

Gas Turbine Jet Engine

Lokotko, A. V.

Doctor of Technical Sciences

ABSTRACT

The concept of a gas turbine jet engine is proposed. The purpose of the development is to increase the thermal efficiency of the engine by increasing the temperature of the working fluid. The engine is a Segner wheel type device with a rotating combustion chamber and tangentially mounted nozzles. The torque is generated by the reaction force of the jets emanating from the nozzles. Full expansion of the working fluid takes place in a system of rotors installed coaxially with the combustion chamber and also equipped with jet nozzles. Cooling of the combustion chamber and chamber nozzles is carried out by a liquid metal coolant circulating due to centrifugal forces in combination with the thermosiphon effect. Calculated estimates show that at a working fluid temperature corresponding to the combustion temperature of a stoichiometric mixture of hydrocarbon fuel with air, the thermal efficiency in the design mode is 0.47, the specific fuel consumption is 0.25 kg / kWh, which is comparable with the corresponding indicators for diesel engines. The device received: a patent of the Russian Federation for an invention and a German patent for a utility model.

Key words: gas turbine jet engine (GTJE), nozzle, design mode, reaction force of outflow jets, centrifugal compressor, multistage expansion, liquid metal coolant.

Gas turbine engines (GTE) have a number of advantages over piston engines. They have a higher power density, a change in torque that is favorable for transport vehicles, i.e. better adaptability coefficient, full expansion of the working fluid, 2-3 times longer resource due to balance and minimization of rubbing surfaces, lower consumption of lubricating fluids, low requirements for fuel quality regardless of the octane number, shorter preparation time for start-up, especially at low temperatures. Meanwhile, GTJE are inferior to piston engines in terms of efficiency. This is determined by the insufficiently high thermal efficiency (efficiency) - the ratio of the useful work to the consumed heat - due to the limitation of the temperature at the turbine inlet due to insufficient heat resistance of the turbine blades. Lowering the temperature of gases to permissible limits in the known GTJE is achieved by supplying a large amount of air, significantly exceeding that required for fuel combustion at a stoichiometric ratio [1]. Additional power is expended on pumping excess air. An increase in the permissible operating temperature in certain cases is achieved by increasing the heat resistance of turbine blades, for example, the use of thermal barrier coatings based on cermets and (or) internal cooling of the blades. The best foreign gas turbine engines have a gas temperature at the turbine inlet of 1500C, with the prospect of increasing it to 1700C [2], but these values are significantly lower than the combustion temperature of stoichiometric mixtures of hydrocarbon fuels with air, equal to ~ 2100C [3]. That is, there are potentially opportunities to increase the temperature of the working fluid and, consequently, to increase the efficiency of the engine. It is known [1]

that an increase in the temperature of the working fluid by every 50 degrees increases the efficiency by 2%.

GTE installed on ground vehicles have relatively low power and, therefore, low air consumption. In this case, the diameter of the turbine impeller turns out to be small, and in the presence of rotor blades, the negative effect of the relative increase in the clearances between the rotor and stator increases, and the efficiency of the turbine decreases.

With the aim of increasing the efficiency, it seems promising to create a jet gas turbine engine with a rotating combustion chamber and the outflow of a working fluid from jet nozzles similar to the Segner wheel known from the physics course. In this case, the turbine blades are eliminated, which makes it possible to increase the temperature, and the gaps between the rotor and the stator of the turbine are eliminated.

In essence, a device with rotating rocket engines is implemented, the thermodynamic efficiency of which, as is known [4-6], is comparable to the efficiency of piston engines. The available rich experience in cooling rocket engine nozzles operating at very high temperatures allows us to hope for the possibility of implementing the combustion process of hydrocarbon fuels at a stoichiometric ratio.

Jet devices that create a torque on the shaft and a gas turbine engine with a rotating combustion chamber and jet nozzles are known [7 - 9].

The closest to the proposed device is the engine described in [9]. The power unit is made in the form of a jet turbine, on the outer surface of the rotating combustion chamber (CC) of which blades of a two-stage centrifugal compressor are installed, simultaneously playing the role of cooling elements of the combustion chamber. Heat recovery is carried out by heating the air entering the combustion chamber before the second stage of the compressor in a rotating recuperator heated by exhaust gases.

The disadvantages of this technical solution are: additional heat supply to the air in the process of increasing pressure, which reduces the compression ratio of the compressor and the efficiency of the power unit as a whole, the difficulty of providing sufficient heat removal from the combustion chamber due to the small coefficient of heat transfer to air. In addition, a single-stage turbine does not allow full expansion of the working fluid in the event of a further increase in the compressor compression ratio.

In the proposed jet GTJE [10], these disadvantages are eliminated. The engine has a rotating combustion chamber with gas outflow from tangentially located non-expanding nozzles. The outflow occurs at the speed of sound at a critical pressure drop. When the pressure in the jet and the surrounding space is equal (design mode), the wave pressure losses arising in the case of supersonic outflow are eliminated. However, the use of nozzles with sonic outflow does not allow the working fluid to be fully expanded in one stage; multi-stage expansion is required. Subsequent expansion of the working fluid on turbine stages with traditional airfoil blades would lead to a small degree of wheel partiality and large ventilation losses. Therefore, in the engine under consideration, the expansion of the working fluid occurs in several rotating chambers (rotors), sequentially covering the combustion chamber, the number of which

depends on the pressure generated by the compressor. The rotors are also equipped with tangentially located nozzles around the periphery, which create a reactive force when gas escapes from them and, accordingly, a torque. The total area of the flow area of the nozzles of each subsequent stage is selected in such a way as to ensure the design mode of the outflow from the nozzles of the previous stage.

