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THE WAYS OF IMPROVING THE EFFICIENCY OF GAS TURBINE PLANTS BASED ON AIRCRAFT GAS TURBINE ENGINES

Introduction

In the world practice, gas turbine plants (GTP) based on aviation gas turbine engines (GTE) have become quite widespread. These can be as newly designed and manufactured GTP, which use design solutions or even individual units of aircraft engines, and GTP, obtained by converting aircraft engines that have exhausted flight service life time on aircraft and to be disposed of. Since convertible GTE, designed in previous decades, tend to have relatively low operating cycle parameters (the pressure ratio in the compressor is not more than 10…12, and the gas temperature in a turbine entrance T_g turbine was not above 1200...1300 K), and the thermal efficiency of such engines, as a rule, does not exceed 25...27 %. In this regard, when converting such engines into GTP, the need to find ways to improve their fuel efficiency becomes very topical problem.

The problem of improving the fuel efficiency of gas turbine engines used in terrestrial gas installations, i.e. reducing fuel consumption per unit of useful energy they produce, is important in two aspects. On the one hand, it improves the economic performance of enterprises using such installations, as the cost of fuel used by such installations to create a unit of the useful energy they produce is the largest share (part) of the cost of these useful products. On the other hand, reducing the amount of burned fuel reduces emissions of combustion products into the atmosphere, which reduces the harmful impact of engines on the environmental situation not only near the location of enterprises that use gas turbines. This second aspect of this problem has taken on particular importance in recent decades, when not only on a one-city or one country, but also globally, the challenge of reducing the harmful effects of gas turbines becomes very important for the environment protection.

Crux of problem

In various investigations devoted to solving the problem of increasing the fuel efficiency of ground-based gas turbine plants, several methods for solving it are considered.

Among these methods, the main one is to increase the parameters of the operating process and, above all, the degree of increase in pressure in the cycle, the gas temperature in entrance of the turbine, as well as the increase in the efficiency of the compressor, combustion chamber and turbine units [1; 2; 3].

All these methods, of course, can be effective in the design of new gas turbine plants, but they certainly require the use of new and expensive materials for the manufacture of structural elements of gas turbines. As applied to gas-turbine units, created on the basis of aviation gas-turbine engines, which have exhausted their service life time on aircraft, the most acceptable way to increase fuel efficiency is to use complex cycles of their operation.

Analysis of recent research and publications

Different aspects of creating ground-based power plants based on aircraft gas turbine engines have been discussed in mane scientific publications.

In particular, in publications [1; 3; 4] several transformations of main structural engine units and functional systems which should be performed during converting aircraft engines into ground base installation are analysed.

In the works/1,5/recovering some portion of the heat of exhaust gases removed from the turbine, is called as one of the possible ways to increase the efficiency of gas turbines created on the basis of converted aircraft engines.

However, in these works there are no quantitative estimates of degree of influence of regeneration on thermal efficiency of engine at different engine operating conditions and first of all at different air temperatures at the engine inlet (and therefore for different climatic conditions of operation). In publication [6] investigation of fuel temperature influence on characteristics of thermodynamic cycle is discussed in details. But this investigation concerns mainly not gas turbine engines but ram jet engines.

Aim of this paper

In the framework of this study, we consider two ways to solve the problem of increasing the fuel efficiency of ground-based gas turbines created on the basis of aircraft turboprops and turboshaft engines that have fulfilled the flight resource: the use of heat recovery from exhaust gases coming from the engine and the return of this heat to the gas turbine's duty cycle and preheating the fuel before it feeding into the combustion chamber.

The efficiency of the heat recovery effect on increasing the thermal efficiency of the gas turbine cycle created on the basis of a turboprop engine with relatively low parameters of a simple Brayton cycle was considered in detail in [4].

Calculations carried out in that article using the thermophysical properties of the working fluid in the form of an ideal gas showed that the use of heat recovery from gases leaving the turbine and returning it to the working cycle by heating the air compressed by the compressor before it is fed into the combustion chamber, the thermal efficiency of the cycle can be increased by 2–3 percent at different regeneration rates.

Since, as it was shown in [4], the temperature of the exhaust gas after the regenerator still remains significant (more than 300 °C), this allows us to propose another way to increase the efficiency of ground-based gas turbine plant due to preheating the fuel by products of burning coming out of regenerator before fuel is fed to the combustion chamber**.**

The purpose of this study is to assess the combined effects of heat regeneration and fuel preheating on fuel consumption and gas turbine power, as well as analysis of the impact of these two factors on the effective use of fuel in GTP.

The calculations have been carried out taking into account the differences of thermophysical properties of the real air in a compressor and real products of combustion in a combustion chamber and turbine as compared to the ideal gas properties usually taken in most researches, including [4].

This article focuses on assessing the effect of preheating the fuel before it is fed into the combustion chamber on the fuel efficiency of gas turbine units built on the basis of aircraft engines that have exhausted life time on airplane or helicopter during their operation in various climatic conditions**.**

To assess the efficiency of the gas turbine process, we will use the generally accepted [5; 6; 7] formula for calculating the thermal efficiency of the Brayton cycle

$$\eta_t = \frac{q_1 - q_2}{q_1} = 1 - \frac{q_2}{q_1}, \qquad (1)$$

where η_t is the thermal efficiency of the cycle; q_1 is the amount of heat released when the supplied fuel is completely burned in the combustion chamber, and q_2 is the amount of heat lost with the combustion products leaving the engine, and also due to heat transfer to the structural elements of the hot part of the engine and incomplete combustion of fuel in the combustion chamber.

