Comparative analysis of methods for increasing fuel efficiency of gas turbine plants

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Abstract. The results of a calculated assessment of the effectiveness of various methods for increasing the fuel efficiency of gas turbine plants created based on converted aircraft engines that have exhausted their flight life on aircraft are given in the article. Converting aircraft engines into ground-based power plants is an important task because it reduces the time for their design and development, and provides a significant reduction in the cost of such plants. At the same time, in terms of fuel efficiency, aircraft engines do not meet the requirements for ground gas turbine plants and therefore need to be improved. The analysis of possible ways to improve the working process was carried out and the calculations of the efficiency and output power of gas turbine plants were done. Various methods to improve the fuel efficiency of ground-based plants were compared. Methods which were considered such as converting the heat of gases leaving the engine, preheating liquid fuel, injecting water vapor in order to cool the air before compression, using multi-stage compression with air intercooling. The object of the study was a low-power turboshaft aircraft engine with relatively low parameters of the working process, since it most closely meets the requirements for mobile power plants.
Calculations of the engine efficiency were carried out for different air temperatures at the engine inlet, which corresponds to the operation of power plants in different climatic conditions.

1. Introduction

One of the methods of development and modernization of the energy industry, which has been implemented in recent decades, is the use of aviation engineering technologies in it and, first, technologies and design solutions used in the design and manufacture of aircraft engines

A distinctive positive feature of highly efficient aircraft engines in comparison with ground-based power gas turbine plants for general use is their relatively low specific mass, defined as the ratio of the engine mass to the power generated by it. This is achieved by the high thermal and mechanical stress of its main parts during engine operation. However, the requirement to ensure a high level of flight safety and high reliability of these engines during operation leads to a significant limitation of the service life-time of such engines.

As a result, aircraft engines, as a rule, are removed from service on an aircraft, while still having significant reserves of strength and durability of its parts and assemblies. In this regard, in recent years, interest has grown in converting aircraft engines, that have exhausted their established flight life, into ground use gas turbine installations, which are widely used not only in the gas transmission system to drive centrifugal blowers as part of high-power gas pumping units, but also in other ground power plants different capacities.

However, at the same time, the level of fuel efficiency of aircraft engines, determined primarily by the level of thermal efficiency of theirs operating cycle often does not meet modern requirements for ground power plants.

Therefore, when converting aircraft gas turbine engines (GTE) for use in ground-based gas turbine plants (GTP), it becomes necessary to use various methods to improve their fuel efficiency. To date, there are several known ways to solve this problem, each of which is used for GTP of various purposes and is characterized by greater or lesser efficiency and implementation complexity.

2. Crux of problem and resent publications

In recent years, a large number of publications have been devoted to the problem of converting aviation GTEs. The most famous among them are the work performed by the employees of the design bureau under the direction of N.D. Kuznetsov. [1, 2], in which the main attention is paid to the analysis of the necessary structural changes of units and functional systems of aircraft engines of high thrust and power when they are converted into ground gas turbine units used as part of gas-pumping units of main gas pipelines, as well as for driving electric generators at electrical power plants. In this case in order to increase the energy efficiency of high-power ground-based GTP it is proposed to reduce heat losses with exhaust gases by using it to heat water in cogeneration units. At the same time, the coefficient of fuel heat utilization in such installations, taking into account the heat of water heated in the regenerators of utilization installations, is on average about 85% [3].

However, this method of increasing energy efficiency is unacceptable for mobile installations based on converted aircraft engines of low and medium power or thrust. Other traditional ways of increasing the efficiency of the flow parts of power gas turbines, such as the compressor and turbine assemblies, have practically exhausted themselves, since their efficiency has reached the level of 91–92%, which has brought these values very close to the theoretically possible values [4].

In this case, the main way to solve the problem of increasing the thermal efficiency of a gas turbine plant is to reduce heat losses with exhaust gases by converting this heat and returning it to the cycle in order to increase the internal efficiency of the gas turbine plant.

For this, the following possible directions for solving the problem are proposed:

- the use of complex thermodynamic cycles with intermediate cooling of air during its compression in the compressor;
- preheating of liquid fuel before it is fed into the combustion chamber;
- injection of water and steam into the air stream in the compressor;
- injection of water into the compressed air stream at the inlet to the combustion chamber;
- use of alternative fuel resources (types of fuels).

