual temperature drop in HPT

$$\Delta T_{t1} = C_2 \Delta T_{k2}.$$

42) Simplified gas temperature at the HPT outlet

$$T'_{4.1} = T_3 - \Delta T_{t1}.$$

43) Determine the specific heat of gas for process of mixing cooling air with gas flow in HPT:

$$c_{pgm1} = f(T'_{4.1}, \alpha).$$

44) Determine the specific heat of air for process of mixing cooling air with gas flow in HPT:

$$c_{pam1} = f(T'_{4.1}).$$

45) Gas temperature at the HPT outlet (considers reducing temperature of gas during mixing with cooling air)

$$T_{4.1} = \frac{\alpha_{k2}\beta_{t1}}{(\alpha_{k2}\beta_{t1} + g_{cool.1})} T'_{4.1} + \frac{c_{pam1} g_{cool.1}}{c_{pgm1}(\alpha_{k2}\beta_{t1} + g_{cool.1})} T_{cool} \cdot$$

46) HPT pressure ratio

$$\pi_{t1} = \left(\frac{T_3 \eta_{t1}}{T_3 \eta_{t1} - \Delta T_{t1}} \right)^{\frac{k_{g1}}{k_{g1}-1}} .$$

47) Pressure at point 4.1 (at the HPT outlet)

$$p_{4.1} = \frac{p_3}{\pi_{t1}}.$$

48) Temperature and pressure at point 3.2 (at the LPT inlet)

$$T_{3.2} = T_{4.1} \text{ and } p_{3.2} = p_{4.1}.$$

49) Medium gas temperature in LPT

$$T_{m2} = T_{3.2} - 0.9 \frac{\Delta T_{k1}}{2\eta_{t2}},$$

where η_{t2} = 0.88–0.90 – LPT efficiency.

50) Determine the specific heat of gas for process in LPT:

$$c_{pg2} = f(T_{m2}, \alpha).$$

51) Specific heat ratio of gas for process in LPT

$$k_{g2} = \frac{c_{pg2}}{c_{pg2} - R_{gas}}.$$

52) Power balance ratio for LPC and LPT

$$C_1 = \frac{c_{pa1}}{\beta_{t2} c_{pg2} \eta_{m2}},$$

where η_{m2} = 0.99–0.995 – mechanical coefficient for LPT.

53) Actual temperature drop in LPT

$$\Delta T_{t2} = C_1 \Delta T_{k1}.$$

54) Simplified gas temperature at the LPT outlet

$$T'_{4.2} = T_{3.2} - \Delta T_{t2}$$

55) Determine the specific heat of gas for process of mixing cooling air with gas flow in LPT:

$$c_{pgm2} = f(T'_{4.2}, \alpha).$$

56) Determine the specific heat of air for process of mixing cooling air with gas flow in LPT:

$$c_{pam2} = f(T'_{4.2})$$

57) Gas temperature at the LPT outlet (considers reducing temperature of gas during mixing with cooling air)

$$T_{4.2} = \frac{\beta_{t2}}{(\beta_{t2} + g_{cool.2})} T'_{4.2} + \frac{c_{pam2} g_{cool.2}}{c_{pgm2}(\beta_{t2} + g_{cool.2})} T_{cool}.$$

58) LPT pressure ratio

$$\pi_{t2} = \left(\frac{T_{3.2} \eta_{t2}}{T_{3.2} \eta_{t2} - \Delta T_{t2}} \right)^{\frac{k_{g2}}{k_{g2} - 1}}.$$

59) Pressure at point 4.2 (at the LPT outlet)

$$p_{4.2} = \frac{p_{3.2}}{\pi_{t2}}.$$

60) Temperature and pressure at point 3.3 (at the PT inlet)

$$T_{3.3} = T_{4.2} \text{ and } p_{3.3} = p_{4.2}\nu_t,$$

where ν_t – turbine intersection pressure coefficient (0.99–0.995).

61) Pressure at point 4 (at the PT outlet)

$$p_4 = \frac{p_{amb}}{\nu_{ex}},$$

where ν_{ex} – exhaust casing pressure coefficient (0.98–0.99).

62) PT pressure ratio

$$\pi_{t3} = \frac{p_{3.3}}{p_4}.$$

63) Simplified medium gas temperature in PT

$$T'_{m3} = T_{3.3} - \frac{\Delta T_{k1} + \Delta T_{k2}}{4}.$$

64) Determine the specific heat of gas for process in PT:

$$c_{pg3} = f(T'_{m3}, \alpha).$$

65) Specific heat ratio of gas for process in PT

$$k_{g3} = \frac{c_{pg3}}{c_{pg3} - R_{gas}}.$$

66) Actual temperature drop in PT

$$\Delta T_{t3} = T_{3.3}\left(1 - \frac{1}{\pi_{t3}^{\frac{k_{g3}-1}{k_{g3}}}}\right)\eta_{t3},$$

where η_{t3} = 0.90–0.91 – PT efficiency.

67) Temperature at point 4 (at the PT outlet)

$$T_4 = T_{3.3} - \Delta T_{t3}.$$

68) Medium gas temperature in PT

$$T_{m3} = T_{3.3} - \frac{\Delta T_{t3}}{2\eta_{t3}}.$$

2.2.4. Calculation of GTU's efficiency

69) Specific power

$$N_{sp} = c_{pg3}\Delta T_{t3}\beta_{t3}\nu_{inl}\eta_{m3},$$

where η_{m3} = 0.99–0.995 – mechanical coefficient for PT.

70) Specific air mass flow through LPC

$$G_{k1.sp} = \frac{N_e}{N_{sp}\eta_g},$$

where η_g = 0.97–0.98 – reduction gear efficiency; N_e – output power of GTU (initial data for work), kW.

71) Air mass flow through LPC

$$G_{k1} = G_{k1.sp}\nu_{inl}.$$

72) Air mass flow through HPC

$$G_{k2} = G_{k1}\alpha_{k2}.$$

73) Air mass flow through combustion chamber

$$G_{cc} = G_{k1}\alpha_{cc}.$$

74) Fuel consumption

$$G_f = 3600 \cdot G_{cc} g_f.$$

75) Gas mass flow through HPT

$$G_{t1} = G_{k2}\beta_{t1}.$$

76) Cooling air consumption through HPT

$$G_{cool.1} = G_{k1} g_{cool.1}.$$

77) Gas mass flow through LPT

$$G_{t2} = G_{k1}\beta_{t2}.$$

78) Cooling air consumption through LPT

$$G_{cool.2} = G_{k1} g_{cool.2}.$$

79) Gas mass flow through PT

$$G_{t3} = G_{k1}\beta_{t3}.$$

80) Gas mass flow through exhaust casing

$$G_{ex} = G_{k1}\beta_{ex}.$$

81) Specific fuel consumption

$$g_e = \frac{G_f}{N_e}.$$

82) GTU's efficiency

$$\eta_e = \frac{3600}{g_e Q_f}.$$

2.3. Working-out of a calculation program

During completing an individual task, the student must use the stages of GTU's thermal calculation, described in chapter 2.8, to working-out a calculation program. This program is compiled using software (Microsoft Excel, MathCad, etc.), which is learned by the student and which is available. The compiled calculation program should be debugged and ready for use.

3. OPTIMIZATION OF GTU'S THERMODYNAMIC CYCLE PARAMETERS

The efficiency of the gas turbine units depends on:

– compressor pressure ratio π_k;

– HPT inlet temperature T_3;

– efficiency of compressor η_k, combustion chamber $\eta_{c.c}$ and turbine η_t;

– mechanical coefficient η_m and gear efficiency η_g;

– pressure losses in transitional parts of GTU.

The dependence of GTU's efficiency on the most parameters is simple. If we want to increase the efficiency **we must increase**:

– HPT inlet temperature T_3;

– the efficiency of compressor η_k, combustion chamber and turbine $\eta_{c.c}$;

– the mechanical coefficient η_m and gear efficiency η_g;

and **we must reduce** pressure losses in transitional parts of GTU.

But the dependence of GTU's efficiency on the compressor pressure ratio is not so simple. An increasing of that parameter first causes an increasing of GTU's efficiency, and then a decreasing of efficiency. There is an optimum for compressor pressure ratio π_k^{opt}, at which the GTU's efficiency takes the maximum value $\eta_e^{\max}$ (Fig. 8).

In the previous chapter, we analyzed the formulas for GTU's thermal calculation. And we know that the compressor pressure ratio is initial data for that calculation. Therefore, if in the beginning of calculation we accept the value of compressor pressure ratio closely located to its optimal value, then at the end we get the value of GTU's efficiency almost maximum. On the contrary, if we accept the value of compressor pressure ratio located far from its optimal value, then we get the value of GTU's efficiency very small.

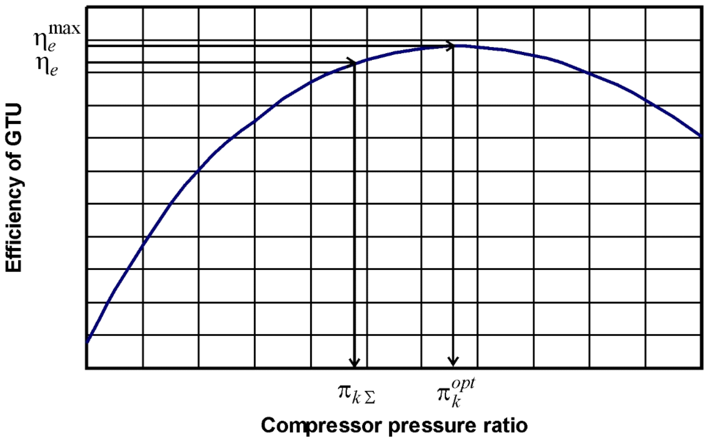

Fig. 8. The efficiency of GTU versus compressor pressure ratio

This is the reason for optimizing the parameters of the GTU's thermodynamic cycle. During this optimization, we must fulfill several thermal calculations of GTU with different initial values of compressor pressure ratio. Based on the calculation results, a graph of the GTU's efficiency versus the compressor pressure ratio is drawn. Using this graph, the optimum value of compressor pressure ratio π_k^{opt} is determined.

Near the optimum value, the graph is very gentle and small deviations from the optimum do not lead to a large decrease in efficiency. For this reason, for the further calculation of GTU, the value of the compressor pressure ratio $\pi_{k\,\Sigma}$ is taken slightly less than optimal.

Smaller values of compressor pressure ratio reduce the number of compressor stages and therefore reduce its complexity, size and weight. Also increases the reliability of the compressor.

But the equation

$$\eta_e^{\max} > \eta_e = f(\pi_{k\Sigma}) > \eta_e^{\max} - 0.005$$

must be fulfilled. Efficiency reduction don't exceed 0.005.

The fulfillment of this part of an individual task is presented in Example 2.

Example 2

Perform the optimization of GTU's thermodynamic cycle parameters for such initial data:

– output power of gas turbine unit N_e = 7 640 kW;

– HPT inlet temperature T_3 = 1500 K;

– ambient temperature T_{amb} = 288 K;

– basic gas turbine engine: UGT 6000 "Zorya-Mashproekt".

At first, we must enter the initial data in the calculation program. Considering the structural scheme and technical parameters of UGT 6000, we set values of the varied parameters (Table 3) and enter that values in the calculation program also.