To ensure a decrease in the fuel consumption supplied to the combustion chamber, it is proposed to preheat it through the use of secondary energy resources, namely due to the utilized heat of the exhaust gas exiting the engine turbine.

The thermal efficiency of the cycle with the regeneration and heating of the fuel with exhaust gas will be calculated by the formula

$$\eta_t = \frac{w}{q_1 - q_{reg} - q_{fh}},$$

where w is the work of the cycle; q_1 is the heat supplied in the combustion chamber; q_{reg} — heat supplied to the working fluid in the regenerator; q_{fh} — the amount of heat spent for heating the fuel.

We will evaluate the fuel efficiency by a coefficient representing the ratio of the fuel consumption spent on useful work of the cycle and fuel consumption spent for heating the gases

$$\eta_{fuf} = \frac{G_1 - G_2}{G_1} = 1 - \frac{G_2}{G_1}, \qquad (2)$$

where G_1 is the mass of fuel proportional to the amount of heat transferred in the combustion chamber to the working fluid per unit time; G_2 is the mass of fuel spent on heating the gases leaving the engine and other heat losses. The ratio described by equation (2) will be called the "fuel utilization factor" and denoted as η_{fuf}.

Provided that a constant initial temperature of the air at the engine inlet and a constant maximum gas temperature in the cycle are maintained, the amount of fuel spent on heating the exhaust gases G_2 does not depend on whether this fuel was heated up or not, before being transferred to the combustion chamber, at that time as the amount of actually consumed fuel, taking into account increased in its enthalpy due to preheating, obviously, will be required less.

Denote this downward-corrected amount of fuel as G_{1cor}.

In accordance with this, formula (2) can be converted to next form as:

$$\eta_{fuf} = \frac{G_1 - G_2}{G_{1cor}}, \qquad (3)$$

where η_{fuf} is the fuel utilization factor; G_{1cor} is the adjusted amount of preheated fuel.

The corrected amount of preheated fuel G_{1cor} can be represented as the difference in the mass of fuel proportional to the amount of heat G_1 and the mass of fuel G_{preh} proportional to the amount of heat supplied to the fuel during its preheating:

$$G_{1cor} = G_1 - G_{preh}. \qquad (4)$$

In further calculations, we turn to the consideration of the amounts of heat corresponding to the combustion of one kilogram of fuel and the heat spent on heating one kilogram of fuel if the fuel was heated by burning a certain amount of the same fuel.

The amount of heat Q_{fph} required for preheating one kilogram of fuel from the initial temperature (we take T_{in} = 0 °C) to the final temperature T_f we calculate as:

$$Q_{fph} = c_p \Delta T = c_p \left(T_f - T_{in} \right),$$

where c_p is the specific heat of the fuel, ΔT is the degree of fuel heating.

The amount of fuel G_{fph} used to obtain the amount of heat Q_{fph} for preheating one kilogram of fuel is determined taking into account the lower calorific value of the fuel Q_f^L as:

$$G_{fph} = Q_{fph} / Q_f^L.$$

Taking into account the effect of the fuel preheating, we determine the corrected amount of fuel G_{1cor}, which must be fed into the combustion chamber

$$G_{1cor} = G_1 - G_{fph}.$$

After these transformations, formula (3) takes the form:

$$\eta_{fuf} = \frac{G_1 - G_2}{G_{1cor}} \text{ or } \eta_{fuf} = \frac{G_1 - G_2}{G_1 - G_{fh}},$$

In this study in order to analyze the influence of these two effects on the energy efficiency of a gas turbine plant based on the PW-6A aircraft turboprop engine that has spent its flight life time and the fuel utilization coefficients were calculated for cycles with constant rate of regeneration and different degreases of the fuel preheating.

These calculations were performed for different initial temperatures of the working fluid that corresponds different climatic conditions.

The calculation method is based on the algorithm for determination the parameters of a GTP operating according the Brayton cycle, the calculation scheme of the plant is shown in Fig. 1 [5].

The proposed gas turbine plant should have two types of heat exchangers:

– regenerator (recuperator), which returns part of the heat of the exhaust gases to the cycle (Fig. 1, *b*), heat exchanger included in the GTP cycle;

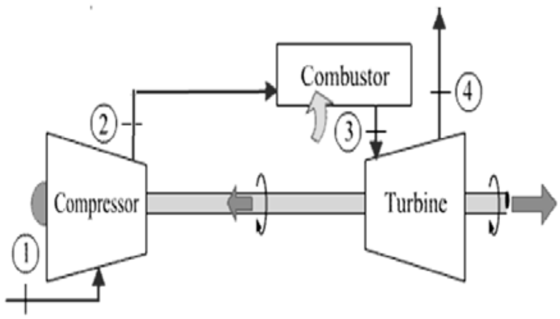
– fuel heater (is not included in the GTP cycle), the exhaust gases are the sources of this heat.