3. Aim of this investigation

The purpose of this investigation is a comparative assessment of various methods for increasing the fuel efficiency of ground-based GTP based on converted aircraft engines of low and middle power. In addition, the task was to evaluate the effectiveness of the joint simultaneous application of several of the known methods for solving this problem

4. Statement of the main research material

The calculation of a gas turbine unit operating according to a simple cycle with heat regeneration, the diagram of which is shown in Figure 1, was carried out in this study according to the methodology given in article [6] for such conditions:

- ambient air temperature $T_{amb} = T_1 = + 15$ ° C, pressure p_1 = 10133 kPa; fuel – aviation kerosene with the lowest calorific value Q_l = 50056 kJ / kg.

Thermal calculation method for gas turbines

Air temperature behind the compressor was calculated by formula:

$$T_2 = T_1\left[1 + \frac{1}{\eta_c}\left(\pi_c^{\frac{k-1}{k}} - 1\right)\right], \qquad (1)$$

where T_2 is actual temperature value at the end of the compression process; η_c is compressor efficiency is taken as η_c=0,85; k –specific heat ratio for air is taken as k = 1,4;

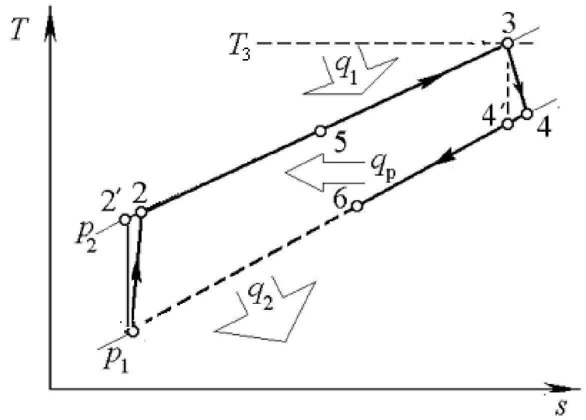

Figure 1. T-S diagram of the cycle with regeneration.

The exhaust temperature T4 from the turbine outlet is calculating (given) by formula:

$$T_4 = T_3\left[1 - \eta_t\left(1 - \pi_t^{\frac{1-k_g}{k_g}}\right)\right], \qquad (2)$$

where T_4 is the actual value of the temperature of the working fluid at the end of the expansion process; η_t is turbine unit efficiency, η_t=0,90; k_g is specific heat ratio for gas (product of burning), k_g = 1,33.

The heat supplied *q1* and removed *q2* in the actual cycle of the GTU can be calculated by the formulas:

$$q_1 = c_p\left(T_3 - T_2\right), \qquad (3)$$

$$q_2 = c_{pg}\left(T_4 - T_1\right), \qquad (4)$$

where c_{pa} and c_{pg} are isobaric specific heat of air and isobaric specific heat of gas, respectively.

Using expressions for determining the actual temperatures T2 and T4, we write down the formulas for calculating q_1 and q_2:

$$q_1 = c_p T_1\left[\frac{T_3}{T_1} - \left(1 + \frac{\pi_c^{\frac{1-k}{k}} - 1}{\eta_c}\right)\right], \qquad (5)$$

$$q_2 = c_{pg} T_1 \left\{ \frac{T_3}{T_1} \left[1 + \eta_t \left(\frac{1}{\pi_t^{\frac{1-k_g}{k_g}}} \right) \right] - 1 \right\} \quad . \tag{6}$$

Thermal efficiency of a real cycle of a GTP with irreversible compression in the compressor unit and expansion in the turbine unit are determined by formulas:

$$\eta_{\mathrm{i}} = 1 - \frac{q_2}{q_2}. \tag{7}$$

Through mathematical expressions for determining the amount of heat supplied in the cycle q_1, determined by formula (5) and the amount of heat q_2 taken by the turbine and calculated by formula (6), we transform formula (7) into a form convenient for direct calculation of the thermal efficiency of the cycle η_{th}:

$$\eta_{th} = \frac{c_{pg} T_3 \eta_t \left(1 - \frac{1}{\pi_t^{\frac{k_g - 1}{k_g}}} \right) - c_p T_1 \left(\frac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c} \right)}{c_p T_1 \left[\frac{T_3}{T_1} - \left(1 + \frac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c} \right) \right]} \quad . \tag{8}$$