Table 3

Values of the varied parameters

Parameter	Value
1. Ambient pressure p_{amb}, MPa	0.101
2. Inlet casing pressure coefficient ν_{inl}	0.99
3. Efficiency of single compressor stage $\eta_{st.k}$	0.90
4. Compressor intersection pressure coefficient ν_k	0.99
5. Combustion chamber pressure coefficient ν_{cc}	0.96
6. Calorific value of fuel Q_f, kJ/kg	42 915
7. Stoichiometric coefficient L_0, kg/kg	14.78
8. Combustion chamber efficiency η_{cc}	0.985
9. Permissible blade metal temperature T_{perm}, K	1100
10. Model air consumption g_m	0.022
11. HPC consumption coefficient α_{k2}	0.995

Table 3

Parameter	Value
12. HPT efficiency η_{t1}	0.88
13. Mechanical coefficient for HPT and LPT η_{m1} (η_{m2})	0.995
14. LPT efficiency η_{t2}	0.89
15. Turbine intersection pressure coefficient ν_t	0.995
16. Exhaust casing pressure coefficient ν_{ex}	0.99
17. PT efficiency η_{t3}	0.91
18. Mechanical coefficient for PT η_{m3}	0.99
19. Reduction gear efficiency η_g	0.98

Using the calculation program, we perform several GTU's thermal calculations with different initial values of the compressor pressure ratio π_k (from 10 to 32). The values of GTU's efficiency η_e obtained during the calculations are entered in Table 4.

Table 4

Results of GTU's optimization

π_k	10	11	12	13	14	15	16	17
η_e	0.324	0.331	0.338	0.344	0.349	0.353	0.356	0.360
π_k	18	19	20	21	22	23	24	25
η_e	0.362	0.364	0.365	0.366	0.366	0.367	0.367	0.366
π_k	26	27	28	29	30	31	32	
η_e	0.365	0.364	0.362	0.360	0.358	0.355	0.352	

Based on the optimization results, we draw a graph of the dependence of GTU’s efficiency on the compressor pressure ratio (Fig. 9).

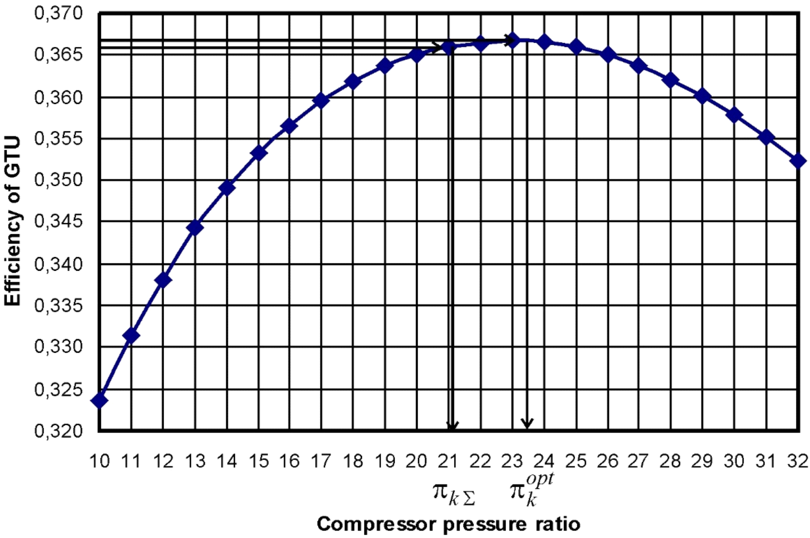

Fig. 9. The dependence of GTU's efficiency on the compressor pressure ratio

We can see that maximum efficiency $\eta_e^{\max} = 0.367$ is achieved with a pressure ratio $\pi_k^{opt} = 23.5$. But from a designing point of view **for further thermal calculation of GTU at nominal operating mode we choose the value of the compressor pressure ratio** $\pi_{k\Sigma}$ **= 21**. In this case, the efficiency of GTU will be

$$\eta_e = f(\pi_{k\Sigma}) = 0.366.$$

The efficiency difference is very small

$$\Delta\eta_e = \eta_e^{\max} - \eta_e = 0.367 - 0.366 = 0.001,$$

but the value of pressure ratio was reduced significantly (from 23.5 to 21). We know that smaller values of compressor pressure ratio reduce the number of compressor stages and therefore reduce its complexity, size and weight. Also increases the reliability of the compressor.

4. THERMAL CALCULATION OF GAS TURBINE UNIT AT NOMINAL OPERATING MODE

Thermal calculation of GTU at nominal operating mode is carried out in order to find the values of parameters and consumptions of the working fluid for its main parts: compressors, combustion chamber and turbines. These data are needed to fulfill the calculations of the mass-dimensional characteristics of the gas turbine unit, as well as the thermodynamic calculations of these parts. Also at this step the parameters of GTU's efficiency and specific power are determined.

The fulfillment of this part of an individual task is presented in Example 3.

Example 3

Perform the thermal calculation of GTU at nominal operating mode for initial data and calculation results of Example 2.

The results of thermal calculation are summarized in Table 5.

Table 5

Thermal calculation of GTU at nominal operating mode

Parameter	Formula	Result
1. Ambient pressure p_{amb}, MPa	Accepted	0.101
2. Ambient temperature T_{amb}, K	Initial data for work	**288**
3. Inlet casing pressure coefficient ν_{inl}	0.98–0.99	0.99
4. Pressure at point 1 (at the LPC inlet) p_1, MPa	$p_{amb}\nu_{inl}$	0.100
5. Temperature at point 1 (at the LPC inlet) T_1, K	Equal to T_{amb}	288

Table 5

Parameter	Formula	Result
6. Compressor pressure ratio $\pi_{k\Sigma}$	See Example 2	**21**
7. HPC pressure ratio π_{k2}	$\sqrt{\pi_{k\Sigma}}$, but $\pi_{k2} \le 4.4$–4.6	4.50
8. LPC pressure ratio π_{k1}	$\dfrac{\pi_{k\Sigma}}{\pi_{k1}}$	4.667
9. Specific heat ratio for process in LPC k'_{a1}	In the first approximation is equal to 1.40	1.40
10. Parameter m_{a1}	$\dfrac{k'_{a1}-1}{k'_{a1}}$	0.286
11. Efficiency of single compressor stage $\eta_{st.k}$	0.89–0.91	0.90
12. Efficiency of LPC η_{k1}	$\dfrac{\pi_{k1}^{m_{a1}}-1}{\pi_{k1}^{\frac{m_{a1}}{\eta_{st.k}}}-1}$	0.877
13. Actual temperature rise in LPC ΔT_{k1}, K	$T_1\left(\pi_{k1}^{m_{a1}}-1\right)\dfrac{1}{\eta_{k1}}$	181.7
14. Temperature at point 2.1 (at the LPC outlet) $T_{2.1}$, K	$T_1+\Delta T_{k1}$	469.7
15. Medium temperature in LPC T_{m1}, K	$\dfrac{1}{2}\left(T_1+T_{2.1}\right)$	378.8
16. Specific heat of air in LPC c_{pa1}, kJ/kg/K	$f(T_{m1})$	1.012
17. Gas constant of air R_{air}, kJ/kg/K	Equal to 0.287	0.287
18. Specific heat ratio for process in LPC k_{a1}	$\dfrac{c_{pa1}}{c_{pa1}-R_{air}}$	1.396

Table 5

Parameter	Formula	Result
19. Check of result	If $\left\lvert \frac{k_{a1}-k'_{a1}}{k_{a1}} \right\rvert > 0.01$, then $k'_{a1}=k_{a1}$ and you must return to step 10	0.003
20. Pressure at point 2.1 (at the LPC outlet) $p_{2.1}$, MPa	$p_1 \pi_{k1}$	0.467
21. Compressor intersection pressure coefficient ν_k	0.99–1.0	0.99
22. Pressure at point 1.2 (at the HPC inlet) $p_{1.2}$, MPa	$p_{2.1} \nu_k$	0.462
23. Temperature at point 1.2 (at the HPC inlet) $T_{1.2}$, K	Equal to $T_{2.1}$	469.7
24. Specific heat ratio for process in HPC k'_{a2}	In the first approximation is equal to k_{a1}	1.37
25. Parameter m_{a2}	$\frac{k'_{a2}-1}{k'_{a2}}$	0.270
26. Efficiency of HPC η_{k2}	$\frac{\pi_{k2}^{m_{a2}}-1}{\pi_{k2}^{\frac{m_{a2}}{\eta_{st.k}}}-1}$	0.879
27. Actual temperature rise in HPC ΔT_{k2}, K	$T_{1.2}\left(\pi_{k2}^{m_{a2}}-1\right)\frac{1}{\eta_{k2}}$	267.9
28. Temperature at point 2 (at the HPC outlet) T_2, K	$T_{1.2}+\Delta T_{k2}$	737.6
29. Medium temperature in HPC T_{m2}, K	$\frac{1}{2}\left(T_{1.2}+T_2\right)$	603.6
30. Specific heat of air in HPC c_{pa2}, kJ/kg/K	$f(T_{m2})$	1.062
31. Specific heat ratio for process in HPC k_{a2}	$\frac{c_{pa2}}{c_{pa2}-R_{air}}$	1.370

Table 5

Parameter	Formula	Result
32. Check of result	If $\left\|\frac{k_{a2}-k'_{a2}}{k_{a2}}\right\|>0.01$, then $k'_{a2}=k_{a2}$ and you must return to step 25	0.0003
33. Pressure at point 2 (at the HPC outlet) p_2, MPa	$p_{1.2}\pi_{k2}$	2.079
34. Combustion chamber pressure coefficient ν_{cc}	0.96–0.98	0.96
35. HPT inlet pressure p_3, MPa	$p_2\nu_{cc}$	1.996
36. HPT inlet temperature T_3, K	Initial data for work	**1500**
37. Specific heat of air $c_{pa}\Big\|_{293}^{T_2}$, kJ/kg/K	$f(T_m=\frac{T_2+293}{2})$	1.043
38. Specific heat of air $c_{pa}\Big\|_{293}^{T_3}$, kJ/kg/K	$f(T_m=\frac{T_3+293}{2})$	1.119
39. Specific heat of "pure" gas $c_{pg}\Big\|_{293}^{T_3}$, kJ/kg/K	$f(T_m=\frac{T_3+293}{2},\alpha=1)$	1.235
40. Calorific value of fuel Q_f, kJ/kg	Accepted	42 915
41. Stoichiometric coefficient L_0, kg/kg	Accepted	14.78
42. Combustion chamber efficiency η_{cc}	0.985–0.995	0.985
43. Relative fuel consumption g_f	$\dfrac{c_{pa}\Big\|_{293}^{T_3}(T_3-293)-c_{pa}\Big\|_{293}^{T_2}(T_2-293)}{Q_f\eta_{cc}-\left[c_{pg}\Big\|_{293}^{T_3}(L_0+1)-c_{pa}\Big\|_{293}^{T_3}L_0\right](T_3-293)}$	0.0229

Table 5

Parameter	Formula	Result
44. Excess air coefficient in the combustion chamber α	$\frac{1}{g_f L_0}$	2.953
45. Permissible blade metal temperature for HPT T_{perm}, K	1073–1123	1100
46. Cooling air temperature T_{cool}, K	Equal to T_2	737.6
47. Coefficient a	If $T_3 - T_{perm} \le 100$, then a = 1.7; if $T_3 - T_{perm} > 100$, then a = 1.4–1.3	1.30
48. Heat transfer coefficient at the ends of the blades k_b	1.05–1.10	1.05
49. Model air consumption g_m	0.020–0.025	0.022
50. Simplified value of HPT consumption coefficient β'_{t1}	Equal to 0.90	0.90
51. Number of cooled blade rows for HPT $n_{cool.1}$	$\frac{2\lg\frac{T_{perm}}{T_3}}{\lg[1-(0.11...0.15)]}$	2
52. Specific cooling air consumption for HPT $g_{cool.1}$[1]	$a k_b g_m \beta'_{t1} \frac{(n_{cool.1}+1)}{2} \times$ $\times \frac{(T_3 - T_{perm})}{(T_{perm} - T_{cool})} \left(\frac{T_3}{T_{cool}}\right)^{0.25}$	0.0534

[1] You must remember that:
– the maximum value of $n_{cool.1}$ for single-stage HPT is 2;
– if $T_3 \le T_{perm}$, then HPT blades do not need to be cooled. Then $n_{cool.1} = 0$ and $g_{cool.1} = 0$.