These two heat exchangers can be combined into one unit, as suggested in the patent [8], and are shown in Fig. 2.

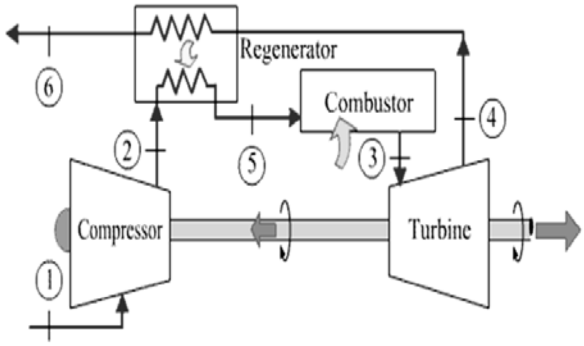
At the same time, exhaust gases are sent primarily to the first section 1 for heating the compressed air compressor before it is fed into the combustion chamber, and after that they are sent to the second section 2 for preheating fuel.

When fuel is burned, combustion products are formed in gas turbine plants.

They carry a large potential of secondary heat energy sufficient both as to heat the compressed air in the compressor and to preheat the fuel.



a



b

Fig. 1. Schemetic diagrams of GTP, operating with different Brayton cycles:
a — conventional basic Brayton cycle; *b* — regenerative Brayton cycle

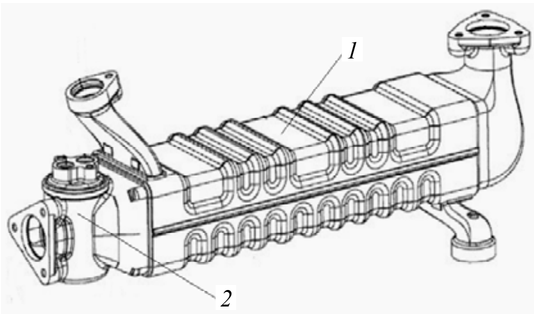


Fig. 2. Doubled heat exchanger [8]: *1* — gas-to-air section; *2* — gas-to-fuel section

Heating the fuel leads to an increase in the enthalpy of the fuel and allows to reduce its consumption. To preheat fuel by exhaust gases, the heater is installed on a branch from the exhaust tract, that is, on the bypass, which allows regulate the mass flow rate of exhaust gas through the heater and even completely disconnect it.

Justification of choice of engine

The engine PW-6A was considered as a part of a ground based Gas Turbine Installation (GTI) operating at reduced modes, which is necessary to obtain acceptable service life time of the turbine assembly parts (operation time usually 50–60 thousand hours is accepted). The following initial data were taken:

– air pressure at the inlet of the engine: p_1 = 10103 Pa;

– pressure ratio: π = 6.3 :1;

– air temperature at the compressor inlet: T_1 = 288 K;

– the temperature of the combustion products before the turbine T_3 = 1340 K;

– lower calorific value of kerosene Q_f^L = 40, 8 MJ/kg;

– degree of regeneration: σ = 0.8;

– the internal relative efficiency of the compressor and turbine η_c = 0.89, η_t = 0.91, respectively.

We calculated the value to which the fuel utilization factor is increased due to preheating of kerosene. The calculation has been done for one kilogram of kerosene. The fuel utilization factor can be expressed as ratio of the quantity of fuel spent in the initial cycle (without fuel preheating) to heat the working fluid (gases) to the amount of fuel actually spent for heating of working fluid in cycle, reduced by amount of fuel, that was saved due to fuel preheating. Amount of fuel used in basic cycle for useful purposes is determined as the difference of the spent fuel mass and the fuel equivalent to heat loss with the exhaust gases. The corrected fuel utilization factor for one kg of fuel is determined as:

$$\eta_{fu\ cor} = \frac{\left[1-\left(1-G_{f\,ac}\right)\right]}{1-G_f},$$

where G_f — heating mass of fuel; $G_{f\,ac}$ — actual amount of fuel used to heat working fluid.

One of the ways to improve the modern mathematical models of gas turbine engines used in thermodynamic calculations is the calculation of the thermodynamic properties of the working fluid (air and combustion products as real gases). The calculation of the properties of the working fluid as real gases is currently based on a number of basic assumptions:

– fuel combustion is complete with an excess air coefficient $\alpha \geq 1.0$, combustion products-non-reactive mixture of CO_2, water vapor, O_2 and atmospheric nitrogen, the volume composition of which depends only on the value of α and the fuel composition;

– working fluids (air and combustion products) are a mixture of components that have the properties of an ideal gas, with constant thermodynamic properties that depend only on temperature.

If all components of the fuel are completely oxidized in the theoretically necessary amount of air (α =1) and there is no heat loss, the temperature of the combustion products will be as high as possible.

As is well known, the actual combustion temperature depends on the heating value of the fuel, the initial temperature of the fuel, the state of the fuel and the oxidizer, overall combustion efficiency, heat transfer conditions, etc.

The specific amount of heat released as a result of the combustion of the air — fuel mixture in the combustion chambers of the gas turbine engine is calculated as:

$$q_1 = c_p\left(T_3 - T_2\right),$$

where c_p — the isobaric specific heat of gases formed during the combustion; T_2, T_3 — air temperature at the outlet of the compressor and the gas turbine inlet.