The results of calculations, illustrating the dependence of the thermal efficiency of the cycle on the air temperature at the engine inlet when engine operates in accordance with a simple Brighton cycle (we will call it for brevity the “base cycle”) were represented in article [7], and are shown in figure 2 by a pair of lines indicated by the number 1. In this case, the solid line corresponds to the operation of the engine in a simple cycle, and the dotted line corresponds to the operation of the engine for the same cycle, but with preheating of liquid fuel by 80 degrees in special heat exchanger. The heat for this purpose is taken from the gas flow after the turbine.

As can be seen from the graphs, as the air temperature at the engine inlet rises, when the engine is operating in both cycles, the thermal efficiency of the engine decreases significantly. When the air temperature at the engine inlet rises from + 10 ^{o}C degrees to + 40 ^{o}C the thermal efficiency of “base cycle” decreases from η_t= 0,31 to η_t= 0,29.

In this case, the effect of preheating the fuel is insignificant and practically does not depend on changes in the air temperature at the engine inlet, since the dotted line is located almost equidistantly relative to the solid line. The physical mechanism of the influence of liquid fuel preheating on the efficiency of the engine was considered in the article [7].

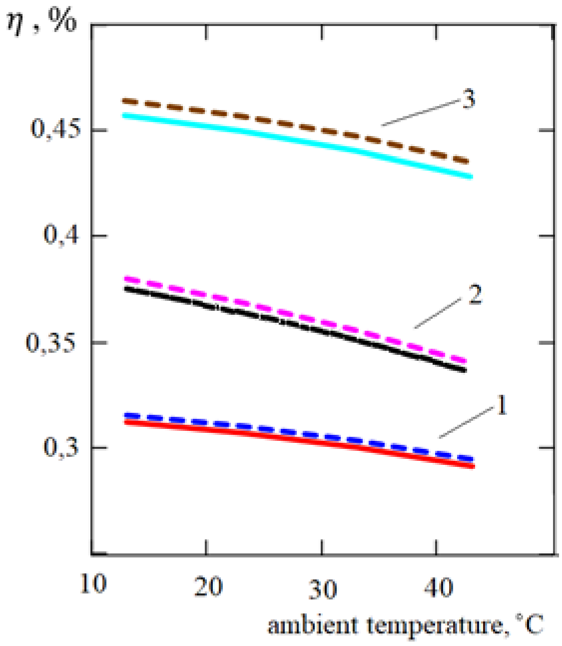

Figure 2. Thermal efficiency vs ambient temperature at fixed pressure ratios.
1 – basic cycle and basic cycle with fuel heating; 2 – basic cycle with regeneration and basic cycle with regeneration and fuel heating ; 3 – cycle with 2 – stage compression and intermediate cooling and regeneration; upper line – cycle with 2-stage compression and intercooling, regeneration and fuel heating

Exhaust gas heat regeneration
Schematic diagram of a gas turbine plant operating on a cycle with heat recovery of exhaust gases for heating the air compressed in the compressor before feeding it into the combustion chamber is shown in Figure 3.

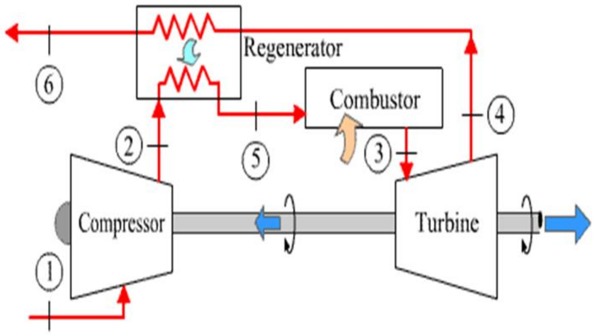

Figure 3. Schematic diagram of gas turbine plant with heat regenerator.