Table 5

Parameter	Formula	Result
53. Simplified gas temperature at the LPT inlet $T'_{3.2}$, K	$T_3 - 0.9 \cdot \frac{\Delta T_{k2}}{2}$	1379.4
54. Permissible blade metal temperature for LPT T_{perm}, K	1073–1123	1100
55. Coefficient a	If $T'_{3.2} - T_{perm} \leq 100$, then a = 1.7; if $T'_{3.2} - T_{perm} > 100$, then a = 1.4–1.3	1.30
56. Simplified value of LPT consumption coefficient β'_{t2}	Equal to 0.98	0.98
57. Number of cooled blade rows for LPT $n_{cool.2}$	$\frac{2\lg \frac{T_{perm}}{T'_{3.2}}}{\lg[1-(0.11...0.15)]}$	2
58. Specific cooling air consumption for LPT g_{cool2}[2]	$a\,k_b\,g_m\,\beta'_{t2}\frac{(n_{cool.2}+1)}{2} \times \times \frac{(T'_{3.2}-T_{perm})}{(T_{perm}-T_{cool})}\left(\frac{T'_{3.2}}{T_{cool}}\right)^{0.25}$	0.0398
59. HPC consumption coefficient α_{k2}	0.99–1.0	0.995
60. Combustion chamber consumption coefficient α_{cc}	$\alpha_{k2} - g_{cool.1} - g_{cool.2} - 0.011$	0.891
61. HPT consumption coefficient β_{t1}	$\frac{(1+g_f)\alpha_{cc}}{\alpha_{k2}}$	0.916
62. LPT consumption coefficient β_{t2}	$\beta_{t1}\alpha_{k2} + g_{cool.1} + 0.006$	0.971

[2]You must remember that:

– the maximum value of n_{cool2} for single-stage HPT is 2;

– if $T'_{3.2} \leq T_{perm}$, then HPT blades do not need to be cooled. Then $n_{cool.2} = 0$ and $g_{cool.2} = 0$.

Table 5

Parameter	Formula	Result
63. PT consumption coefficient β_{t3}	$\beta_{t2} + g_{cool.2} + 0.004$	1.014
64. Exhaust casing consumption coefficient β_{ex}	$\beta_{t3} + 0.004$	1.018
65. HPT efficiency η_{t1}	0.86–0.88	0.88
66. Medium gas temperature in HPT T_{m1}, K	$T_3 - 0.9\frac{\Delta T_{k2}}{2\eta_{t1}}$	1363.0
67. Specific heat of gas for process in HPT c_{pg1}, kJ/kg/K	$f(T_{m1}, \alpha)$	1.253
68. Gas constant of turbine gas R_{gas}, kJ/kg/K	0.288–0.290	0.288
69. Specific heat ratio of gas for process in HPT k_{g1}	$\frac{c_{pg1}}{c_{pg1} - R_{gas}}$	1.298
70. Mechanical coefficient for HPT η_{m1}	0.99–0.995	0.995
71. Power balance ratio for HPC and HPT C_2	$\frac{c_{pa2}}{\beta_{t1} c_{pg1} \eta_{m1}}$	0.930
72. Actuak temperature drop in HPT ΔT_{t1}, K	$C_2 \Delta T_{k2}$	249.2
73. Simplified gas temperature at the HPT outlet $T'_{4.1}$, K	$T_3 - \Delta T_{t1}$	1250.8
74. Specific heat of gas for process of mixing colling air with gas flow in HPT c_{pgm1}, kJ/kg/K	$f(T'_{4.1}, \alpha)$	1.234
75. Specific heat of air for process of mixing colling air with gas flow in HPT c_{pam1}, kJ/kg/K	$f(T'_{4.1})$	1.177

Table 5

Parameter	Formula	Result
76. Gas temperature at the HPT outlet $T_{4.1}$, K	$\frac{\alpha_{k2}\beta_{t1}}{(\alpha_{k2}\beta_{t1}+g_{cool.1})}T'_{4.1}+$ $+\frac{c_{pam1}g_{cool.1}}{c_{pgm1}(\alpha_{k2}\beta_{t1}+g_{cool.1})}T_{cool}$	1220.5
77. HPT pressure ratio π_{t1}	$\left(\frac{T_3\eta_{t1}}{T_3\eta_{t1}-\Delta T_{t1}}\right)^{\frac{k_{g1}}{k_{g1}-1}}$	2.485
78. Pressure at point 4.1 (at the HPT outlet) $p_{4.1}$, MPa	$\frac{p_3}{\pi_{t1}}$	0.803
79. Temperature at point 3.2 (at the LPT inlet) $T_{3.2}$, K	Equal to $T_{4.1}$	1220.5
80. Pressure at point 3.2 (at the LPT inlet) $p_{3.2}$, MPa	Equal to $p_{4.1}$	0.803
81. LPT efficiency η_{t2}	0.88–0.90	0.89
82. Medium gas temperature in LPT T_{m2} , K	$T_{3.2}-0.9\frac{\Delta T_{k1}}{2\eta_{t2}}$	1128.7
83. Specific heat of gas for process in LPT c_{pg2}, kJ/kg/K	$f(T_{m2},\alpha)$	1.212
84. Specific heat ratio of gas for process in LPT k_{g2}	$\frac{c_{pg2}}{c_{pg2}-R_{gas}}$	1.312
85. Mechanical coefficient for LPT η_{m2}	0.99–0.995	0.995
86. Power balance ratio for LPC and LPT C_1	$\frac{c_{pa1}}{\beta_{t2}c_{pg2}\eta_{m2}}$	0.865
87. Actual temperature drop in LPT ΔT_{t2}, K	$C_1\Delta T_{k1}$	157.1
88. Simplified gas temperature at the LPT outlet $T'_{4.2}$, K	$T_{3.2}-\Delta T_{t2}$	1063.4

Table 5

Parameter	Formula	Result
89. Specific heat of gas for process of mixing colling air with gas flow in LPT $c_{p\,gm2}$, kJ/kg/K	$f(T'_{4.2}, \alpha)$	1.199
90. Specific heat of air for process of mixing colling air with gas flow in LPT $c_{p\,am2}$, kJ/kg/K	$f(T'_{4.2})$	1.148
91. Gas temperature at the LPT outlet $T_{4.2}$, K	$\frac{\beta_{t2}}{(\beta_{t2} + g_{cool.2})} T'_{4.2} + \frac{c_{p\,am2} g_{cool.2}}{c_{p\,gm2}(\beta_{t2} + g_{cool.2})} T_{cool}$	1049.3
92. LPT pressure ratio π_{t2}	$\left(\frac{T_{3.2}\eta_{t2}}{T_{3.2}\eta_{t2} - \Delta T_{t2}} \right)^{\frac{k_{g2}}{k_{g2}-1}}$	1.930
93. Pressure at point 4.2 (at the LPT outlet) $p_{4.2}$, MPa	$\frac{p_{3.2}}{\pi_{t2}}$	0.416
94. Turbine intersection pressure coefficient ν_t	0.99–0.995	0.995
95. Temperature at point 3.3 (at the PT inlet) $T_{3.3}$, K	Equal to $T_{4.2}$	1049.3
96. Pressure at point 3.3 (at the PT inlet) $p_{3.3}$, MPa	$p_{4.2}\nu_t$	0.414
97. Exhaust casing pressure coefficient ν_{ex}	0.98–0.99	0.99
98. Pressure at point 4 (at the PT outlet) p_4, MPa	$\frac{p_{amb}}{\nu_{ex}}$	0.102
99. PT pressure ratio π_{t3}	$\frac{p_{3.3}}{p_4}$	4.059
100. Simplified medium gas temperature in PT T'_{m3}, K	$T_{3.3} - \frac{\Delta T_{k1} + \Delta T_{k2}}{4}$	936.9

Table 5

Parameter	Formula	Result
101. Specific heat of gas for process in PT c_{pg3}, kJ/kg/K	$f(T'_{m3}, \alpha)$	1.173
102. Specific heat ratio of gas for process in PT k_{g3}	$\frac{c_{pg3}}{c_{pg3} - R_{gas}}$	1.325
103. PT efficiency η_{t3}	0.90–0.91	0.91
104. Actual temperature drop in PT ΔT_{t3}, K	$T_{3.3}\left(1 - \frac{1}{\pi_{t3}^{\frac{k_{g3}-1}{k_{g3}}}}\right)\eta_{t3}$	278.0
105. Temperature at point 4 (at the PT outlet) T_4, K	$T_{3.3} - \Delta T_{t3}$	771.4
106. Medium gas temperature in PT T_{m3}, K	$T_{3.3} - \frac{\Delta T_{t3}}{2\eta_{t3}}$	896.6
107. Mechanical coefficient for PT η_{m3}	0.99–0.995	0.99
108. Specific power N_{sp}, kW·s/kg	$c_{pg3}\Delta T_{t3}\beta_{t3}\nu_{inl}\eta_{m3}$	324.1
109. Reduction gear efficiency η_g	0.97–0.98	0.98
110. Output power of GTU N_e, kW	Initial data for work	**7640**
111. Specific air mass flow through LPC $G_{k1.sp}$, kg/s	$\frac{N_e}{N_{sp}\eta_g}$	24.1
112. Air mass flow through LPC G_{k1}, kg/s	$G_{k1.sp}\nu_{inl}$	23.8
113. Air mass flow through HPC G_{k2}, kg/s	$G_{k1}\alpha_{k2}$	23.7
114. Air mass flow through combustion chamber G_{cc}, kg/s	$G_{k1}\alpha_{cc}$	21.2

Table 5

Parameter	Formula	Result
115. Fuel consumption G_f, kg/hr	$3600 \cdot G_{cc} g_f$	1749.8
116. Gas mass flow through HPT G_{t1}, kg/s	$G_{k2}\beta_{t1}$	21.7
117. Cooling air consumption through HPT $G_{cool.1}$, kg/s	$G_{k1} g_{cool.1}$	1.3
118. Gas mass flow through LPT G_{t2}, kg/s	$G_{k1}\beta_{t2}$	23.1
119. Cooling air consumption through LPT $G_{cool.2}$, kg/s	$G_{k1} g_{cool.2}$	0.9
120. Gas mass flow through PT G_{t3}, kg/s	$G_{k1}\beta_{t3}$	24.2
121. Gas mass flow through exhaust casing G_{ex}, kg/s	$G_{k1}\beta_{ex}$	24.3
122. Specific fuel consumption g_e, kg/kW/hr	$\frac{G_f}{N_e}$	0.229
123. GTU's efficiency η_e	$\frac{3600}{g_e Q_f}$	0.366

5. CALCULATION OF THE COMPRESSOR DIMENSIONS

The function of the GTU's compressor part is to increase the pressure of the working fluid (atmospheric air). The compressor is driven by the power from the turbine of the same GTU.