This form of the equation for determination the amount of heat is considered fair and this expression is generally accepted.

For calculation the temperature of the combustion products, it is assumed that all the rejected heat Q is transferred to the combustion products. Thus, the temperature of the combustion products depends on the amount of heat Q, the mass of combustion products and their specific heats, as well as the initial temperature of the working fluid. The amount of heat can be determined from the equation:

$$Q = m_c c_{pc}\left(T_3 - T_2\right),$$

where m_c is the mass of combustion products; c_p is the isobaric specific heat of the combustion products; T_2 is the temperature of the working fluid at the outlet of the compressor, T_3 is the temperature of the working fluid at the inlet of the turbine.

In the combustion process the heat supply occurs with variable parameters, therefore, the thermodynamic properties of the working fluid and fuel are also not constant. Burning process begins under conditions when the working fluid is almost pure air with a temperature of T_2, and the atomized fuel, as a rule, has a temperature different from T_2. The specific heat of air and fuel (fresh fuel-air mixture), of course, differs from the specific heat of combustion products. In addition, the specific heat of different substances of combustion products is not the same, so

$$Q = \sum m_j c_{pj}\left(T_3 - T_2\right),$$

where m_j is the mass of the j-th component of combustion products; c_{pj} is the specific heat at constant pressure of the j-th component of combustion products.

Since $c_{pj} = f(T)$, this expression for determining the amount of heat is valid only for an infinitesimal temperature range, i.e.

$$\delta Q = \sum m_j c_{pj} dT,$$

where expression $m_j c_{pj} dT$ is the change in the enthalpy of the j-th component of the combustion products.

Increasing the accuracy of calculation of the Brayton cycle processes of real gas requires more strictly determination of the working fluid thermodynamic properties over a wide range of temperatures and pressures. Calculation the relationship among the temperature and pressure of a gas in adiabatic compression and expansion processes requires accurate knowledge of the adiabatic exponent. The validity and correctness of determination of the specific heat c_p of working fluid, the adiabatic exponent k of the compression and expansion processes depend on the reliability of evaluation of the efficiency of the GTP as a whole. Not taking into account changes in the thermodynamic properties of the working fluid leads to large errors in the calculations of real efficiency of gas turbine plants in the direction of their increase in comparison with efficiency of real GTP. It is enough to note that the main thermodynamic properties of air (the working fluid in the compressor) differ by 20–30 % if compared with the thermodynamic properties of gases (products of burning). The customers of GTP, in case of using GTP at the thermal power station, will have real losses in the form of unrealized benefits and not confirmed economic parameters of gas turbine plant.

The burning process begins only when fuel droplets are transforms into vapour. When air is mixed with fuel vapours, a gas mixture is formed with new parameters that differ from the original ones. During burning process calculation, it is necessary to know the values of the expected parameters of the working fluid (combustion products). This is especially important for transient modes. There is no solution to this problem in the literature. The papers [9; 10] contain information about the thermodynamic properties of gaseous substances and gas mixtures. However, these results are not complete, or they are obtained for a limited set of initial data and conditions that affect thermodynamic parameters.

Determination of the thermophysical properties of the working fluid

In this paper, the parameters of the gas mixture state were determined using the known parameters of state of the gas mixture components.

Consider the concrete working fluids (air, fuel, and combustion products).

To determine the values of the isobaric specific heat c_p, the gas constant *R*, and the adiabatic exponent *k* of working fluids, it is necessary to know their chemical composition.

The volume composition of the air was taken as a percentage according to ISO 6976 [9] data. It is given in table 1.

Table 1

Air volume composition

Elements	N_2	O_2	Ar	CO_2	Ne	He	CH_4	Kr	H_2	NO_2	CO	Xe
Volume fraction, %	78.102	20.946	0.916	0.033	0.0018	0.0005	0.00015	0.0001	0.00005	0.00003	0.00002	0.00001

The composition of combustion products is determined by their temperature, pressure, and chemical composition of the fuel.

The composition of the combustion products can be determined by the given gas temperature at the outlet of the combustion chamber T_3, the excess air coefficient α, and the thermal balance written for the combustion chamber.

Table 1 values of the true specific heat of the main components of air and combustion products were taken as initial data [9; 10; 11].

The calculation of isobaric specific heats of the main components of combustion products in a given range of pressures and temperatures was performed using analytical expressions obtained as a function of temperature and pressure, taking into account the effect of thermal dissociation in the paper [10].

Specific heat graphical dependence on temperature and pressure $c_p = f(T, p)$ represents the surface.

As example the nature of the change in the nitrogen isobaric specific heat depending on pressure and temperature is shown in Fig. 3 [10].

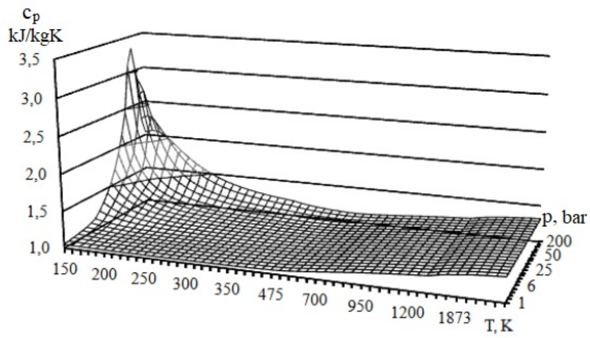


Fig. 3. Dependence isobaric specific heat of nitrogen from temperature and pressure [9]

Table 2 shows the calculated dependencies for the main components of the combustion products in the specified pressure range of 0.1 ... 200 bar and temperatures of 150...2870 K.