A main feature of the thermodynamic calculation of the heat efficiency of such a gas turbine plant is the need to calculate the air temperature after the T_5 regenerator, which depends on the effectiveness of the regenerator and the temperature difference behind the T_4 turbine and after the T_2 compressor.
The temperature at the end of regeneration T_5 is defined by the effectiveness of the regenerator σ_r :

$$T_5 = T_2 + \sigma_r \left(T_4 - T_2\right). \tag{9}$$

Ehe heat supplied to the working fluid in the regenerator is determined in accordance with the formula:

$$q_{\mathrm{r}} = \sigma_r c_c \left(T_4 - T_2\right) \tag{10}$$

The cycle thermal efficiency in this case is determined by next formula:

$$\eta_{th} = \frac{w_{net}}{q_1}, \tag{11}$$

where is heat converted into useful output work of cycle.

For direct calculation of the thermal efficiency of the cycle η_{th} in this case it is possible to use next formulas:

$$\eta_{th} = \frac{c_{pg} T_3 \eta_t \left(1 - \frac{1}{\pi_t^{\frac{k_g - 1}{k_g}}} \right) - c_p T_1 \left(\frac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c} \right)}{c_p \left[(T_3 - T_2) - \sigma_r (T_4 - T_2) \right]} \tag{12}$$

$$\eta_{th} = \frac{c_{pg} T_3 \eta_t \left(1 - \frac{1}{\pi_t^{\frac{k_g - 1}{k_g}}} \right) - c_p T_1 \left(\frac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c} \right)}{c_p \left[\left(T_3 - T_1 \left[1 + \frac{1}{\eta_c} \left(\pi_c^{\frac{k-1}{k}} - 1 \right) \right] \right) - \sigma_r \left(T_4 - T_1 \left[1 + \frac{1}{\eta_c} \left(\pi_c^{\frac{k-1}{k}} - 1 \right) \right] \right) \right]} \tag{13}$$

Pair of lines 2 in figure 2 show result of the calculation of the thermal efficiency for the cycle with the regeneration (effectiveness for regeneration equal 0.8). Here solid line 2 corresponds to the base cycle with heat regeneration and dot line 2 corresponds to the base cycle with regeneration and with fuel preheating.

As can be seen from these graphs, the heat regeneration of the exhaust gases greatly increases the efficiency of the engine. Therefore, at an air temperature at the inlet to the compressor + 15 degrees Celsius, efficiency for the base cycle increases from 0.32 to 0.38. However, this efficiency gain decreases significantly with increase in compressor inlet air temperature. That is, the effect of heat recovery from exhaust gases at high ambient temperatures is significantly lower than at low temperatures. This can be explained by the fact that the air temperature behind the compressor in this case increases, and at a constant gas temperature in front of the turbine, the degree of heating of the working fluid in the cycle decreases, and therefore the efficiency of the gas turbine unit decreases.

The Brayton cycle with multistage compression, intercooling and regeneration

In a plant with a simple design, it is possible to increase the useful work by reducing the power consumption for the compressor drive. This can be achieved by adding multistage compression and intercooling into a gas turbine unit.

The advantages of using regeneration represent alternative ways of solving the problem of increasing the efficiency of a gas turbine unit. The combined using of multistage compression and intercooling in combination with heat regeneration from exhaust gases is one of the possibilities to increase power and efficiency of a gas turbine plant. Intercooling the air during compression reduces the compression work in the compressor and thus increases the useful cycle work. A decrease in the work on the compressor drive during intercooling is accompanied by the air temperature decrease at the inlet to the combustion chamber T_2, and this leads to an increase in fuel consumption at a constant permissible gas temperature in front of the turbine (T_3 = const) and at a constant air mass flow rate.

Parameters such as the pressure ratio, the effectiveness of regeneration, the initial temperature of the working fluid, and the temperature of the gas at the exit from the combustion chamber have a contradictory effect on fuel consumption. Therefore, to assess the effectiveness of the combined using of intercooling and heat regeneration requires multivariate analysis. In this work, we studied the influence of the effectiveness of regeneration, the initial temperature of the working fluid on the fuel consumption.

Minimization of the compression work in the compressor due to the uniform distribution of the pressure increase in the compression stages is considered in [8].