The gas turbine units use the compressors of high efficiency and mass flow. These are such types of vane compressors as:

– axial-flow compressors;
– centrifugal-flow compressors;
– axial-centrifugal-flow compressors.

Currently the dominant type of GTU's compressor is an axial-flow multistage compressor. The use of the remaining types is limited only by some narrow fields of application in which the use of an axial-flow compressor is undesirable for one reason or another.

Consider the determination of the main dimensions of the GTU's compressor part based on axial-flow multistage compressors.

First of all, the radial and axial dimensions of low pressure compressor (LPC) and high pressure compressor (HPC) will be determined. The number of compressors stages and the speed of their rotors will be estimated also. As a result of these calculations, data will be obtained to sketch the layout of GTU's compressor part.

When designing the GTU and sketching its layout, the dimensions of compressor transitional parts (inlet casing, front casing, intersection, rear casing) are determined, focusing on the design of the existing marine GTUs. During designing care should be taken to ensure that the direction of the flow line has no sharp fractures. Sudden changes in the flow direction caused by significant fluctuations of the medium diameters of the parts lead to increased losses of pressure and, as a consequence, to decreased efficiency of the GTU.

The dimensions of the GTU's compressors are calculated sequentially, starting with LPC. The scheme of low pressure compressor is shown in Fig. 10. The steps for calculating LPC dimensions are given below.

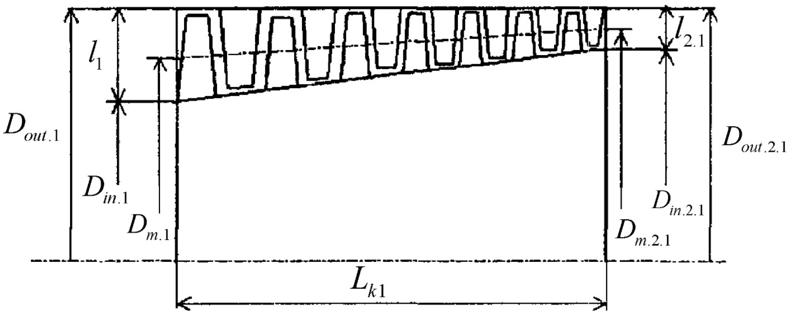

Fig. 10. The scheme of low pressure compressor

1) The following parameters were determined during thermal calculation of GTU at nominal operating mode:

– air mass flow through LPC G_{k1};

– temperature at point 1 (at the LPC inlet) T_1;

– pressure at point 1 (at the LPC inlet) p_1.

2) Specific heat of air at the LPC inlet is determined depending on the temperature T_1:

$$c_{pa1} = f(T_1).$$

3) Specific heat ratio for air at the LPC inlet

$$k_{a1} = \frac{c_{pa1}}{c_{pa1} - R_{air}},$$

where R_{air} = 0.287 kJ/kg/K – gas constant of air.

4) Circumferential velocity at outer diameter of the LPC inlet is equal to:

$$u_{out.1} = 270\text{–}340 \text{ m/s}.$$

(If we calculates HPC, then $u_{out.1.2}$ = 240–340 m/s.)

5) Axial velocity at the LPC inlet

$$c_{a1} = \left(0.45...0.55\right) u_{out.1}.$$

6) Specific flow velocity at the LPC inlet

$$\lambda_{c_{a1}} = \frac{c_{a1}}{\sqrt{\frac{2\,k_{a1}}{k_{a1}+1} R_{air}\, T_1 \cdot 10^3}}.$$

7) Gas-dynamic flow density function at the LPC inlet

$$q(\lambda_{c_{a1}}) = \lambda_{c_{a1}} \left[\frac{k_{a1}+1}{2} \left(1 - \frac{k_{a1}-1}{k_{a1}+1} \lambda^2_{c_{a1}} \right) \right]^{\frac{1}{k_{a1}-1}}.$$

8) Required flow area at the LPC inlet

$$F_1 = \frac{G_{k1}\sqrt{T_1}}{\sqrt{\left(\frac{2}{k_{a1}+1}\right)^{\frac{k_{a1}+1}{k_{a1}-1}} \cdot \frac{k_{a1}}{R_{air}\cdot 10^3}}\; p_1\, q(\lambda_{c_{a1}})\cdot 10^6}.$$

9) Diameter ratio for LPC is equal to:

$$\bar{d}_1 = \frac{D_{in.1}}{D_{out.1}} = 0.45–0.55.$$

(If we calculates HPC, then $\bar{d}_{1.2}$ = 0.70–0.85.)
10) Outer diameter at the LPC inlet

$$D_{out.1} = \sqrt{\frac{4F_1}{\pi\left(1-\bar{d}_1^2\right)}}.$$

11) LPC rotor speed

$$n_{k1} = \frac{60\,u_{out.1}}{\pi\,D_{out.1}}.$$

(The LPC rotor speed must be less than HPC rotor speed. The equation $n_{k1} < n_{k2}$ must be executed.)

12) Inner diameter at the LPC inlet

$$D_{in.1} = D_{out.1}\bar{d}_1.$$

13) Length of the first stage blades

$$l_1 = 0.5\left(D_{out.1} - D_{in.1}\right).$$

14) Medium diameter at the LPC inlet

$$D_{m.1} = 0.5\left(D_{out.1} + D_{in.1}\right).$$

15) The following parameters were determined during thermal calculation of GTU at nominal operating mode:

– pressure at point 2.1 (at the LPC outlet) $p_{2.1}$;

– temperature at point 2.1 (at the LPC outlet) $T_{2.1}$.

16) Specific heat of air at the LPC outlet

$$c_{pa2.1} = f(T_{2.1}).$$

17) Specific heat ratio for air at the LPC outlet

$$k_{a2.1} = \frac{c_{pa2.1}}{c_{pa2.1} - R_{air}}.$$

18) Axial velocity at the LPC outlet

$$c_{a2.1} = (0.90...0.95)c_{a1}.$$

19) Specific flow velocity at the LPC outlet

$$\lambda_{c_{a2.1}} = \frac{c_{a2.1}}{\sqrt{\frac{2\,k_{a2.1}}{k_{a2.1}+1} R_{air}\, T_{2.1} \cdot 10^3}}.$$

20) Gas-dynamic flow density function at the LPC outlet

$$q(\lambda_{c_{a2.1}}) = \lambda_{c_{a2.1}} \left[\frac{k_{a2.1}+1}{2} \left(1 - \frac{k_{a2.1}-1}{k_{a2.1}+1} \lambda^2_{c_{a2.1}} \right) \right]^{\frac{1}{k_{a2.1}-1}}.$$

21) Required flow area at the LPC outlet

$$F_{2.1} = \frac{G_{k1}\sqrt{T_{2.1}}}{\sqrt{\left(\frac{2}{k_{a2.1}+1}\right)^{\frac{k_{a2.1}+1}{k_{a2.1}-1}} \cdot \frac{k_{a2.1}}{R_{air} \cdot 10^3}}\; p_{2.1}\, q(\lambda_{c_{a2.1}}) \cdot 10^6}.$$

22) Choosing the compressor stages layout. For LPC this is a layout with a constant outer diameter $D_{out} = const$. Then outer diameter at the LPC outlet:

$$D_{out.2.1} = D_{out.1}.$$

(For HPC we choose a layout with a constant inner diameter $D_{in} = const$ and inner diameter at the LPC outlet $D_{in.2} = D_{in.1.2}$.)

23) Inner diameter at the LPC outlet

$$D_{in.2.1} = \sqrt{D^2_{out.2.1} - \frac{4}{\pi} F_{2.1}}.$$

(For HPC we calculate outer diameter at the LPC outlet $D_{out.2} = \sqrt{D_{in.2}^2 + \frac{4}{\pi} F_2}$.)

24) Length of the last stage blades

$$l_{2.1} = 0.5\left(D_{out.2.1} - D_{in.2.1}\right).$$

25) Medium diameter at the LPC outlet

$$D_{m.2.1} = 0.5\left(D_{out.2.1} + D_{in.2.1}\right).$$

26) The LPC pressure ratio π_{k1} was determined during thermal calculation of GTU at nominal operating mode.

27) Pressure ratio of single LPC stage is equal to:

$$\pi_{st.k} = 1.20\text{–}1.30.$$

(For HPC $\pi_{st.k}$ = 1.15–1.20.)

28) Number of LPC stages (round to integer value)

$$z_{k1} = \frac{\lg \pi_{k1}}{\lg \pi_{st.k}}.$$

29) Medium LPC diameter

$$D_{mk} = 0.5(D_{m1} + D_{m2.1}).$$

30) Medium length of LPC blades

$$l_{mk} = 0.5(l_1 + l_{2.1}).$$

31) Ratio

$$\lambda_k = \frac{D_{mk}}{l_{mk}}.$$

32) Coefficient $\overline{\delta}_k = f(\lambda_k)$ is determined using Table 6.

Table 6

Values of coefficient $\overline{\delta}_k$

λ_k	4	5	6	7	8	9	10	11	12	13	14	15	16	17	≥18
$\overline{\delta}_k$	0.91	1.02	1.12	1.21	1.29	1.37	1.47	1.54	1.60	1.65	1.71	1.75	1.81	1.85	1.88

33) Axial dimension of single LPC stage

$$b_{st.k} = \overline{\delta}_k l_{mk} .$$

34) Axial dimension of LPC main casing (without length of inlet casing, front casing and rear casing)

$$L_{k1} = z_{k1} b_{st.k} .$$

35) Axial dimension of inlet casings for "Zorya-Mashproekt" engines is

$$L_{inl} \approx (0.35...0.40)\, D_{out.1} .$$

36) Axial dimension of front casing

$$L_{fr} \approx (0.30...0.35)\, D_{out.1} .$$

37) Axial dimension of intersection between LPC and HPC

$$L_{int.k} \approx (0.30...0.35)\, D_{out.2.1} .$$

The calculation of the HPC is carried out according to the same formulas considering the comments made earlier.

The fulfillment of this part of an individual task is presented in Example 4.

Example 4

Determine the main dimensions of the GTU's compressor part for initial data and calculation results of Examples 2 and 3.

From the analysis of the basic gas turbine engine UGT 6000, we know that the GTU's compressor part consists of an inlet casing, front casing, low pressure compressor, intersection, high pressure compressor and rear casing.