The rotary torques of the rotors are summed up by means of a gearbox and transmitted to the power take-off shaft. The rotors rotate interdependently by means of specially selected gear ratios of the pairs of gears with the speeds obtained as a result of the gas-dynamic calculation of the engine flow path. In the rotating combustion chamber and in subsequent stages of expansion, the chemical energy of the fuel is converted into mechanical work, therefore, in accordance with the law of conservation of energy, the amount of work corresponds to a decrease in the enthalpy of the working fluid. The gas temperature at the outlet of the nozzles of each stage will gradually decrease and, given the possibility of using sufficiently heat-resistant materials, the stages following the combustion chamber will not require forced cooling.

Cooling of the combustion chamber and chamber nozzles is carried out by means of a liquid metal coolant (sodium + potassium alloy). In this case, the cooling of the CC body and the nozzles is achieved much easier than that of the turbine blades in the known devices. The transfer of heat from the heating medium to the incoming air is carried out after the last stage of the compressor, thereby ensuring heat recovery. This improves the efficiency of the engine. In fig. 1 shows the structural diagram of the engine. The GTJE contains a housing 1, a axial compressor 2, a device for feeding fuel 3 into the combustion chamber 4. A rotating combustion chamber 4 and coaxially with it covering its expansion stages of the working fluid 5, made in the form of hollow rotors, are equipped with tangentially installed jet nozzles 6. Combustion chamber connected to the rotor blade drum of the compressor 2 and rotates it. Fuel is supplied to the combustion chamber through injectors 7. Each rotor of the expansion stages of the working fluid is installed in bearings 8 with the possibility of rotation independent of the combustion chamber, and the direction of rotation of each subsequent rotor is opposite to the direction of rotation of the previous rotor, the rotation of the rotors is kinematically connected to each other by means of a gearbox 9, transmitting the total torque to the power take-off shaft 10.

The combustion chamber 4 has a cooling jacket containing "hot" 11 and "cold" 12 cavities filled with liquid metal. The cavities communicate with each other at a small radius of the cooling jacket and in the area of heat release. Heat transfer to the cooling air is carried out in sections 14 of the cooling jacket. The circulation of the liquid metal coolant occurs due to centrifugal forces in combination with the thermosiphon effect, which manifests itself due to the strong dependence of the density of the liquid metal on temperature.

The high pressure of the working fluid in the combustor and rotating rotors is maintained by means of the same type of labyrinth seals.

GAS-DYNAMIC CALCULATION OF THE FLOW PATH

The procedure for calculating the engine is given for the estimated net power $N \sim 100$ kW when using hydrocarbon fuel with a calorific value $H_u = 42700$ kJ / kg.

The power value is chosen as an example to concretize the calculated estimates. The total power of the turbine should be higher, taking into account the efficiency $\varphi = 1 - l_k / l_T$, where l_k , l_T , respectively, of the compressor and turbine operation. The φ value is calculated by the formula [1]:

$\kappa - 1$

$$\varphi = 1 - \tau \sigma_k \kappa \quad (1)$$

where: $\tau = T_h / T_1 = 288 / 2300 = 0.1252$ is the ratio of the temperature at the compressor inlet T_h to the temperature in front of the nozzle unit T_1 . Accepted value $T_t = 2300\text{K}$, corresponding to the combustion temperature T_{re} of a stoichiometric mixture of liquid hydrocarbon fuel with air [3]. Note that at a temperature of 2300K there is no noticeable dissociation of combustion products, and there is no need to take it into account [12].

σ_k is the compression ratio of the compressor.

A centrifugal or axial compressor can be used to compress air. The centrifugal compressor has a higher head. In the tank GTE mastered by industry, a two-stage centrifugal compressor provides a total compression ratio of $\sigma_c \sim 16$ [12]. Let us take the value $\sigma_k \approx 20$, which can be achieved in a multistage axial compressor.

$\kappa = 1.4$ is the adiabatic index for air.

The calculation gives $\varphi \approx 0.7$. Thus, 30% of the turbine power will be taken by the compressor, so the "gross" turbine power should be about $N = 130 \text{ kW}$.

Meanwhile, the optimal compression ratio for the specified value of τ , corresponding to the highest efficiency [1], determined by the formula: σ_{opt}

$$\sigma_{\text{opt}} = \frac{1}{\tau} \frac{\kappa}{2(\kappa - 1)} \quad -$$

is achieved at a compression ratio of $\sigma_{\text{opt}} \approx 38$ and, accordingly, there are prospects for boosting the engine in this parameter.

Let us determine the fuel consumption g to obtain a given power, taking into account the thermal efficiency typical for modern heat engines. η_t ($\eta_t \approx 40\%$):

$$g = N / \eta_t \text{Hu} = 0.0076 \text{ kg / sec.}$$

The mass air flow G required for combustion of this fuel flow is determined by the stoichiometric coefficient equal to $l_0 = 14.3$: $G = gl = 0.0076 \cdot 14.3 = 0.109 \text{ kg / sec.}$

Let's take the "calculated" air consumption $G = 0.11 \text{ kg / sec.}$

GTE belongs to the jet type engines. The specific (per 1 kg of mass consumption) total energy at the nozzle entrance i_0 is determined by the enthalpy i and the velocity

w^2

$$i_0 = i + \frac{w^2}{2} \quad -$$

The calculation of the motor steps is carried out in the following sequence. Determined (the digital index indicates the number of the stage): velocity of gas outflow from the nozzle

$$\text{(critical) [12]: } W_1 = \sqrt{\frac{2\kappa}{\kappa+1} RT_1} \text{ [m/sek]}$$

For combustion products: $\kappa = 1.33$; $R = 288.12 \text{ J / kg} \cdot \text{deg}$.