Comparison of two schemes of multistage compression with intermediate cooling (two-stage compression (Figure 4) and three-stage compression (Figure5)) And waste gas heat recovery is carried out. For further numerical studies, a GTU scheme with a two-stage compressor (with a uniform distribution of the degree of pressure increase between the stages) and intermediate air-cooling between the stages was chosen.

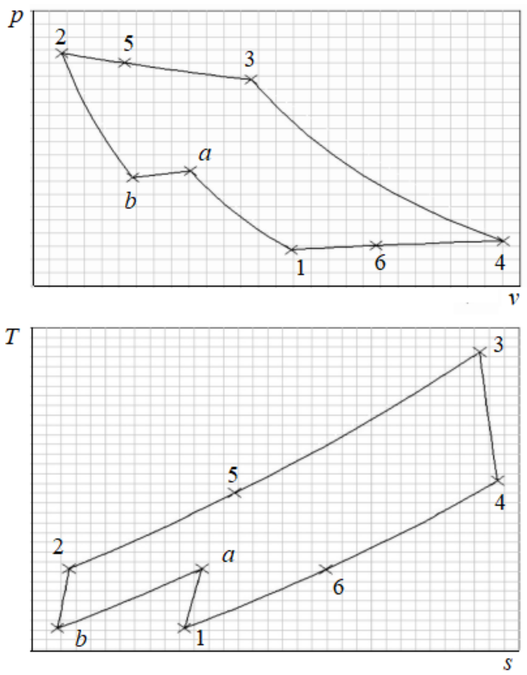

Figure 4. *p-v* and *T-S* diagram of the cycle with two-stage compression, intercooling and regeneration.

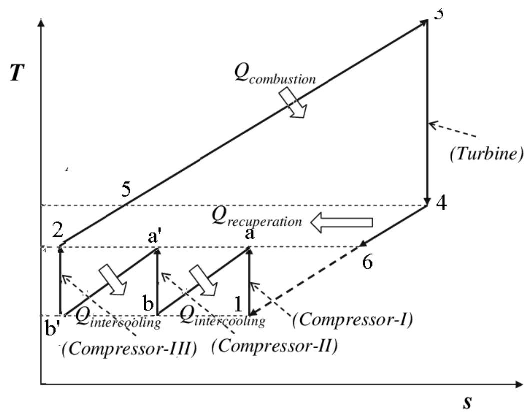

Figure 5. *p-v* and *T-S* diagram of the cycle with three-stage compression, intercooling and regeneration.

The real thermodynamic cycle of a gas turbine with irreversible compression in the compressor and expansion in the turbine is shown in figure 4.

The temperature of air at the end of each compression process:

$$T_2 = T_a = T_1\left[1+\frac{1}{\eta_c}\left(\pi_c^{\frac{k-1}{zk}} - 1\right)\right], \tag{14}$$

where z is number of compression stages.

The cycle thermal efficiency:

$$\eta_{th} = \frac{c_{pg} T_3 \eta_t \left(1 - \frac{1}{\pi_t^{\frac{k_g - 1}{k_g}}} \right) - c_p z T_1 \left(\frac{\pi_c^{\frac{k-1}{zk}} - 1}{\eta_c} \right)}{c_p \left[\left(T_3 - T_1 \left[1 + \frac{1}{\eta_c} \left(\pi_c^{\frac{k-1}{k}} - 1 \right) \right] \right) - \sigma_r \left(T_4 - T_1 \left[1 + \frac{1}{\eta_c} \left(\pi_c^{\frac{k-1}{zk}} - 1 \right) \right] \right) \right]} \qquad (15)$$

Gas turbine units with steam or water injection into the compressor

Evaporative cooling is used to lower the air temperature using the specific heat of vaporization, changing the liquid state of water to a gaseous state. In this process, the energy in the air does not change. Dry, warm air is replaced with cool and humid air. The positive effect of injection is appeared in the evaporative cooling of air during the compression process [9].

The technological scheme of injection of superheated water in a simple cycle gas turbine plant is shown in Figure 6. The figure shows that it is possible to inject water both into the inlet (1) of the compressor, and into the intermediate stages of the compressor. In this study, only water injection into the compressor inlet was considered.