The calculation of LPC, inlet casing, front casing and intersection is summarized in Table 7.

Table 7

Calculation of LPC, inlet casing, front casing and intersection

Parameter	Formula	Result
1. Air mass flow through LPC G_{k1}, kg/s	Table 5	23.8
2. Temperature at point 1 (at the LPC inlet) T_1, K	Table 5	288
3. Pressure at point 1 (at the LPC inlet) p_1, MPa	Table 5	0.100
4. Specific heat of air at the LPC inlet $c_{p\,a1}$, kJ/kg/K	$f(T_1)$	0.991
5. Gas constant of air R_{air}, kJ/kg/K	Equal to 0.287	0.287
6. Specific heat ratio for air at the LPC inlet k_{a1}	$\dfrac{c_{p\,a1}}{c_{p\,a1}-R_{air}}$	1.408
7. Circumferential velocity at outer diameter at the LPC inlet $u_{out.1}$, m/s	Accepted	270
8. Axial velocity at the LPC inlet c_{a1}, m/s	$(0.45...0.55)\,u_{out.1}$	135
9. Specific flow velocity at the LPC inlet $\lambda_{c_{a1}}$	$\dfrac{c_{a1}}{\sqrt{\dfrac{2k_{a1}}{k_{a1}+1}R_{air}T_1\cdot 10^3}}$	0.434
10. Gas-dynamic flow density function at the LPC inlet $q(\lambda_{c_{a1}})$	$\lambda_{c_{a1}}\left[\dfrac{k_{a1}+1}{2}\left(1-\dfrac{k_{a1}-1}{k_{a1}+1}\lambda_{c_{a1}}^2\right)\right]^{\frac{1}{k_{a1}-1}}$	0.632

Table 7

Parameter	Formula	Result
11. Required flow area at the LPC inlet F_1, m^2	$\dfrac{G_{k1}\sqrt{T_1}}{\sqrt{\left(\dfrac{2}{k_{a1}+1}\right)^{\frac{k_{a1}+1}{k_{a1}-1}} \cdot \dfrac{k_{a1}}{R_{air}\cdot 10^3}}\, p_1\, q(\lambda_{c_{a1}})\cdot 10^6}$	0.158
12. Diameter ratio for LPC $\bar{d}_1$	Accepted	0.50
13. Outer diameter at the LPC inlet $D_{out.1}$, m	$\sqrt{\dfrac{4F_1}{\pi\left(1-\bar{d}_1^2\right)}}$	0.518
14. LPC rotor speed n_{k1}, rpm	$\dfrac{60u_{out.1}}{\pi D_{out.1}}$	9 962
15. Inner diameter at the LPC inlet $D_{in.1}$, m	$D_{out.1}\bar{d}_1$	0.259
16. Length of the first stage blades l_1, m	$0.5\left(D_{out.1}-D_{in.1}\right)$	0.129
17. Medium diameter at the LPC inlet $D_{m.1}$, m	$0.5\left(D_{out.1}+D_{in.1}\right)$	0.388
18. Pressure at point 2.1 (at the LPC outlet) $p_{2.1}$, MPa	Table 5	0.467
19. Temperature at point 2.1 (at the LPC outlet) $T_{2.1}$, K	Table 5	469.7
20. Specific heat of air at the LPC outlet $c_{pa2.1}$, kJ/kg/K	$f(T_{2.1})$	1.033
21. Specific heat ratio for air at the LPC outlet $k_{a2.1}$	$\dfrac{c_{pa2.1}}{c_{pa2.1}-R_{air}}$	1.385
22. Axial velocity at the LPC outlet $c_{a2.1}$, m/s	$(0.90...0.95)c_{a1}$	121.5
23. Specific flow velocity at the LPC outlet $\lambda_{c_{a2.1}}$	$\dfrac{c_{a2.1}}{\sqrt{\dfrac{2k_{a2.1}}{k_{a2.1}+1}R_{air}T_{2.1}\cdot 10^3}}$	0.307

Table 7

Parameter	Formula	Result
24. Gas-dynamic flow density function at the LPC outlet $q(\lambda_{c_{a2.1}})$	$\lambda_{c_{a2.1}}\left[\frac{k_{a2.1}+1}{2}\left(1-\frac{k_{a2.1}-1}{k_{a2.1}+1}\lambda_{c_{a2.1}}^2\right)\right]^{\frac{1}{k_{a2.1}-1}}$	0.466
25. Required flow area at the LPC outlet $F_{2.1}$, m^2	$\frac{G_{k1}\sqrt{T_{2.1}}}{\sqrt{\left(\frac{2}{k_{a2.1}+1}\right)^{\frac{k_{a2.1}+1}{k_{a2.1}-1}}\cdot\frac{k_{a2.1}}{R_{air}\cdot 10^3}}\ p_{2.1}\ q(\lambda_{c_{a2.1}})\cdot 10^6}$	0.059
26. Compressor stages layout	Accepted $D_{out}=const$	
27. Outer diameter at the LPC outlet $D_{out.2.1}$, m	Equal to $D_{out.1}$	0.518
28. Inner diameter at the LPC outlet $D_{in.2.1}$, m	$\sqrt{D_{out.2.1}^2-\frac{4}{\pi}F_{2.1}}$	0.439
29. Length of the last stage blades $l_{2.1}$, m	$0.5(D_{out.2.1}-D_{in.2.1})$	0.039
30. Medium diameter at the LPC outlet $D_{m.2.1}$, m	$0.5(D_{out.2.1}+D_{in.2.1})$	0.479
31. LPC pressure ratio π_{k1}	Table 5	4.667
32. Pressure ratio of single LPC stage $\pi_{st.k}$	Accepted	1.25
33. Number of LPC stages (round to integer value) z_{k1}	$\frac{\lg\pi_{k1}}{\lg\pi_{st.k}}$	7 (6.90)
34. Medium LPC diameter D_{mk}, m	$0.5(D_{m1}+D_{m2.1})$	0.434
35. Medium length of LPC blades l_{mk}, m	$0.5(l_1+l_{2.1})$	0.084
36. Ratio λ_k	$\frac{D_{mk}}{l_{mk}}$	5.14
37. Coefficient $\overline{\delta}_k$	$f(\lambda_k)$ (see Table 6)	1.04

Table 7

Parameter	Formula	Result
38. Axial dimension of single LPC stage $b_{st.k}$, m	$\bar{\delta}_k l_{mk}$	0.088
39. Axial dimension of LPC main casing L_{k1} , m	$z_{k1} b_{st.k}$	0.614
40. Axial dimension of inlet casing L_{inl} , m	$(0.35...0.40) D_{out.1}$	0.207
41. Axial dimension of front casing L_{fr} , m	$(0.30...0.35) D_{out.1}$	0.181
42. Axial dimension of intersection between LPC and HPC $L_{int.k}$, m	$(0.30...0.35) D_{out.2.1}$	0.180

The calculation of high pressure compressor and rear casing is summarized in Table 8.

Table 8

Calculation of high pressure compressor and rear casing

Parameter	Formula	Result
1. Air mass flow through HPC G_{k2} , kg/s	Table 5	23.7
2. Temperature at point 1.2 (at the HPC inlet) $T_{1.2}$, K	Table 5	469.7
3. Pressure at point 1.2 (at the HPC inlet) $p_{1.2}$, MPa	Table 5	0.462
4. Specific heat of air at the HPC inlet $c_{pa1.2}$, kJ/kg/K	$f(T_{1.2})$	1.033
5. Gas constant of air R_{air} , kJ/kg/K	Equal to 0.287	0.287

Table 8

Parameter	Formula	Result
6. Specific heat ratio for air at the HPC inlet $k_{a1.2}$	$\dfrac{c_{pa1.2}}{c_{pa1.2} - R_{air}}$	1.385
7. Circumferential velocity at outer diameter at the HPC inlet $u_{out.1.2}$, m/s	Accepted	240
8. Axial velocity at the HPC inlet $c_{a1.2}$, m/s	$(0.45...0.55)\, u_{out.1.2}$	108
9. Specific flow velocity at the HPC inlet $\lambda_{c_{a1.2}}$	$\dfrac{c_{a1.2}}{\sqrt{\dfrac{2k_{a1.2}}{k_{a1.2}+1} R_{air}\, T_{1.2} \cdot 10^3}}$	0.273
10. Gas-dynamic flow density function at the HPC inlet $q(\lambda_{c_{a1.2}})$	$\lambda_{c_{a1.2}} \left[\dfrac{k_{a1.2}+1}{2}\left(1-\dfrac{k_{a1.2}-1}{k_{a1.2}+1}\lambda^2_{c_{a1.2}}\right)\right]^{\frac{1}{k_{a1.2}-1}}$	0.418
11. Required flow area at the HPC inlet $F_{1.2}$, m^2	$\dfrac{G_{k2}\sqrt{T_{1.2}}}{\sqrt{\left(\dfrac{2}{k_{a1.2}+1}\right)^{\frac{k_{a1.2}+1}{k_{a1.2}-1}} \cdot \dfrac{k_{a1.2}}{R_{air}\cdot 10^3}}\; p_{1.2}\, q(\lambda_{c_{a1.2}}) \cdot 10^6}$	0.066
12. Diameter ratio for HPC $\bar{d}_{1.2}$	Accepted	0.70
13. Outer diameter at the HPC inlet $D_{out.1.2}$, m	$\sqrt{\dfrac{4F_{1.2}}{\pi\left(1-\bar{d}^2_{1.2}\right)}}$	0.406
14. HPC rotor speed n_{k2}, rpm	$\dfrac{60\, u_{out.1.2}}{\pi D_{out.1.2}}$	11 290
15. Inner diameter at the HPC inlet $D_{in.1.2}$, m	$D_{out.1.2}\bar{d}_{1.2}$	0.284

Table 8

Parameter	Formula	Result
16. Length of the first stage blades $l_{1.2}$, m	$0.5\left(D_{out.1.2}-D_{in.1.2}\right)$	0.061
17. Medium diameter at the HPC inlet $D_{m.1.2}$, m	$0.5\left(D_{out.1.2}+D_{in.1.2}\right)$	0.345
18. Pressure at point 2 (at the HPC outlet) p_2, MPa	Table 5	2.079
19. Temperature at point 2 (at the HPC outlet) T_2, K	Table 5	737.6
20. Specific heat of air at the HPC outlet c_{pa2}, kJ/kg/K	$f(T_2)$	1.089
21. Specific heat ratio for air at the HPC outlet k_{a2}	$\dfrac{c_{pa2}}{c_{pa2}-R_{air}}$	1.358
22. Axial velocity at the HPC outlet c_{a2}, m/s	$(0.90...0.95)c_{a1.2}$	97
23. Specific flow velocity at the HPC outlet $\lambda_{c_{a2}}$	$\dfrac{c_{a2}}{\sqrt{\dfrac{2k_{a2}}{k_{a2}+1}R_{air}\,T_2\cdot 10^3}}$	0.197
24. Gas-dynamic flow density function at the HPC outlet $q(\lambda_{c_{a2}})$	$\lambda_{c_{a2}}\left[\dfrac{k_{a2}+1}{2}\left(1-\dfrac{k_{a2}-1}{k_{a2}+1}\lambda_{c_{a2}}^2\right)\right]^{\frac{1}{k_{a2}-1}}$	0.307
25. Required flow area at the HPC outlet F_2, m^2	$\dfrac{G_{k2}\sqrt{T_2}}{\sqrt{\left(\dfrac{2}{k_{a2}+1}\right)^{\frac{k_{a2}+1}{k_{a2}-1}}\cdot\dfrac{k_{a2}}{R_{air}\cdot 10^3}}\;p_2\,q(\lambda_{c_{a2}})\cdot 10^6}$	0.025
26. Compressor stages layout	Accepted $D_{in}=\text{const}$	
27. Inner diameter at the HPC outlet $D_{in.2}$, m	Equal to $D_{in.1.2}$	0.284