Kinetic energy of the jet:

$$E_1 =$$

$$\frac{G \cdot W^2}{2} \quad \text{---} \quad \frac{1}{2}$$

[W]

The total area of the nozzle throat [12]:

$$\bar{F} = \frac{G \sqrt{T_1}}{m \cdot p_{01}} \quad \text{---} \quad 1$$

$$\text{[m}^2\text{]}$$

where: $m = 0.0397 \text{ [kg} \cdot \text{deg / J]}^{0.5}$ for hydrocarbon fuel combustion products;

$P_{01} \text{ [N / m}^2\text{]}$ - total pressure in the pre-nozzle volume. The number of nozzles in the rotor is assumed to be four.

The kinetic energy of the outflowing jets E_1 must be converted into the energy of rotation of the combustion chamber (rotor) N . For the ideal case (without taking into account losses), this can be written as:

$$E_{*1} = P_1 \cdot W_1 \text{ [N} \cdot \text{m / sec]} = [W] \quad (2) \text{ Here: } P_1 = I_1 - p_2 \cdot F_1 \text{ is the thrust force of the nozzles;}$$

$W_1 \text{ [m / s]}$ - the peripheral speed of the nozzles;

$I_1 = p_{01} \cdot F_1 \cdot f(\lambda) \text{ [N]}$ is the impulse of the jet;

p_2 is the pressure in the cavity of the subsequent rotor, where the jets flow;

$$f(\lambda) = \frac{\lambda^{\kappa-1} (1 + \lambda^{\kappa+1})}{\lambda^{\kappa+1} + 1} \quad \text{---} \quad 1$$

$$\text{) } \lambda^{\kappa-1} = 1,2591 \text{ (for } \lambda = 1) \text{ is the impulse function [13];}$$

$$\lambda_- = W / a \text{ - speed coefficient;}$$

a - is the speed of sound corresponding to the temperature at the critical outflow; for the design flow regime: $p_2 = p_{01} \cdot \pi(\lambda_*)$

$$P \quad \kappa-1$$

$$\frac{\kappa}{2}$$

$$\pi(\lambda) =$$

$P_0 = (1 - \kappa + 1 \lambda^{\kappa-1})$ - a function that determines the ratio of the pressure in the

flow to the total pressure. $\pi(\lambda) = 0.5404$ for $\lambda^* = 1$ with $\kappa = 1.33$.

From the formula (1), according to the known power N_1 , the circumferential speed of the nozzles is determined: $W_1 = N_1 / P_1$ [m / s]

The speed of the nozzles W_1 and the diameter D_1 of the location of the nozzles determine the number of rotor revolutions: $n_1 = \lfloor 60W_1 / \pi D_1 \rfloor = \lfloor (60 \text{ m}) / (\text{sec} \cdot \text{m}) \rfloor = [\text{rpm}]$

According to the described algorithm, the calculation of the second and subsequent stages of the engine is performed, taking as the initial parameters obtained as a result of the calculation of the previous stage. The results of calculating the gas-dynamic parameters for four stages of the engine are shown in Table 1

Each pair of the previous and subsequent rotors rotates in opposite directions. The peripheral speeds of movement of the adjacent surfaces of the rotors relative to the nozzles W_1 are equal to the sum of the absolute values of the peripheral speeds. These values of the sums are very close to the velocities of gas outflow from the nozzles $W^* 1$, (see Table 1), i.e. the jet, as it were, "stands motionless" in the inter-rotor space. This causes low hydraulic losses when the jet flows onto the inner surface of the subsequent rotor with the return of the remaining impulse of the jet to the rotor.

The decrease in temperature during the outflow from the nozzles is estimated from the change in the enthalpy of the gas under the assumption of complete deceleration of the jets in the subsequent rotors [13]: $N = c_p (T_1 - T_2) G$

Here: $c_p = 1.055$ [kJ / kg · deg] - heat capacity of combustion products at constant pressure. From here the temperature in the next stage is determined:
 $T_2 = T_1 - N / (c_p G)$ [deg]

Residual parameters of the gas after the fourth stage: temperature $T_{out} = 1152$ K, pressure 1.02 ata, i.e. approx. 0.6 bar overpressure. It can be seen that in this case, four nozzle stages provide an almost complete expansion of the working fluid.

The total engine power is the sum of the power of all stages:
 $N_{\Sigma} = 41.6 + 35.6 + 30.0 + 25.2 = 132.4$ kW.

From this value, subtract the power required to drive the compressor. Compressor specific work [1]:

$$L_k = c_p T_h (\sigma^{(\kappa-1) / \kappa} - 1) = 408.03 \text{ [kJ / kg]}$$

Here: $\kappa = 1.4$ is the adiabatic index for cold air. Compressor drive power:

$$N_k = l_k \cdot G = 408.03 \cdot 0.11 = 44.88 \text{ kW.}$$

It can be seen (see Table 1) that the power required to drive the compressor is comparable to the power of the first nozzle stage - a rotating combustion chamber. This makes it possible to

use a "two-shaft" engine scheme with an independent compressor drive from a rotating combustion chamber. As you know, this is favorable from the point of view of improving the torque characteristics of a gas turbine engine [11]. In this case, the useful power will be equal to the sum of the capacities of 2 - 4 stages:

$$N_{\text{sum}} = 35.6 + 30.0 + 25.2 = 90.8 \text{ kW.}$$

To determine the specific fuel consumption, it is necessary to clarify the thermal efficiency η_t according to the calculation results. [1]:

$$\eta'_t = \frac{\Delta T_{\text{KC}} - \Delta T_{\text{ВЫХ}}}{\Delta T_{\text{KC}}}$$

Here: $\Delta T_{\text{cc}} = T_1 - T_c = 2300 - 677.8 = 1622.2$ - heating in the combustion chamber;

$T_1 = 2300$ - combustion temperature in the combustion chamber;

$$\kappa - 1$$

$T_{\kappa} = T_h(\sigma_{\kappa})^{\frac{\kappa - 1}{\kappa}} = 677.8\text{K}$ - air temperature after the compressor;

$\Delta T_{\text{out}} = T_{\text{out}} - T_h = 1152 - 288 = 864^\circ$ - temperature difference with exhaust gases;

$T_h = 288\text{K}$ - ambient temperature. The calculation gives: $\eta_t = 0.467$.