In the study [10] it was revealed that the use of staged air compression and steam injection into the gas path of a gas turbine plant is expedient. It gives a gain in efficiency and useful power of the unit.

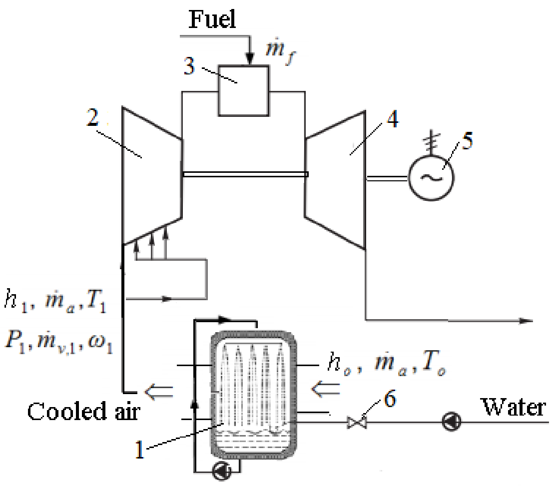

Figure 6. A simple open type gas turbine with a direct evaporative cooler

Demineralized water is provided into fogging nozzle by a pump. Pressurized liquid water is converted into a fog by means of special atomizing nozzles. The cooling effect is provided by water evaporation. With this strategy, the amount of injected water is strictly necessary for air saturation and water evaporation is completed before air enters into the compressor.

The temperature of air after fog cooling can be obtain from an energy balance on the dry air, water spray, and water vapor before and after the system.

$$T_1 = T_{0b} - \varepsilon (T_{0b} - T_{0w}),$$

where T_{0b} is dry-bulb temperature, T_{0w} is wet-bulb temperature, ε is evaporative cooling effectiveness.

Mass flow rate of the water is given by:

$$\dot{m}_w = (h_{v1} - h_{w0}) = \dot{m}_a (h_{a0} - h_{a1}) + \omega_0 \dot{m}_a (h_{v0} - h_{v1}), \qquad (16)$$

where $\dot{m}_w$ – the mass flow rate of water, $\dot{m}_a$ – the mass flow rate of dry air, h_{w2} – the enthalpy, $(h_{a0} - h_{a1})$– the enthalpy change of dry air, $(h_{v0} - h_{v1})$– the enthalpy change of water vapor containing in the air, ω_0 – the specific humidity of inlet air of water per kg of dry air.

$$\dot{m}_w = \dot{m}_a(\omega_1 - \omega_0), \tag{17}$$

where ω_0, ω_1 are the air specific humidity in the inlet and outlet of evaporator, respectively.

The cooling load of an evaporator $\dot{Q}_{cl}$:

$$\dot{Q}_{cl} = \dot{m}_a c_{pa}(T_0 - T_1). \tag{18}$$

$$\dot{W}_c = \dot{m}_a c_{pa}(T_2 - T_1) + \dot{m}_v(h_{g2} - h_{g1}), \tag{19}$$

where h_{g1} and h_{g2} are the enthalpies of saturated water vapor at the compressor inlet and exit; $\dot{m}_v = \dot{m}_a\omega_1$ is the mass of water vapor.

Heat supplied to the cycle was determine from the heat balance as it is proposed in classic textbook [12]:

$$\dot{Q}_1 = (\dot{m}_a + \dot{m}_f)c_{pg}T_3 - \dot{m}_a c_{pa}T_2 + \dot{m}_v(h_{v3} - h_{v2}), \tag{20}$$

h_{v2}, h_{v3} are the enthalpies of water vapor at the combustion chamber inlet and exit states, respectively. They can be calculated from the tables of thermophysical properties of water and steam[11].

The total gases mass flow rate at the turbine inlet:

$$\dot{m} = \dot{m}_a + \dot{m}_v + \dot{m}_f = \dot{m}_a(1 + \omega_1 + f), \tag{21}$$

where $\dot{m}_f$ is fuel mass flow rate; $f = \frac{\dot{m}_f}{\dot{m}_a}$ is fuel air ratio.