Table 8

Parameter	Formula	Result
28. Outer diameter at the HPC outlet $D_{out.2}$, m	$\sqrt{D_{in.2}^2 + \frac{4}{\pi} F_2}$	0.336
29. Length of the last stage blades l_2 , m	$0.5\left(D_{out.2} - D_{in.2}\right)$	0.026
30. Medium diameter at the HPC outlet $D_{m.2}$, m	$0.5\left(D_{out.2} + D_{in.2}\right)$	0.310
31. HPC pressure ratio π_{k2}	Table 5	4.583
32. Pressure ratio of single HPC stage $\pi_{st.k}$	Accepted	1.185
33. Number of HPC stages (round to integer value) z_{k2}	$\frac{\lg \pi_{k2}}{\lg \pi_{st.k}}$	9 (8.97)
34. Medium HPC diameter D_{mk} , m	$0.5(D_{m1.2} + D_{m2})$	0.328
35. Medium length of HPC blades l_{mk} , m	$0.5(l_{1.2} + l_2)$	0.043
36. Ratio λ_k	$\frac{D_{mk}}{l_{mk}}$	7.55
37. Coefficient $\bar{\delta}_k$	$f(\lambda_k)$ (see Table 6)	1.42
38. Axial dimension of single HPC stage $b_{st.k}$, m	$\bar{\delta}_k l_{mk}$	0.062
39. Axial dimension of HPC main casing L_{k2} , m	$z_{k2} b_{st.k}$	0.555
40. Axial dimension of rear casing L_r , m	$(0.35...0.40)\, D_{out.2}$	0.162

6. CALCULATION OF THE COMBUSTION CHAMBER DIMENSIONS

The function of GTU's combustion chamber is to heat the working fluid by burning hydrocarbon fuel.

In the combustion chambers of gas turbine units, the burning of fuel (liquid or gaseous) is carried out directly in the air flow coming from the compressor. The chemical energy of the fuel is converted into heat and transferred to the working fluid, which then moves to the turbine part of the GTU.

Depending on the type of fuel used, the application and structural scheme of the gas turbine unit, various types and layouts of combustion chambers are used.

By connecting with other parts of the gas turbine units, the combustion chambers are divided into external and built-in combustion chambers.

In turn, by design the built-in combustion chambers can be divided into:

– annular combustion chambers;
– multiple combustion chambers;
– tubo-annular (can-annular) combustion chambers;
– individual combustion chambers.

And according to the working fluid movement, the built-in combustion chambers can be straight-flow or reverse-flow ones.

Consider the calculation of the reverse-flow tubo-annular combustion chamber. The choice of that type of combustion chamber is explained by the fact that this type is dominant for "Zorya-Mashproekt" gas turbine units. Therefore, reverse-flow tubo-annular combustion chamber will be part of gas turbine units for most individual tasks.

Calculation of the combustion chamber is performed using statistics on existing designs of GTU's combustion chambers.

The scheme of reverse-flow combustion chamber is shown in Fig. 11.

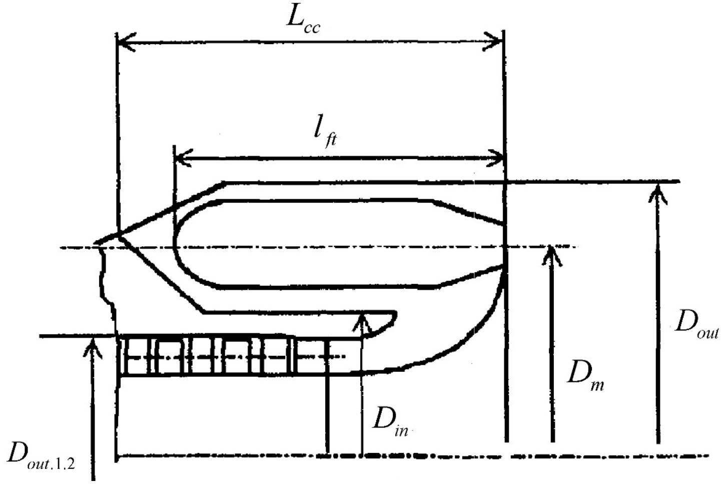

Fig. 11. The scheme of reverse-flow combustion chamber

A reverse-flow combustion chamber in a gas turbine unit with an axial-flow compressor is located above the outer casing of the high pressure compressor. After this compressor, air passes through its rear casing. Then it rotates at 180° and enters the annular space between the casings and flame tubes of the combustion chamber. Before entering the snout, there is another turn of the air flow by 180°.

As a result, we have a very strong vortex air flow in the annular space, which provides a uniform distribution of dilution air between the flame tubes. To ensure this, the medium diameter of the flame tubes installation is calculated as geometric.

The steps for calculating the reverse-flow tubo-annular combustion chamber dimensions are given below.

1) The following parameters were determined during thermal calculation of GTU at nominal operating mode:

– fuel consumption G_f;

– calorific value of fuel Q_f;
– stoichiometric coefficient L_0;
– combustion chamber efficiency η_{cc};
– pressure at point 2 (at the HPC outlet) p_2;
– air mass flow through combustion chamber G_{cc};
– temperature at point 2 (at the HPC outlet) T_2.

2) Heat load on the combustion chamber working volume for marine GTUs is equal to:

$$Q_v = 700\text{–}1000 \text{ kJ/m}^3\text{/hr/Pa}.$$

3) Combustion chamber working volume (internal volume of the flame tubes)

$$V_{ft} = \frac{G_f Q_f \eta_{cc}}{Q_v \, p_2 \cdot 10^6}.$$

4) Number of flame tubes n_{ft} is accepted with further clarification (in step 18). In a first approximation, it is taken equal to the number of flame tubes of the basic gas turbine unit.

5) Internal volume of the single flame tube

$$v_{ft} = \frac{V_{ft}}{n_{ft}}.$$

6) Coefficient of flame tube non-cylindricity is equal to:

$$a = 0.70\text{–}0.75.$$

7) Relative length of the flame tube

$$\bar{l} = 3\text{–}4.$$

8) Diameter of the flame tube

$$d_{ft} = \sqrt[3]{\frac{4 v_{ft}}{\pi \, \bar{l} \, a}}.$$

9) Flame tube length

$$l_{ft} = d_{ft}\,\bar{l}.$$

10) Circumferential clearance between flame tubes

$$\delta = (0.10...0.20)\,d_{ft}.$$

11) Medium diameter of the flame tubes installation

$$D_m = \frac{d_{ft} + \delta}{\sin\left(\dfrac{180^\circ}{n_{ft}}\right)}.$$

12) Primary combustion air excess coefficient is equal to:

$$\alpha_{pr} = 1.2\text{–}1.6.$$

13) Air mass flow through the snouts of all flame tubes

$$G_{pr} = \frac{\alpha_{pr}\, G_f\, L_0}{3600}.$$

14) Air mass flow through the gap between casing and flame tubes

$$G_{ch} = G_{cc} - G_{pr}.$$

15) Cross-section area of all flame tubes

$$F_{ft} = \frac{\pi}{4} d_{ft}^2\, n_{ft}.$$

16) Summary cross-section area of the combustion chamber

$$F_{cc} = (2.2...2.5)\, F_{ft}.$$

17) Diameter of the combustion chamber inner casing

$$D_{in} = D_m - \frac{F_{cc}}{\pi D_m}.$$

18) At this step, we must check the layout of the combustion chamber with the compressor part of GTU. Our combustion chamber is reverse-flow and is located above the high pressure compressor. Therefore, the diameter of the inner casing of the combustion chamber must be higher than the outer diameters of this compressor:

$$D_{in} > D_{out.1.2} \text{ and } D_{in} > D_{out.2}.$$

If this check is not performed, we need to re-accept the number of flame tubes in step 4. An increase in the number of flame tubes leads to an increase in the diameter of the combustion chamber inner casing.

When you checking the layout of the combustion chamber, it must be remembered that the equation $D_{in} >> D_{out.1.2}$ is also not desirable. The number of flame tubes should be chosen as the minimum for the correct combustion chamber layout.

19) Diameter of the combustion chamber outer casing

$$D_{out} = D_m + \frac{F_{cc}}{\pi D_m}.$$

20) Combustion chamber length

$$L_{cc} = (1{,}1...1{,}2)\, l_{ft}.$$

The fulfillment of this part of an individual task is presented in Example 5.

Example 5

Determine the main dimensions of the reverse-flow tubo-annular combustion chamber for initial data and calculation results of Examples 2–4.

We know that the basic gas turbine engine UGT 6000 has 10 flame tubes. But your initial data and designing results are not completely identical with the basic engine. For that reason we must perform several calculations of combustion chamber for different values of . After calculations, we will choose the number of flame tubes most optimal for layout of our gas turbine unit.

A series of combustion chamber calculations are summarized in Table 9.

Table 9

Calculation of the combustion chamber

Parameter	Formula	Result				
1. Fuel consumption G_f, kg/hr	Table 5	1750				
2. Calorific value of fuel Q_f, kJ/kg	Table 5	42915				
3. Stoichiometric coefficient L_0, kg/kg	Table 5	14.78				
4. Combustion chamber efficiency η_{cc}	Table 5	0.985				
5. Pressure at point 2 (at the HPC outlet) p_2, MPa	Table 5	2.079				
6. Air mass flow through combustion chamber G_{cc}, kg/s	Table 5	21.2				
7. Temperature at point 2 (at the HPC outlet) T_2, K	Table 5	737.6				
8. Heat load on the combustion chamber working volume Q_v, kJ/m^3/hr/Pa	Accepted	900				
9. Combustion chamber working volume V_{ft}, m^3	$\dfrac{G_f Q_f \eta_{cc}}{Q_v\, p_2 \cdot 10^6}$	0.0395				
10. Number of flame tubes n_{ft}	Accepted	**8**	**10**	**12**	**14**	**16**
11. Internal volume of the single flame tube v_{ft}, m^3	$\dfrac{V_{ft}}{n_{ft}}$	0.0049	0.0040	0.0033	0.0028	0.0025