Specific fuel consumption per hour at the calculated value of η , referred to the net power:

$$g_s = (g \cdot \eta_t \cdot 3600) / \eta_t N = 0.0076 \cdot 0.4 \cdot 3600 / 0.467 \cdot 90.8 = 0.258 \text{ [kg / kWh]}.$$

The obtained value of the specific fuel consumption is very close to this indicator for piston engines.

Meanwhile, in [1], an expression is given for the thermal efficiency η_t of the engine, taking into account the typical efficiency of the turbine $\eta_T = 0.9$ and the compressor $\eta_k = 0.86$:

$$\eta_T (1 - \frac{1}{\sigma}) - \frac{r}{\sigma} (\sigma^m - 1)$$

$$\eta =$$

$$\frac{\sigma^m}{\sigma^m}$$

$$\eta_K$$

$$\frac{1 - r - \frac{r}{\sigma} (\sigma^m - 1)}{\sigma^m}$$

$$\eta_K$$

$$\eta$$

Here: ; $\sigma = 20$ - compressor compression ratio;

$$\tau = T_h / T_{cs} = 288 / 2300 = 0.1252 \text{ - temperature coefficient. } m = (k - 1) / k$$

A calculation taking into account these parameters gives $\eta_t = 0.473$. It is seen that the values of η_t calculated in two ways are close.

Note (Table 1) that the ratio of the thrust of the stage nozzles to the momentum of the jets at the parameters corresponding to the critical outflow for all stages fluctuates around the value of ~ 0.58 . To increase the thrust, it is possible to reduce the pressure in the cavities of the jet flow, coupled with the use of supersonic (expanding) nozzles. However, this will lead to an increase in pressure losses in the shock waves during the outflow of nonisobaric jets.

Apparently, in the future, it is advisable to solve the problem of finding the optimum power depending on the degree of expansion of the nozzles.

Meanwhile, additional factors should be taken into account that affect the accuracy of the calculation both in the negative and in the positive direction: gas friction losses in the inter-disk spaces and the active effect of oncoming jets on subsequent nozzle stages (rotors), heat recovery from the cooling system. It is also possible that the Coanda effect (adhesion of jets to cylindrical surfaces) will affect the flow structure in the jet flow cavities.

ACCOUNTING FOR FRICTION LOSSES, INTERACTION OF JETS WITH THE SURFACE

Let us estimate the gas friction losses in the intervals between the rotating rotors. It is known that the moment of resistance is different for a freely rotating disc and a disc in a casing, and in the latter case, the resistance is less [15]. Losses arise in the slotted gaps on the end parts of the rotors and on the outer surfaces of the rotors, in the places where the nozzles are installed.

When the width of the end gaps is small, the thickness of the boundary layer is smaller, the peripheral velocity in the gap between the rotors is distributed linearly (corresponds to the Couette flow [15]). The rotors are counter-rotating, so the angular velocity ϖ is equal to the sum of the angular velocities of adjacent rotors. The effective moment of resistance is calculated by the formula for a two-sided flow around a disk [15]:

$$M = \pi \varpi R^4 \mu$$

s

(3)

The gap is taken equal $t_{0s} = 3$ mm, the radius R is equal to the radius of the installation of the nozzles, the coefficient of dynamic viscosity for the combustion products (CO_2) at temperatures in the rotor cavities is taken from [16] and is given in Table 2. There are also shown the moments calculated by formula (2) resistance and power loss to overcome the resistance in the end gaps, defined as $N = M^* \varpi$. The calculated values of these parameters are shown in Table 2. The interaction of the outflowing jets with the surface of the subsequent rotor could be a positive factor in allowing the additional power to be realized. It is known that when a jet collides at an angle with a flat screen, the kinetic energy of the flow is conserved [17]. The force of interaction of the jets with the surface F_{int} is determined from the momentum equation: $F_{int} = G (v_{ci} - \Delta w)$

where: G - gas consumption through the engine;

v_{ci} is the speed of the outflow of jets from the nozzles of the i -th rotor;

$\Delta w = |v_i| + |v_{(i+1)}|$ - the sum of the absolute values of the peripheral speeds of adjacent rotors. However, the estimates performed show that the difference between the jet outflow velocities and the values of Δw (the expression in parentheses in the above formula) is very small; therefore, the effect of interaction between the jets and the surface is not considered.

THERMAL CALCULATION OF THE COMBUSTION CHAMBER

The thermal calculation of the combustion chamber is carried out in order to clarify the possibility of creating a structure that is heat-resistant for a long time. The problem of stationary thermal conductivity at a known temperature of the medium washing the wall of the combustor is considered. As a result of the calculation, it is necessary to determine the

magnitude of the heat flux through the wall so that the wall temperature does not exceed the permissible value. The combustion chamber is cooled using liquid metal coolants, for example, an alloy of sodium and potassium [1, 18, 19].

It is also necessary to find out the possibility of dumping heat to the air after the fourth stage of the compressor in order to implement a cycle with heat recovery in conditions of significant air heating during compression in the compressor. This is known to be beneficial in terms of increasing the thermal efficiency of the engine.