The net power output of a system with air cooling is

$$\dot{W} = \dot{W}_t - \dot{W}_c - \dot{W}_{pump}, \tag{22}$$

where $\dot{W}_c$ is compressor power, $\dot{W}_t$ is turbine power, $\dot{W}_{pump}$ is the pumping power to circulate the water inside the cooler. The power $\dot{W}_{pump}$ is small compared to the other terms and can be ignored. The thermal efficiency can be expressed as:

$$\eta_{th} = \frac{(1+\omega_1+f)c_{pg}T_3\eta_t\left(1 - \frac{1}{\pi_t^{\frac{k_g-1}{k_g}}}\right) - \left[c_p T_1\left(\frac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c}\right) + \omega_1(h_{g2} - h_{g1})\right]}{(1+f)c_{pg}T_3 - c_p T_1\left(\frac{\pi_c^{\frac{k-1}{k}} - 1}{\eta_c} + 1\right) + \omega_1(h_{v3} - h_{v2})}. \tag{23}$$

The thermal efficiency of gas turbine plant with inlet fogging and regeneration:

$$\eta_{th} = \frac{(1+\omega_1+f)c_{pg}T_3\eta_t\left(1-\dfrac{1}{\pi_t^{\frac{k_g-1}{k_g}}}\right)-\left[c_pT_1\left(\dfrac{\pi_c^{\frac{k-1}{k}}-1}{\eta_c}\right)+\omega_1\left(h_{g2}-h_{g1}\right)\right]}{(1+f)c_{pg}T_3-c_pT_1\left(\dfrac{\pi_c^{\frac{k-1}{k}}-1}{\eta_c}+1\right)+\omega_1\left(h_{v3}-h_{v2}\right)-q_r}, \tag{24}$$

where q_r – heat supplied to the working fluid in the regenerator:

$$q_r = \sigma_r c_p\left(T_4 - T_1\left[1+\frac{1}{\eta_c}\left(\pi_c^{\frac{k-1}{k}}-1\right)\right]\right)-\sigma_r\omega_1\left(h_{v5}-h_{v2}\right). \tag{25}$$

Comperison of multy-staged compression cycle with regeneration and bases cycle

Results of calculations performed here to estimate influents of multy-staged compression cycle with regeneration in thermal efficiency and power of a gas turbine plant are shown in Figure 7 and Figure 8.

Two schemes of gas turbine plant (multistage compression and regeneration) were considered, the first with two-stage compression (blue line) and the second with three-stage compression (red line).

It can be seen from Figure 7 that with an increase in the ambient temperature, the thermal efficiency decreases, and the power of the installation increases (see Figure 8). An increase in the number of compression stages leads to a greater increase in efficiency. The effect of an increase in ambient air temperature on plant efficiency decreases with an increase in the number of compression stages. For the three-stage compression, the increase in power with increasing ambient temperature is slightly greater compared to two-stage scheme of plant.

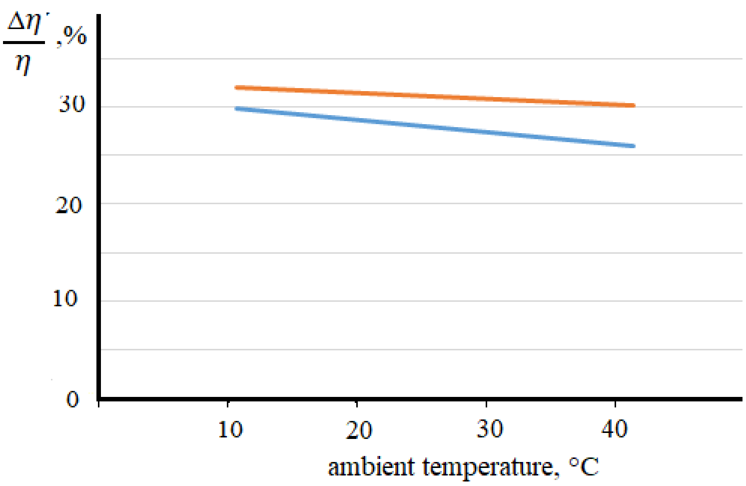

Figure 7. Influence of ambient temperature on the thermal efficiency.