Table 9

Parameter	Formula	Result				
12. Coefficient of flame tubes non-cylindricity a	Accepted	0.7				
13. Relative length of the flame tubes $\bar{l}$	Accepted	3.5				
14. Diameter of the flame tube d_{ft}, m	$\sqrt[3]{\frac{4 v_{ft}}{\pi \bar{l} a}}$	0.137	0.127	0.120	0.114	0.109
15. Flame tube length l_{ft}, m	$d_{ft} \bar{l}$	0.479	0.445	0.419	0.398	0.380
16. Circumferential clearance between flame tubes δ, m	$(0.10...0.20) d_{ft}$	0.021	0.019	0.018	0.017	0.016
17. Medium diameter of the flame tubes installation D_m, m	$\frac{d_{ft} + \delta}{\sin\left(\frac{180°}{n_{ft}}\right)}$	0.412	0.473	0.532	0.588	0.641
18. Primary combustion air excess coefficient α_{pr}	Accepted	1.5				
19. Air mass flow through the snouts of all flame tubes G_{pr}, kg/s	$\frac{\alpha_{pr} G_f L_0}{3600}$	10.8				
20. Air mass flow through the gap between casing and flame tubes G_{ch}, kg/s	$G_{cc} - G_{pr}$	10.4				
21. Cross-section area of all flame tubes F_{ft}, m^2	$\frac{\pi}{4} d_{ft}^2 n_{ft}$	0.1178	0.1269	0.1349	0.1420	0.1484
22. Summary cross-section area of the combustion chamber F_{cc}, m^2	$(2.2...2.5) F_{ft}$	0.2769	0.2982	0.3169	0.3336	0.3488
23. Diameter of the combustion chamber inner casing D_{in}, m	$D_m - \frac{F_{cc}}{\pi D_m}$	**0.198**	**0.273**	**0.342**	**0.407**	**0.468**

Table 9

Parameter	Formula	Result				
24. Diameter of the combustion chamber outer casing D_{out}, m	$D_m + \frac{F_{cc}}{\pi D_m}$	0.626	0.674	0.722	0.768	0.814
25. Combustion chamber length L_{cc}, m	$(1.1...1.2) l_{ft}$	0.551	0.512	0.482	0.457	0.438

From Table 9 we can see that an increase in the number of flame tubes n_{ft} (row 10) leads to an increase in the diameter of the combustion chamber inner casing D_{in} (row 23).

Choose the value of n_{ft} = 14. This value satisfies the equations:

$$D_{in} > D_{out.1.2} \ (0.407 > 0.406)$$

and

$$D_{in} > D_{out.1.2} \ (0.407 > 0.336).$$

The number of flame tubes was chosen as the minimum for the correct combustion chamber layout.

7. CALCULATION OF THE TURBINE DIMENSIONS

The function of the GTU's turbine part is to produce the mechanical energy of rotation.

The turbine is a heat engine in which the potential energy of the working fluid is converted into mechanical energy of the rotor. The potential energy is characterized by an enthalpy, which is functionally dependent on the pressure and temperature of the working fluid. The pressure of the working fluid in the gas turbine units increases in the compressor part. The temperature of the working fluid increases in the combustion chamber. And we know that the working fluid in GTU's turbine part is the product of hydrocarbon fuel's combustion.

In the majority of gas turbine units, axial-flow turbines are used; in low power GTUs, radial-inflow turbines are sometimes used.

Consider the determination of the main dimensions of the GTU's turbine part.

We accept that GTU's turbine part consists of three axial-flow turbines: high pressure turbine, low pressure turbine and power turbine. The dimensions of the turbines are calculated sequentially, starting with HPT. The scheme of axial-flow turbine is shown in Fig. 12.

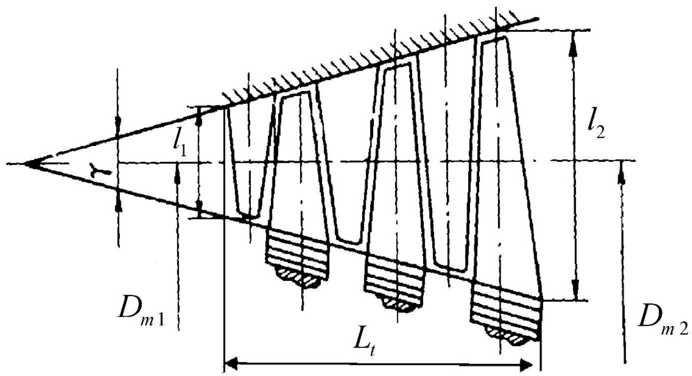

Fig. 12. The scheme of axial-flow turbine

The calculation of all turbines is carried out according to the same formulas considering the some features. The steps for calculating the dimensions of axial-flow gas turbine are given below.

1) The following parameters were determined during thermal calculation of GTU at nominal operating mode:

– gas mass flow through turbine G_t (accordingly, G_{t1}, G_{t2} and G_{t3});

– turbine inlet temperature T_1 (accordingly, T_3, $T_{3.2}$ and $T_{3.3}$);

– turbine inlet pressure p_1 (accordingly, p_3, $p_{3.2}$ and $p_{3.3}$);

– excess air coefficient in the combustion chamber α.

2) Specific heat of gas before turbine

$$c_{pg1} = f(T_1, \alpha).$$

3) Specific heat ratio for gas before turbine

$$k_{g1} = \frac{c_{pg1}}{c_{pg1} - R_{gas}},$$

where R_{gas} = 0.288–0.290 kJ/kg/K – gas constant of turbine gas.

4) Medium diameter at the turbine inlet D_{m1} is accepted with further clarification in step 26.

For HPT $D_{m.1}$ is approximately equal to medium diameter of the flame tubes installation.

For LPT $D_{m.1}$ is approximately equal to medium diameter before HPT.

For PT $D_{m.1}$ is some more than medium diameter of LPT.

5) Turbine rotor speed n_t is accepted based on the GTU's design features.

For HPT n_t is equal to the rotor speed of HPC.

For LPT n_t is equal to the rotor speed of LPC.

For PT n_t is equal to the rotor speed of load (consumer).

6) Circumferential velocity at the medium diameter of the turbine inlet

$$u_1 = \frac{\pi D_{m.1} n_t}{60}.$$

7) Axial velocity at the turbine inlet

$$c_{a1} = (0.30...0.40)\, u_1.$$

8) Specific flow velocity at the turbine inlet

$$\lambda_{c_{a1}} = \frac{c_{a1}}{\sqrt{\frac{2 k_{g1}}{k_{g1}+1} R_{gas} T_1 \cdot 10^3}}.$$

9) Gas-dynamic flow density function at the turbine inlet

$$q(\lambda_{c_{a1}}) = \lambda_{c_{a1}} \left[\frac{k_{g1}+1}{2} \left(1 - \frac{k_{g1}-1}{k_{g1}+1} \lambda_{c_{a1}}^2 \right) \right]^{\frac{1}{k_{g1}-1}}.$$

10) Required flow area at the turbine inlet

$$F_1 = \frac{G_t \sqrt{T_1}}{\sqrt{\left(\frac{2}{k_{g1}+1} \right)^{\frac{k_{g1}+1}{k_{g1}-1}} \cdot \frac{k_{g1}}{R_{gas} \cdot 10^3}}\; p_1 \, q(\lambda_{c_{a1}}) \cdot 10^6}.$$

11) Length of the first stage blades

$$l_1 = \frac{F_1}{\pi D_{m.1}}.$$

12) Outer diameter at the turbine inlet

$$D_{out.1} = D_{m.1} + l_1 .$$

13) Inner diameter at the turbine inlet

$$D_{in.1} = D_{m.1} - l_1 .$$

14) Number of turbine stages z_t is accepted according to the basic gas turbine engine.

For example, such engines of "Zorya-Mashproekt" as UGT 3000, UGT 6000, UGT 10000, UGT 15000 and UGT 25000 has single-stage HPT and single-stage LPT. If your basic gas turbine engine is one of them, then you must accept z_{t1} = 1 and z_{t2} = 1.

Depending on the application, the number of stages of a power turbine z_{t3} can be from 2 to 6.

15) The following parameters were determined during thermal calculation of GTU at nominal operating mode:

– turbine outlet pressure p_2 (accordingly, $p_{4.1}$, $p_{4.2}$ and p_4);

– turbine outlet temperature T_2 (accordingly, $T_{4.1}$, $T_{4.2}$ and T_4).

16) Specific heat of gas at the turbine outlet

$$c_{p\,g2} = f(T_2, \alpha).$$

17) Specific heat ratio for gas at the turbine outlet

$$k_{g2} = \frac{c_{p\,g2}}{c_{p\,g2} - R_{gas}}.$$

18) Axial velocity at the turbine outlet

$$c_{a2} = (1.00...1.05)c_{a1} .$$

19) Specific flow velocity at the turbine outlet

$$\lambda_{c_{a2}} = \frac{c_{a2}}{\sqrt{\frac{2\,k_{g2}}{k_{g2}+1} R_{gas}\, T_2 \cdot 10^3}}.$$

20) Gas-dynamic flow density function at the turbine outlet

$$q(\lambda_{c_{a2}}) = \lambda_{c_{a2}} \left[\frac{k_{g2}+1}{2} \left(1 - \frac{k_{g2}-1}{k_{g2}+1} \lambda_{c_{a2}}^2 \right) \right]^{\frac{1}{k_{g2}-1}}.$$

21) Required flow area at the turbine outlet

$$F_2 = \frac{G_t \sqrt{T_2}}{\sqrt{\left(\frac{2}{k_{g2}+1} \right)^{\frac{k_{g2}+1}{k_{g2}-1}} \frac{k_{g2}}{R_{gas} \cdot 10^3}} \; p_2 \, q(\lambda_{c_{a2}}) \cdot 10^6}.$$

22) Choosing the turbine stages layout. The most popular is the layout with a constant medium diameter $D_m = \text{const}$. For power turbine, the layout with a constant inner diameter or the layout with a constant outer diameter can be adopted also.

We accept the layout with a constant medium diameter for all three turbines. Then medium diameter after turbine:

$$D_{m.2} = D_{m.1}.$$

23) Length of the last stage blades

$$l_2 = \frac{F_2}{\pi D_{m.2}}.$$

24) Inner diameter at the turbine outlet

$$D_{in.2} = D_{m.2} - l_2.$$

25) Outer diameter at the turbine outlet

$$D_{out2} = D_{m.2} + l_2.$$

26) At this step we must check the equation

$$\frac{D_{m.2}}{l_2} > 3{,}5.$$

If this equation is not fulfilled, then the turbine has an unacceptably big length of the blades. Therefore, we must to return to step 4 and increase the medium diameter of the turbine D_{m1}. (An increase in the medium diameter D_{m1} improves the fulfillment of this equation.)

27) Medium diameter of turbine

$$D_{m.t} = 0.5(D_{m.1} + D_{m.2}).$$

28) Medium length of turbine blades

$$l_{m.t} = 0.5(l_1 + l_2).$$

29) Ratio

$$\lambda_t = \frac{D_{m.t}}{l_{m.t}}.$$

30) Coefficient $\overline{\delta}_t = f(\lambda_t)$ is determined using Table 10.

Table 10

Values of coefficient $\overline{\delta}_t$

λ_t	4	5	6	7	8	9	10	11	12	13	14	15	16	17	≥18
$\overline{\delta}_t$	0.62	0.77	0.91	1.05	1.17	1.27	1.37	1.46	1.53	1.60	1.66	1.71	1.74	1.79	1.82

31) Axial dimension of single turbine stage

$$b_{st.t} = \overline{\delta}_t l_{m.t}.$$

32) Axial dimension of turbine main casing

$$L_t = z_t b_{st.t}.$$

33) Axial dimension of intersection between LPT and PT for "Zorya-Mashproekt" engines is

$$L_{int.t} \approx (0.35...0.40)\, D_{out.2}^{LPT}.$$

34) Axial dimension of rear bearing casing for power turbine

$$L_{rb} \approx (0.30...0.35)\, D_{out.2}^{PT}.$$

The fulfillment of this part of an individual task is presented in Example 6.