Calculation of the heat transfer coefficient from the flow of hot gases to the wall. is carried out in criterion form according to the formula, which in most sources [16, 18] has the form

$$Nu = 0,023Re^{0,8}Pr^{0,4}\epsilon_l.$$

Here: $Nu = \alpha L / \lambda$ is the Nusselt number;

α is the heat transfer coefficient;

L - linear dimension;

λ is the coefficient of thermal conductivity of the medium washing the wall.

$Re = \rho WD / \mu$ is the Reynolds number. $Pr = (\mu c_p) / \lambda$ is the Prandtl number.

μ - is the coefficient of dynamic viscosity:

$\epsilon_l \approx 1.55$ is a coefficient that takes into account the relative length of the combustion chamber depending on the number Re [16].

To determine the Re number, it is necessary to find the characteristic velocity in the combustion chamber. An annular cylindrical section with a diameter of 0.2 m with a ring width of $l = 0.03\text{ m}$ and an area of $F_x = 0.0188\text{ m}^2$ was taken as "characteristic".

Density ρ of combustion products (CO_2) at pressure $p = 20\text{ atm}$ and temperature

$$p = 20 \cdot 10^4 \text{ кг}$$

$$T = 2300\text{K} \quad \rho = \frac{p}{RT} = \frac{19,27 \cdot 10^4}{19,27 \cdot 2300} = 4,51 \text{ кг}$$

constant for CO_2 [13]. Here $R = 19.27\text{ kgm} / \text{kg} \cdot \text{deg}$ is the gas

Velocity in the annular section: $w_x = G / (F_x \rho) = (0.11) / (0.0188 \cdot 4.15) \approx 1.4\text{ m} / \text{s}$

$$\rho w D = 4,51 \cdot 1,4 \cdot 0,2$$

Reynolds number:

$$\approx 1,8 \cdot 10^4$$

$$\mu = 0,7 \cdot 10^{-4}$$

$$Re = \frac{\rho w D}{\mu} = \frac{4,51 \cdot 1,4 \cdot 0,2}{0,7 \cdot 10^{-4}} = 1,8 \cdot 10^4$$

$\mu = 0.7 \cdot 10^{-4}\text{ Pa} \cdot \text{s}$ is the coefficient of dynamic viscosity for CO_2 at 2300C [19].

Prandtl number: $Pr = \frac{\mu c_p}{\lambda} = \frac{0,7 \cdot 10^{-4} \cdot 0,317 \cdot 3600}{0,0924} = 0,86.$

$$\lambda = 0,0924$$

$$0,0924$$

$c_p = 0.317 \text{ kcal / kg} \cdot \text{deg}$ is the heat capacity of CO_2 at 1500K, pressure of 30 atm [19].

$\lambda_1 = 0.0924 \text{ kcal / m} \cdot \text{hour} \cdot \text{deg}$ is the coefficient of thermal conductivity of CO_2 at a temperature of 1200C, a pressure of 30 atm [20].

Nusselt number: $\text{Nu} = 0,023 \cdot (1,7 \cdot 10^4)^{0,8} (0,86)^{0,4} \cdot 1,55 = 74,7$.

Heat transfer coefficient from hot gases to the wall of the combustion chamber:

$$\alpha_1 = \frac{\text{Nu} \lambda_1}{D} = 74,7 \cdot 0,0924 = 0,00958 \text{ kcal/m}^2 \cdot \text{sec} \cdot \text{deg} \text{ or, in the SI system, multiplying}$$

$$D \quad 0,2 \cdot 3600$$

by 4.19 kJ / kcal , we get $\alpha_1 = 0.0401 \text{ kW / m}^2 \text{ deg}$, or $\alpha_1 = 40.1 \text{ W / m}^2 \text{ deg}$ determined by the heat transfer coefficient α_1 from the side of the combustion products, must be removed by the liquid metal coolant. Heat transfer during the flow of molten metals in pipes, determined by the Nusselt number, depends on the Peclet number (Pe) in accordance with the formula [19]: $\text{Nu} = a + b\text{Pe}$.

where: a and b are coefficients depending on the type of heat exchanger. In this case, for an annular heat exchanger with heating of the inner wall: $a \approx 4.82$, $b \approx 0.0222$ [19]. The Peclet number is a combination of velocity u [m/s], linear dimension L [m] and thermal diffusivity a [m^2 / s]: $\text{Pe} = uL / a$.

In the device under consideration, with the assumed organization of the flow of liquid metal due to centrifugal forces in combination with the thermosiphon effect, the flow rate is not known in advance. Let us determine the order of the heat transfer coefficient for a certain average value of the Pe number, for which the Nu numbers were calculated in [19], in particular, for $\text{Pe} \approx 10^3$ $\text{Nu} = 14$.

An alloy of alkali metals 25% Na + 75% K at a temperature of 1000K has the following thermophysical characteristics:

thermal conductivity coefficient $\lambda = 27 \text{ kcal / m hour deg}$; thermal diffusivity $a = 0.18 \text{ m}^2 / \text{hour}$. Let us determine the speed for the selected number Pe and size $l = 0.2 \text{ m}$:

$$u =$$

$$\text{Pe} \cdot a =$$

$$L$$

$$10^3 \cdot 0,18$$

$$0,2 \cdot 3600 = 0,25 \text{ m / s}$$

$$\text{Nu} \cdot \lambda \quad 14 \cdot 27$$

$$\alpha =$$

Heat transfer coefficient to metal: α_2

$$= 0,525 \text{ kcal} \cdot \text{sec} \cdot \text{deg / m}^2,$$

$$L \quad 0,2 \cdot 3600$$

or in the SI system: $\alpha_2 = 2.2 \text{ kW / m}^2 \text{ deg}$, $= 2200 \text{ W / m}^2 \text{ deg}$.