Table 2

Dependencies for calculation of isobaric specific heat

Gas	Isobaric specific heat	Coefficients
Nitrogen	$c_p\big\|_{T=\text{const}} = \sum_{j=0}^{4} X_j p^j$	$\{X_j\}$ — coefficients depending of temperature
Oxygen	$c_p\big\|_{T=\text{const}} = \sum_{j=0}^{4} X_j p^j$	
Argon	$c_p\big\|_{T=\text{const}} = X_0 + X_1 p$	
Steam	$c_p\big\|_{T=\text{const}} = \sum_{j=0}^{5} X_j p^j$	
Carbon dioxide	$c_p\big\|_{T=\text{const}} = \sum_{j=0}^{5} Y_j T^j$	$\{Y_j\}$ — coefficients depending of pressure

The gas constant. Adiabatic exponent

The gas constant is numerically equal to the ratio of the universal gas constant to the molar mass of the substance. The specific gas constant for a mixture of gases R_m is equal to the sum of the products of the gas constants of individual gases R_i by their mass fractions g_i, i.e.:

$$R_m = \sum_{i=1}^{j} R_i g_i$$

or with sufficient accuracy it can be determined by the formula:

$$R_m = 8314{,}41 \sum_{i=1}^{j} \frac{g_i}{M_i} = \frac{8314{,}41}{\sum_{i=1}^{j} M_i r_i},$$

where g_i and r_i are the mass and volume fractions of i-th component in the mixture; M_i is the molar mass of i-th component.

The adiabatic exponent for pure gas:

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

The adiabatic exponent of combustion products depends on the parameters of the gas state (pressure and temperature) and the composition of the mixture and is determined by the formula

$$k = \sum_i r_i k_i,$$

where k_i is the adiabatic exponent of the i-th component of the mixture, r_i is the volume fraction of the i-th component.

The calculated dependences of the specific heat and adiabatic exponent for air and combustion products are shown in Fig. 4, 5.

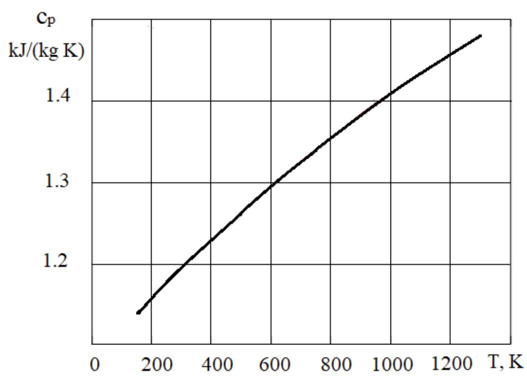


Fig. 4. Change of the air specific heat depending

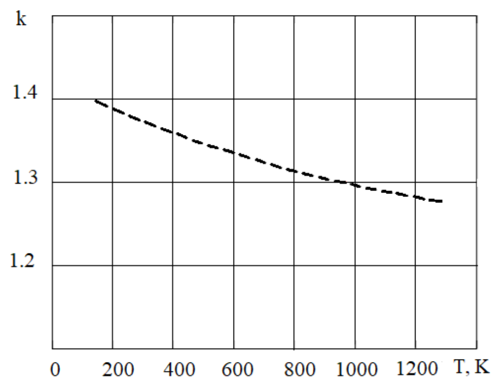


Fig. 5. Dependence of the air adiabatic exponent on temperature on temperature

Results of investigation

As part of this work, calculations were made of values the fuel utilization factor η_{fuf} and the thermal efficiency of the cycle for the GTP operating on the simple cycle of Brighton (i.e. without regeneration of heat of the exhaust gases), but with preliminary heated liquid fuel (aircraft kerosene brand TS-1). The calculation was made for different degrees of fuel heating from zero degrees Celsius to two hundred degrees Celsius. The results of these calculations are presented in table 3.

As it can be seen from the results of the calculations, when fuel is heated by the transfer of heat from the outgoing gases, the fuel utilization factor increases as the fuel is heated. At the same time the thermal efficiency factor of the GTU cycle increases too from 0,30 up to 0,307 i.e. by 0,98 %.

In addition, preheating the fuel helps to improve the reliability of the engine starting process, especially at low ambient air temperatures.

According to the proposed method of calculation of the fuel utilization factor for operational cycle with preheating of fuel before it is given to combustor, described above and its influence on thermal efficiency of such cycle, when engine operates in accidence with simple Brayton cycle without regeneration of heat taking into account the thermal properties of the real working body in the GTP, the following results were obtained (table 3).

Table 3 provides the results of calculations made for liquid fuel (aviation kerosene brand TC-1).

The lower calorific value of kerosene Q_f^L = 40, 8 MJ/kg [12; 13] when it is heated from the initial temperature of T_{in} = 0 °C up to different final temperatures of T_f. = 50 °C … 200 °C.