Example 6

Determine the main dimensions of the GTU's turbine part for initial data and calculation results of Examples 2–5.

From the analysis of the basic gas turbine engine UGT 6000, we know that the GTU's turbine part consists of three axial-flow turbines: single-stage high pressure turbine (HPT), single-stage low pressure turbine (LPT) and three-stage power turbine (PT). There is no intersection between HPT and LPT.

Turbine rotor speed n_t for HPT is equal to the rotor speed of HPC (11290 rpm). For LPT n_t is equal to the rotor speed of LPC (9962 rpm). For PT n_t = 7300 rpm.

All three turbines has the layout with a constant medium diameter $D_m = \text{const}$. The medium diameter D_{m1} for HPT (0.585 *m*) is accepted some smaller than medium diameter of the flame tubes installation (0.588 *m*). HPT and LPT has the same medium diameter (0.585 *m*). The medium diameter for PT D_{m1} = 0.760 m is accepted with further clarification.

The axial dimension of intersection between LPT and PT $L_{int.t}$ and the axial dimension of rear bearing casing for PT L_{rb} are determined according to recommendations for "Zorya-Mashproekt" engines.

The results of the calculation are summarized in Table 11.

Table 11

Calculation of GTU's turbine part

Parameter	Formula	Result		
		HPT	LPT	PT
1. Gas mass flow through turbine G_t, kg/s	Table 5	21.7	23.1	24.2
2. Turbine inlet temperature T_1, K	Table 5	1500	1221	1049
3. Turbine inlet pressure p_1, MPa	Table 5	1.996	0.803	0.414
4. Excess air coefficient in the combustion chamber α	Table 5	2.953	2.953	2.953
5. Specific heat of gas at the turbine inlet c_{pg1}, kJ/kg/K	$f(T_1, \alpha)$	1.273	1.229	1.196
6. Gas constant of turbine gas R_{gas}, kJ/kg/K	Accepted	0.288	0.288	0.288
7. Specific heat ratio for gas at the turbine inlet k_{g1}	$\frac{c_{pg1}}{c_{pg1} - R_{gas}}$	1.292	1.306	1.317
8. Medium diameter at the turbine inlet D_{m1}, m	Accepted	0.585	0.585	0.760
9. Turbine rotor speed n_t, rpm	Accepted	1129 0	9962	7300
10. Circumferential velocity at medium diameter at the turbine inlet u_1, m/s	$\frac{\pi D_{m.1} n_t}{60}$	346	305	290
11. Axial velocity at the turbine inlet c_{a1}, m/s	$(0.30...0.40)\, u_1$	121	107	102
12. Specific flow velocity at the turbine inlet $\lambda_{c_{a1}}$	$\frac{c_{a1}}{\sqrt{\frac{2 k_{g1}}{k_{g1}+1} R_{gas} T_1 \cdot 10^3}}$	0.173	0.169	0.173

Table 11

Parameter	Formula	Result		
		HPT	LPT	PT
13. Gas-dynamic flow density function at the turbine inlet $q(\lambda_{c_{a1}})$	$\lambda_{c_{a1}}\left[\frac{k_{g1}+1}{2}\left(1-\frac{k_{g1}-1}{k_{g1}+1}\lambda_{c_{a1}}^2\right)\right]^{\frac{1}{k_{g1}-1}}$	0.273	0.266	0.272
14. Required flow area at the turbine inlet F_1, m^2	$\frac{G_t\sqrt{T_1}}{\sqrt{\left(\frac{2}{k_{g1}+1}\right)^{\frac{k_{g1}+1}{k_{g1}-1}}\cdot\frac{k_{g1}}{R_{gas}\cdot 10^3}}\,p_1\,q(\lambda_{c_{a1}})\cdot 10^6}$	0.0393	0.0959	0.1757
15. Length of the first stage blades l_1, m	$\frac{F_1}{\pi D_{m.1}}$	0.021	0.052	0.074
16. Outer diameter at the turbine inlet $D_{out.1}$, m	$D_{m.1}+l_1$	0.606	0.637	0.834
17. Inner diameter at the turbine inlet $D_{in.1}$, m	$D_{m.1}-l_1$	0.564	0.533	0.686
18. Number of turbine stages z_t	Accepted	1	1	3
19. Turbine outlet pressure p_2, MPa	Table 5	0.803	0.416	0.102
20. Turbine outlet temperature T_2, K	Table 5	1221	1049	771
21. Specific heat of gas at the turbine outlet c_{pg2}, kJ/kg/K	$f(T_2,\alpha)$	1.229	1.196	1.135
22. Specific heat ratio for gas at the turbine outlet k_{g2}	$\frac{c_{pg2}}{c_{pg2}-R_{gas}}$	1.306	1.317	1.340
23. Axial velocity at the turbine outlet c_{a2}, m/s	$(1.00...1.05)c_{a1}$	127	112	107
24. Specific flow velocity at the turbine outlet $\lambda_{c_{a2}}$	$\frac{c_{a2}}{\sqrt{\frac{2k_{g2}}{k_{g2}+1}R_{gas}T_2\cdot 10^3}}$	0.201	0.191	0.212

Table 11

Parameter	Formula	Result		
		HPT	LPT	PT
25. Gas-dynamic flow density function at the turbine outlet $q(\lambda_{c_{a2}})$	$\lambda_{c_{a2}}\left[\frac{k_{g2}+1}{2}\left(1-\frac{k_{g2}-1}{k_{g2}+1}\lambda_{c_{a2}}^{2}\right)\right]^{\frac{1}{k_{g2}-1}}$	0.315	0.299	0.329
26. Required flow area at the turbine outlet F_2, m^2	$\frac{G_t\sqrt{T_2}}{\sqrt{\left(\frac{2}{k_{g2}+1}\right)^{\frac{k_{g2}+1}{k_{g2}-1}}\cdot\frac{k_{g2}}{R_{gas}\cdot 10^3}}\,p_2\,q(\lambda_{c_{a2}})\cdot 10^6}$	0.0761	0.1521	0.5025
27. Turbine stages layout	Accepted	$D_m = \text{const}$		
28. Medium diameter at the turbine outlet $D_{m.2}$, m	Equal to $D_{m.1}$	0.585	0.585	0.760
29. Length of the last stage blades l_2, m	$\frac{F_2}{\pi D_{m.2}}$	0.041	0.083	0.211
30. Inner diameter at the turbine outlet $D_{in.2}$, m	$D_{m.2}-l_2$	0.544	0.502	0.549
31. Outer diameter at the turbine outlet $D_{out.2}$, m	$D_{m.2}+l_2$	0.626	0.668	0.971
32. Check of result	If $\frac{D_{m.2}}{l_2}\le 3.5$, then you must return to step 8	+	+	+
33. Medium diameter of turbine $D_{m.t}$, m	$0.5(D_{m.1}+D_{m.2})$	0.585	0.585	0.760
34. Medium length of turbine blades $l_{m.t}$, m	$0.5(l_1+l_2)$	0.031	0.068	0.142
35. Ratio λ_t	$\frac{D_{m.t}}{l_{m.t}}$	18.6	8.7	5.3

Table 11

Parameter	Formula	Result		
		HPT	LPT	PT
36. Coefficient $\bar{\delta}_t$	$f(\lambda_t)$	1.82	1.24	0.82
37. Axial dimension of single turbine stage $b_{st.t}$, m	$\bar{\delta}_t l_{m.t}$	0.057	0.084	0.117
38. Axial dimension of turbine main casing L_t, m	$z_t b_{st.t}$	0.057	0.084	0.350
39. Axial dimension of intersection between LPT and PT $L_{int.t}$, m	$(0.35...0.40)\, D_{out.2}^{LPT}$	0.234		
40. Axial dimension of rear bearing casing for PT L_{rb}, m	$(0.30...0.35)\, D_{out.2}^{PT}$	0.291		

8. SKETCHING OF GTU'S LAYOUT

At the first stages of designing a gas turbine unit, in order to determine its main sizes, the dimensional calculation and layout are carried out.

First of all, the radial and axial dimensions of compressors, combustion chamber and turbines are determined. During this calculation, the number of stages of compressors and turbines, the speed of their rotors, and the number of flame tubes in the combustion chamber are estimated. As a result, we obtain data for performing the layout of GTU and can draw a longitudinal section of its flowing path.

The radial and axial dimensions of the transitional parts of the GTU (the inlet casing, the front and rear compressor casings, the intersections between the compressors and turbines, etc.) required for the layout are usually determined based on the existing designs of gas turbine units.

When designing, it is necessary to ensure that the direction of the midline of the working fluid in the gas turbine unit is as smooth as possible. Sudden changes in the flow direction in the GTU due to the difference in the diameters of components lead to increased pressure losses and, consequently, to a deterioration in the efficiency of the gas turbine unit.

An example of layout of a gas turbine unit is shown in Fig. 13. All axial and radial dimensions necessary for sketching were determined in the previous parts of the work. Figure contains only the most important dimensions that was used.

The total length of the gas turbine unit is determined as the sum of the axial dimensions of its components:

$$L_{gt} = L_{inl} + L_{fr} + L_{k1} + L_{int.k} + L_{k2} +$$

$$+ L_r + L_{t1} + L_{t2} + L_{int.t} + L_{t3} + L_{rb}.$$

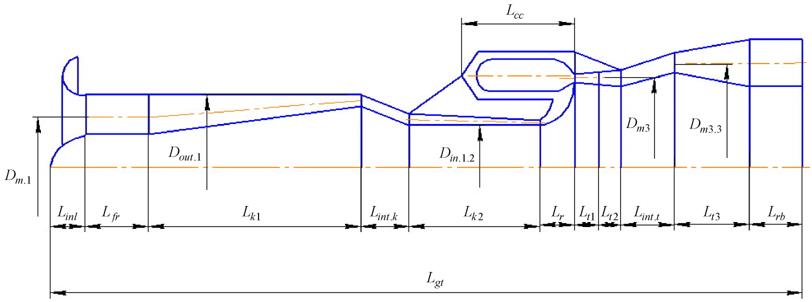

Fig. 13. The layout of gas turbine unit

Recall that the layout shown in Fig. 13 assumes the presence of a two-spool axial-flow compressor, a reverse-flow tubo-annular combustion chamber, and three axial-flow turbines in a gas turbine unit. Intersection between the 1st and 2nd turbines is absent. Such a layout is typical, for example, for most of the marine gas turbine units of "Zorya-Mashproekt", which are used, as a rule, as the basic gas turbine units for this individual task.

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ЗМІСТ

ДЛЯ НОТАТОК

Навчальне видання

ПАТЛАЙЧУК Володимир Миколайович
ВАЩИЛЕНКО Микола Віталійович

DESIGN OF GAS TURBINE UNITS

Recommendations for individual task

МЕТОДИЧНІ ВКАЗІВКИ
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просп. Героїв України, 9, м. Миколаїв, 54025
E-mail : publishing@nuos.edu.ua
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