It is seen that the value of the coefficient α_2 at the metal flow rate

0.25 m / s in 55 times the value of α_1 . Consequently, the decisive factor in the heat transfer process is the coefficient of heat transfer from the gaseous medium α_1 .

The combustion chamber body can be made of a heat-conducting bronze alloy, and the rotor bodies can be made of heat-resistant alloys either based on niobium BN2A, or molybdenum CM3, or tungsten BB2, which provide sufficient strength at temperatures up to 1200C [21]. Consider the effect of the combustion chamber wall on the heat transfer process (for example, steel). The average coefficient of thermal conductivity for heat-resistant steels at a temperature of 800C $\lambda = 30 \text{ W / m} \cdot \text{deg}$ [21]. The heat transfer process is described by formula (3), where t_1 is the wall temperature [22]:

$$\frac{t_{cp1} - t_1}{t_{cp1} - t_{cp2}} = \frac{\alpha_1}{\frac{1}{\delta} + \frac{\alpha_1}{\lambda} + \alpha_2} \quad (3)$$

The thermal resistance of a steel wall with a thickness of $\delta = 5 \text{ mm}$ has the value

$$\delta / \lambda = (0.005 \text{ m}) / (30 \text{ W / m} \cdot \text{deg}) = 1.66 \cdot 10^{-4} \text{ m}^2 \cdot \text{deg / W}$$

Substitute the obtained numerical values of the coefficients into formula (3).

$$\frac{1}{\frac{t_{cp1} - t_1}{t_{cp1} - t_{cp2}}} = \frac{40,1}{\frac{1}{40,1} + 1,66 \cdot 10^{-4} + \frac{1}{2200}} = 0,98 \approx 1$$

Hence it follows: $t_{av1} - t_1 \approx t_{av1} - t_{av2}$, i.e. $t_1 \approx t_{av2}$

This means that at high values of the heat transfer coefficient α_2 , characteristic of liquid metal coolants, the coolant acquires the wall temperature. The same conclusion is contained in [19, 22]. Thus, the thermal resistance of a steel wall (especially a bronze one) can be excluded from consideration of the problem of heat transfer.

Let us determine the heat transfer coefficient on the heat-receiving side, after the last stage of the axial compressor. Let us take the characteristic area of the flow area of the air flow F_k the same as the characteristic area of the combustion chamber, i.e. $F_{cc} = 0.0188 \text{ m}^2$. Other parameters necessary for the calculation: air flow $G = 0.11 \text{ kg / sec}$, pressure $P = 20 \text{ kg / cm}^2$, temperature $T_{comp} = T_{av3} = 734 \text{ K}$.

Air density: $\rho = P / RT = 20 \cdot 10^4 / 29.27 \cdot 734 = 9.31 \text{ kg / m}^3$.

Air speed: $w = G / F_{comp} \cdot \rho = 0.11 / 0.0188 \cdot 9.31 = 0.628 \text{ m / s}$. Dynamic viscosity of air at 734K: $\mu = 41.65 \cdot 10^{-6} \text{ kg / m} \cdot \text{s}$ [20].

The characteristic linear dimension required for calculating the Reynolds number is the same as when calculating the Re number in the combustion chamber: $D = 0.2$ m.

Reynolds number: $Re = wD / \mu = 9.31 \cdot 0.2 / 41.65 \cdot 10^{-6} = 28075$. Prandtl number for air: $Pr = 0.71$.

Nusselt number: $Nu_3 = 0.023 \cdot Re^{0.8} \cdot Pr^{0.4} \cdot \varepsilon_1 = 0.023 \cdot (28075)^{0.8} \cdot 0.71^{0.4} \cdot 1.55 = 112.61$. Heat transfer coefficient from the wall of the compressor casing to air:

$$\alpha = \frac{Nu_3 \lambda_B}{D} = \frac{112.61 \cdot 5.88 \cdot 10^{-2}}{0.2} = 0.0092 \text{ kcal} / \text{m}^2 \cdot \text{sek} \cdot \text{deg}$$

Here $\lambda_w = 5.88 \cdot 10^{-2} \text{ kcal} / \text{m} \cdot \text{hour} \cdot \text{deg}$ is the coefficient of thermal conductivity of air at temperature $T_{av3} = 734\text{K}$ [20].

It is required to determine the temperature of the liquid metal coolant after cooling T_{av2} , depending on the parameter - the area of heat release F_3 . Let us write the heat balance equation from the condition that the heat flux received by the walls of the combustion chamber should be transferred by the liquid metal coolant through the walls after the last stage of the compressor, thereby carrying out the process of transferring heat to the incoming air.

$$F_{cc} \alpha_1 (T_{av1} - T_{av2}) = F_3 \alpha_3 (T_{cr2} - T_{cr3})$$

Here F_{cc} , α_1 , α_3 , T_{av3} are known values calculated above.

Making simple transformations, we get the formula:

$$F_{cc} \alpha_1 T_{cp1} + \alpha_3 T_{cp3}$$

$$T_{cp3} =$$

$$\frac{F_3}{\alpha_3 + F_{cc} \alpha_1}$$

$$F$$

The temperature T_{av2} , depending on the relative area F_{cc} / F_3 , is given in Table 3. It can be seen that when the value $F_x / F_3 = 0.1$, the temperature $T_{av2} = 881\text{K}$, i.e. 608C . This value of the coolant temperature is widely used in the circuits of nuclear power plants [19] and, apparently, can be acceptable for the considered design. To ensure this, the surface area of the heat release F_3 must be 10 times the surface area of the combustion chamber F_{cc} . Taking into account the compactness of the compressor station, as well as the possibility of intensifying the heat exchange in the compressor (for example, by installing the finning), such a configuration of the heat exchange surface downstream of the last compressor stage can be realized.