Preheating kerosene to higher temperatures does not seem feasible, as in this case it is not effective in the primary combustion chamber area with the air with a lower temperature than the fuel temperature.

Table 3

Results of fuel preheating evaluation

Final temperature of fuel T_f	0 °C	50 °C	100 °C	125 °C	150 °C	200 °C
Amount of heat carried in with preheated fuel Q_{fph}, кJ/кg	0	103,50	216,00	282,00	347,00	494,0
Amount of fuel G_{fph} used to obtain the amount of heat Q_{fph}, kg	0	0,0025	0,0053	0,007	0,0085	0,012
Corrected thermal efficiency of cycle, %	30,0	30,07	30,16	30,2	30,28	30,73
Rate of increasing the thermal efficiency of cycle due to fuel preheating, %	–	0,255	0,53	0,7	0,86	0,98

Rate of increasing the thermal efficiency of cycle due to fuel preheating is show in Fig. 6. The thermal efficiency increases with increase degree of fuel preheating. Slope of the curve is more at low degree of fuel preheating.

Graphs have been drawn according the calculated parameters. The calculated dependences of the thermal efficiency change for different air temperatures at the compressor inlet, taking into account change in the thermodynamic properties of the working fluid depending on the temperature (solid line) and for the thermodynamic properties of working body as ideal gas (dotted line) are shown in Fig. 7. Calculation done for ideal gas properties of the working fluid leads to increased results in efficiency of gas turbine plants about by 2 % as compared with efficiency determined for real gas working fluid.

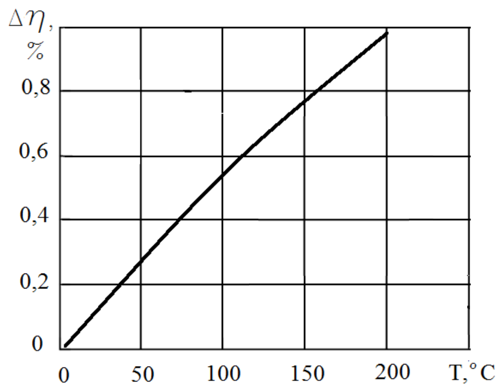


Fig. 6. Rate of increasing the thermal efficiency of cycle due to fuel preheating

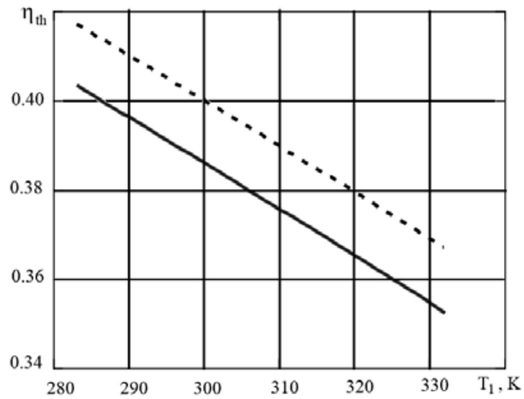


Fig. 7. Changing the thermal efficiency for different air temperature T_1 at the inlet of the compressor

Fig. 8 shows the estimated dependence of the cycle thermal efficiency change for different air temperatures T_1 at the entrance the compressor with regeneration (σ = 0,8) (solid line) and for cycle with the same degree of regeneration and fuel preheating up to 200 °C (dotted line). As it is seen the effect of fuel preheating on the thermal efficiency can reach approximately by 1 % especially at low ambient temperatures. At high ambient temperatures this effect is considerably reduced. Even more clearly, the impact of these tways to improve the fuel efficiency of a gas turbine plant based on convertible aircraft engines with low working process parameters is shown in Fig. 9.

A generalized graph illustrating the extent of the effect of heat regeneration and preheating of fuel on the thermal efficiency of the gas turbine plant is shown in Fig. 9. Here, line *1* corresponds to a gas turbine plant operating on a basic Brayton cycle, line *2* corresponds to a cycle with heat regeneration, and line *3* corresponds to a gas turbine plant operating on a cycle with heat regeneration and with preheated fuel.

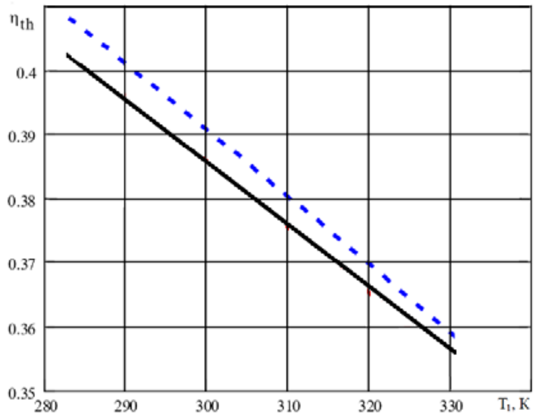


Fig. 8. Dependences of the thermal efficiency for different air temperature T_1 at the inlet of the compressor

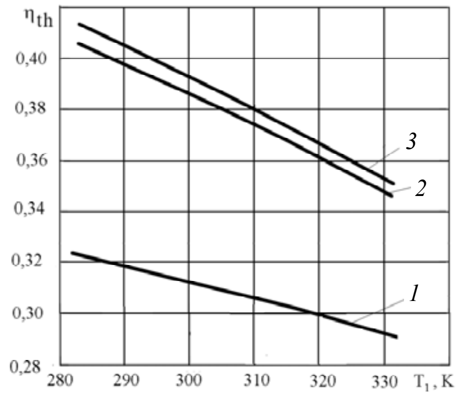


Fig. 9. Variation of thermal efficiency of GTP, operating in accordance with different cycles: *1* — basic Brayton cycle; *2* — cycle with regeneration; *3* — cycle with regeneration and fuel preheating

As it is visible in the grath the most considerable increase in the thermal efficiency (about 8 %) is achived due to regeneration. This effect is more considerable for low ambiante temperatures.