Obviously, taking into account the above conclusion, the temperature of the CC wall will be equal to the temperature of the liquid metal coolant, i.e. 608C , which is permissible for copper or heat-resistant steel alloys.

Let us consider how to ensure the conditions for the flow of liquid metal. Figure 1 shows that the combustion chamber has a jacket covering it, divided into cavities that allow metal to circulate through them. The cavity directly in contact with the camera will heat up more. The

pressure in the fluid p , which arises during the rotation of the combustor due to the centrifugal force, is determined by the formula [22]: $p = \rho \cdot \omega^2 \cdot r^2$.

where: ρ is the density of the liquid metal, depending on temperature;

$\omega = 3462.3 \text{ 1 / sec}$ - angular velocity of rotation of the CC;

$r = 0.125 \text{ m}$ is the maximum radius of the cooling jacket that encloses the combustion chamber.

The density of the metal alloy in question is temperature dependent. So, for temperatures of 800, 900, 1000K, the density of the alloy 25% Na + 75% K, respectively, will be: $\rho = 747; 722; 698 \text{ kg / m}^3$ [19]. The calculation shows that when the temperature difference in the "hot" and "cold" cavities of the cooling jacket is 100° (which is quite realistic), i.e. at $\Delta\rho = 25 \text{ kg / m}^3$, the pressure difference in the cavities, at the radius of the hole, will be $\Delta p \approx 4.7 \text{ MPa}$.

This pressure drop should ensure the circulation of the liquid metal. The flow rate can be determined after additional studies of the coefficient of hydraulic resistance in a channel with a complex spatial configuration, which is represented by the cooling jacket of the combustion chamber. There are no reference materials for calculating such channels.

The pressure of the liquid metal in the cooling cavity under the conditions under consideration:
 $p = 747 \cdot (3460)^2 \cdot (0.125)^2 \approx 140 \text{ MPa}$

The body of the compressor station must be designed for such a pressure of liquid metal.

EFFECT OF HEAT RECOVERY

Usually, the increase in the efficiency of the GTE, depending on the degree of heat recovery, is associated with the use of heat from the gases leaving the turbine [1]. In this case, heat from the cooling system is used, and estimates can be performed as follows.

The amount of heat transferred from the liquid metal to the air after the last compressor stage:

$$Q_3 = F_3 \alpha_3 (T_{av2} - T_{av3})$$

Here are the known values calculated above:

F_3 - area of heat release $F_3 = 10 F_{cc}$; $F_3 = 0.982 \text{ m}^2$;

$\alpha_3 = 0.0092 \text{ kcal / m}^2 \cdot \text{sec} \cdot \text{deg}$ - coefficient of heat transfer to air;

$T_{av2} = 881 \text{ K}$ is the temperature of the liquid metal;

$T_{cp3} = 734 \text{ K}$ - air temperature after the last compressor stage.

The calculation gives: $Q_3 = 0.982 \cdot 0.0092 \cdot (881 - 734) \approx 1.33 \text{ kcal / s}$

Specific fuel consumption was determined above: $g_f = 258 \text{ g / kW} \cdot \text{h}$ For a full power of 132.4 kW , the fuel consumption per second will be: $g = 258 \cdot 132.4 / 3600 = 9.5 \text{ g / s}$

Assuming the calorific value of diesel fuel $Hu \approx 10,000 \text{ kcal / kg}$, the second heat consumption from fuel combustion is 95 kcal / sec .

Relative decrease in the second fuel (heat) consumption due to regeneration:

$$\eta_f = (95 - 1.33) / 95 = 0.986,$$

that is, the transfer of heat from the cooling system to the intake air increases the engine's economy by approximately 1.4%.

CONCLUSION

The calculated estimates show that the gas turbine jet engine ensures the achievement of the set goal - increasing the efficiency of the gas turbine engine. Thermal efficiency in the design mode is 0.47, specific fuel consumption 0.26 kg / kWh, which is comparable to the corresponding figures for diesel engines.

The engine is "sharply tuned" to the optimal operating mode, so its use is apparently most expedient under constant load conditions, for example, to drive generators in distributed power systems, on hybrid cars - for charging batteries, or on small boats in as a marine engine. The engine, apparently, should have improved environmental properties in the sense of reducing noise during operation due to the complete expansion of the working fluid at the exhaust (as opposed to piston internal combustion engines)

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Table 1

Parameter	Stage 1 (combustion chamber)	Stage 2	Stage 3	Stage 4
Temperature K	2300	1935	1628	1370
Pressure, ata	20	10,8	5,7	3,0
Outflow velocity, m / s	870	805,0	738,4	677,4
Stage power, kW	41,6	35,6	30,0	25,2
Crete area sec. sum, cm ²	0,6781	1,1625	2,00	3,432
Diameter of one nozzle, mm	4,64	6,1	8,0	10,5
Impulse of jets, N	167,3	153,2	140,8	129,2
Draft of nozzles, N	95,5	88,2	82,0	74,7
Ratio: thrust / momentum	0,575	0,576	0,582	0,578
Diameter of the nozzle circle, D [m]	0,25	0,30	0,35	0,40
Peripheral speed m / s	435,7	336,6	261,4	211,3
Speed n [1 / min]	33063	25717	19969	16142
Angular velocity $\omega = 2\pi n / 60$ [1 / sec]	3462,3	2693,1	2091,1	1690,4