A comparative estimation of the distribution of fuel consumption in the GTU for the useful work of the cycle, for fuel preheating and for thermal losses is shown in Fig. 10.

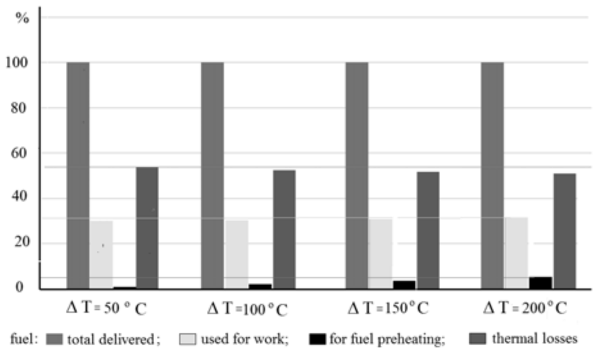


Fig. 10. Distribution of fuel consumption in the GTU

As can be seen from the diagrams, when fuel is heated, the part of fuel spent for the useful work of the cycle increases, and the part of fuel associated with the loss of heat with exhaust gases decreases, that causes an increase in the fuel efficiency of the GTU.

Conclusions

As a result of the calculations, the following conclusions can be made:

1. Heat regeneration, heating the working fluid (air) after it is compressed in the compressor and before being fed into the combustion chamber by exhaust gases leaving the engine is a fairly effective way to increase the fuel efficiency of terrestrial gas turbine plants created on the basis of aircraft engines that have spent their flight life on airplanes and helicopters.

2. Preheating the liquid fuel using for this purpose the heat of the exhaust gases coming out of the engine helps to increase the coefficient of fuel utilization. The effect of preheating the fuel in this way is manifested through the intensification of the evaporation process of fuel droplets sprayed by jet fuel injectors and the formation, thus forming a homogeneous fuel-air mixture in the combustion zone of the combustion chamber.

3. Preheating of gaseous fuel (natural gas) has virtually no effect on engine fuel efficiency and the coefficient of fuel utilization, but can be used as a way to improve the reliability of starting a gas turbine plant, especially when it is operated at low temperatures and low ambient air pressures (in high-mountain conditions).

4. Joint using heat of exhaust gases leaving the engine for heating the working fluid (air) after its compression in the compressor before feeding it to the combustion chamber and for preheating the liquid fuel is the most rational way to improve the fuel efficiency of GTU created on the basis of aircraft engines that have spent their life on aircraft of civil aviation.

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ШЛЯХИ ПІДВИЩЕННЯ ЕФЕКТИВНОСТІ ЕНЕРГОУСТАНОВОК НА БАЗІ АВІАЦІЙНИХ ГАЗОТУРБІННИХ ДВИГУНІВ

Щорічно у світі велика кількість авіаційних двигунів, які були спроектовані і виготовлені 10–20 років тому і які вичерпали свій льотний ресурс на повітряних суднах цивільної і військової авіації, знімаються з експлуатації через зниження їхньої експлуатаційної надійності. При цьому більша часина їхніх конструктивних вузлів і деталей ще мають значні запаси міцності і можуть бути використані у складі наземних установок різного призначення. Рівень параметрів робочого процесу, і перш за все, ступеня підвищення тиску повітря в компресорі та температури газу перед турбіною, є відносно малими, і тому коефіцієнти корисної дії таких двигунів відносно низькі, а відповідно до цього, паливна ефективність таких двигунів не відповідає сучасним вимогам.

Проблема підвищення паливної ефективності стаціонарних газотурбінних установок, створюваних на базі авіаційних двигунів, що вичерпали свій ресурс на повітряних суднах, є актуальною з кількох причин. З одного боку, вартість палива становить значну частку в експлуатаційних витратах підприємств, що їх використовують, і негативно впливає на економічні показники цих підприємств. З іншого боку, значні витрати палива обумовлюють утворення великої кількості продуктів згоряння , що викидається в атмосферу, забруднюючи її. Для вирішення цієї проблеми в статті пропонується удосконалення робочого процесу таких силових установок шляхом регенерації тепла, що міститься у вихлопних газах і попередній підігрів палива перед його подаванням в камеру згоряння.

Метою роботи є оцінка рівня спільного впливу регенерації тепла вихідних газів і ступеня попереднього підігрівання палива на паливну ефективність газотурбінних установок.