Table 2

Parameter	1 ст. (к.с.)	2 ст.	3 ст.	4 ст.
Temperature T [K]	2300	1935	1628	1370
Dynamic viscosity of CO ₂ $\mu \cdot 10^7$ [кг/м с]	6,47	6,0	5,43	5,15
Angular velocity $\varpi = 2\pi n/60$ [1/сек]	3462,3	2693,1	2091,1	1690,4
Frictional moment in end gaps [N · m]	—	$4,72 \cdot 10^{-4}$	$3,83 \cdot 10^{-3}$	$4,89 \cdot 10^{-3}$
Friction power in end gaps [Н·м/сек] = Вт	—	2,9	18,3	18,5
Sum of absolute values of peripheral speeds Δw [м/с]	—	772,3	598,0	427,7

Table 3

F_{kc}/F_3	T_{cp2K}
1	1533
0,5	1270
0,1	881

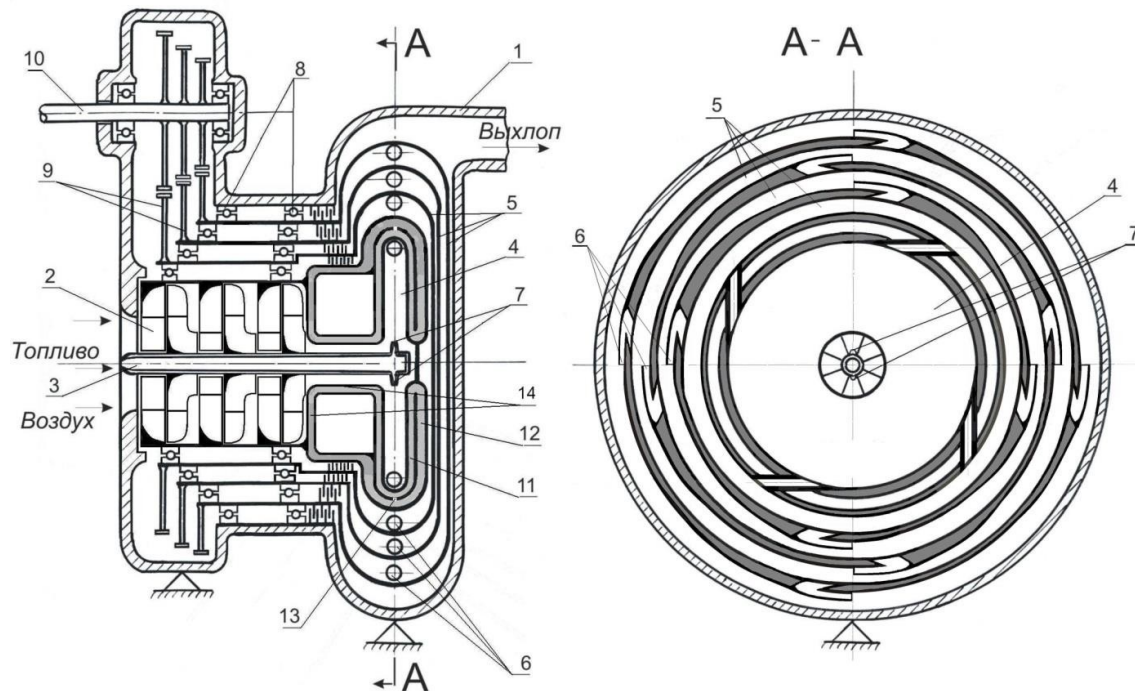


Fig. 1 Diagram of a gas turbine jet engine

1 - case; 2 -axial compressor; 3 -fuel supply pipeline; 4 - rotating combustion chamber; 5 - rotors; 6 - nozzles; 7 - nozzles; 8 - bearings; 9 - reducer; 10 - power take-off shaft; 11 - "hot" cavity of the cooling jacket; 12 - "cold" cavity of the cooling jacket; 13 - hole; 14 - section of heat dumping from the coolant to the incoming air. fuel air exhaust.

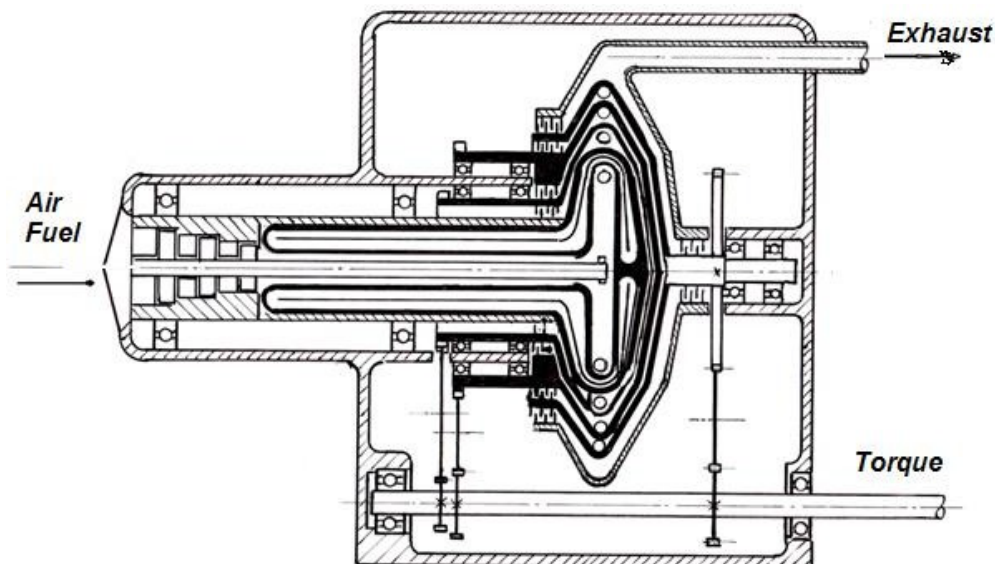


FIG. 2 Engine diagram close to real design