Як об'єкт дослідження було обрано газотурбінну установку, яка базується на використанні авіаційного турбовального двигуна, що відпрацював свій льотний ресурс , з доданим до нього подвійним теплообмінником для підігрівання стисненого в компресорі повітря перед його надходженням до камери згоряння і для попереднього підігрівання палива. Проведеними розрахунками показано, що спільний ефект від цих двох заходів може суттєво підвищувати внутрішній коефіцієнт корисної дії газотурбінної установки. При цьому розрахунки проводились з урахуванням зміни теплофізичних властивостей повітря в компресорі і продуктів згоряння зі зміною температури як реальних газів.

Ключові слова: газотурбінний двигун; термодинамічні властивості; регенератор теплоти; попереднє підігрівання палива; ефективність.

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THE WAYS OF IMPROVING THE EFFICIENCY OF GAS TURBINE PLANTS BASED ON AIRCRAFT GAS TURBINE ENGINES

Every year in the world, a large number of aircraft engines designed and manufactured throught ten to twenty years ago and spent their flight service life time on civil and military aircraft are decommissioned due to a decrease of their operational reliability. In the same time, the most of their structural units and parts still have significant safety margins and can be used as part of ground installations for various purposes. The level of parameters of the working process of these engines, and first of all, the compressor pressure ratio of the compressor and the temperature of the gases in front of the turbine, is relatively low, therefore the efficiency of such engines is relatively low, and therefore the fuel efficiency of such engines does not meet modern requirements.The problem of improving the fuel efficiency of stationary gas turbine plants based on aircraft engines that have exhausted their life on aircraft is urgent for several reasons. On the one hand, the cost of fuel is a significant part of the operating costs of the enterprises that use them, and adversely affects the economic performance of these enterprises. On the other hand, significant fuel consumption causes the formation of large quantities of combustion products, which are emitted into the atmosphere, polluting it. To solve this problem, in article it proposes to improve the working process of such power plants by Regeneration of the Heat of Exhaust Gases and preheating the fuel before it is fed into the combustion chamber.

The purpose of this work is to evaluate the level of joint effect of heat recovery of the exhaust gases and the degree of preheating of the fuel on the fuel efficiency of gas turbine plants. The object of the study was a gas turbine plant based on an aircraft turbo engine that exhausted its flight life. It was proposed to add a dual heat exchanger into gas turbine plant to heat the compressed air before entering the combustion chamber and preheating the fuel. The calculations show that the combined effect of these two measures can significantly increase the internal efficiency of a gas turbine plant. The calculations were carried out taking into account changes in the thermophysical properties of air in the compressor and combustion products with changes in temperature as for real gases.

Keywords: gas turbine engine; thermodynamic properties; heat regeneration; fuel preheating; efficiency.

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ПУТИ ПОВЫШЕНИЯ ЭФФЕКТИВНОСТИ ЭНЕРГОУСТАНОВОК НА БАЗЕ АВИАЦИОННЫХ ГАЗОТУРБИННЫХ ДВИГАТЕЛЕЙ

Ежегодно в мире большое количество авиационных двигателей, спроектированных и изготовленных десять-двадцать лет тому назад и отработавших свой лётный ресурс на воздушных судах гражданской и военной авиации, снимаются с эксплуатации из-за снижения уровня их эксплуатационной надежности. При этом большая часть их конструктивных узлов и деталей еще имеют значительные запасы прочности и могут быть использованы в составе наземных установок различного назначения. Уровень параметров рабочего процесса, и прежде всего, степени повышения давления воздуха в компрессоре и температуры газов перед турбиной, относительно невысокие, поэтому коэффициенты полезного действия таких двигателей относительно низкие, а значит и топливная эффективность таких двигателей не соответствует современным требованиям. Проблема повышения топливной эффективности стационарных газотурбинных установок, создаваемых на базе авиационных двигателей, которые исчерпали свой летный ресурс на воздушных судах, является актуальной по нескольким причинам. С одной стороны, стоимость топлива составляет значительную долю в эксплуатационных расходах предприятий, которые их используют, и негативно влияет на экономические показатели этих предприятий. С другой стороны, значительные расходы топлива обуславливают образование большого количества продуктов сгорания, которые выбрасывается в атмосферу, загрязняя ее. Для решения этой проблемы в статье предлагается усовершенствование рабочего процесса таких силовых установок путем регенерации тепла отработанных газов и предварительного подогрева топлива перед его подачей в камеру сгорания.

Целью работы является оценка уровня общего воздействия регенерации тепла отходящих газов и степени предварительного подогрева топлива на топливную эффективность газотурбинных установок.

В качестве объекта исследования была выбрано газотурбинная установка, которая базируется на использовании авиационного турбовального двигателя, отработавшего свой лётный ресурс, с добавлением в его схему сдвоенного теплообменника, который служит как для подогрева сжатого в компрессоре воздуха перед его поступлением в камеру сгорания, так и для предварительного подогрева топлива.

Проведенным расчетам показано, что общий эффект от этих двух мер может существенно повысить внутренний коэффициент полезного действия газотурбинной установки. При этом расчеты проводились с учетом изменения теплофизических свойств воздуха в компрессоре и продуктов сгорания с изменением температуры как реальных газов.

Ключевые слова: газотурбинный двигатель; термодинамические свойства; регенератор теплоты; предварительный подогрев топлива; эффективность.

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