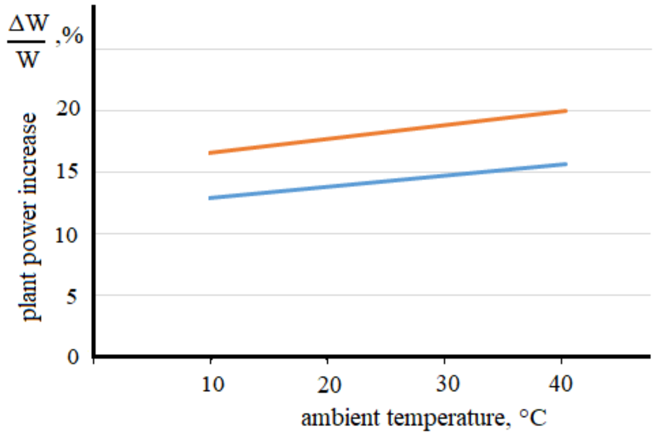

Figure 8. Influence of ambient temperature on the plant power.

In this work, a computational study of the influence on the thermal efficiency of the gas turbine plant cycle of such factors as preheating of liquid fuel, heat regeneration, multistage compression with intercooling and air cooling before the compression process by injection of water vapor was carried out. The results of this research for each of the factors were presented above in the form of graphs of the dependence of the cycle efficiency on the ambient air temperature.

A generalized graph of the dependence of the cycle efficiency on several factors is shown in Figure 7. It can be seen from the graph that the regeneration of the heat of the exhaust gases by heating the compressed air before the combustion chamber (line 2 in the graph of Figure 7) has a significant effect on the increase in the efficiency of the cycle.

Due to this method of increasing the efficiency of the cycle, the increase in the efficiency of the gas turbine plant can reach 9% at an air temperature at the engine inlet of 288 K. However, this increase in efficiency significantly decreases with an increase in the air temperature at the engine inlet and is 4-5% at a temperature of 325 K.

The positive effect of preheating liquid fuel before it is fed into the combustion chamber, which is 0.02% -0.025% at an air temperature at the engine inlet of 288 K, also decreases with an increase in this temperature.

The next most important factor having influence on the efficiency of the cycle is the injection of water vapor into the air stream in front of the compressor. By injecting water or steam the degree of pressure ratio increases slightly, which has a positive effect on the level of engine efficiency. In addition, this increases the mass flow rate of the working fluid. At the same time, the gas temperature in front of the turbine slightly decreases.

Conclusions

Based on the results of the calculations, the following conclusions can be made:

1. All four methods of increasing the fuel efficiency of gas turbine plants created on the basis of converted aircraft engines with low parameters of thermal cycle selected for the analysis, provide some increase in the internal efficiency of the thermal cycle, in which the gas temperature in front of the turbine is maintained at a constant level.

2. The most effective way to achieve this goal is to recover the heat from the exhaust gases leaving the engine and use this heat to heat up the compressed air in the compressor before it is fed into the combustion chamber.

3. A significant increase in the thermal efficiency of the cycle can be obtained by using a cycle with intermediate air cooling during its compression in the compressor.

4. Injection of water and steam into the air flow entering the compressor inlet does not lead to a significant increase in the efficiency of the cycle, although it does increase the useful work of the cycle.

5. Preheating the liquid fuel before it is fed into the combustion chamber using the heat of the gases leaving the engine slightly increases the efficiency of the cycle.

6. The combined use of these methods in pairs gives a noticeable increase the internal efficiency of the thermal cycle, although this significantly complicates the design of the gas turbine plant.

Prospects for further research

The degree of influence of each of the considered ways to improve fuel efficiency gas turbine plants based on converted aircraft engines, of course, depends on the level of the initial parameters of the working process implemented in them. In this work, all the calculations performed were carried out for a gas turbine based on an aircraft engine of relatively low power with low levels of pressure and temperature increase in the operating cycle. However, to date, aircraft engines with a higher level of initial parameters of the operating cycle are working out their flight life, when converted to ground-based gas turbines, other quantitative and even qualitative relationships between the effects of the above-discussed ways of increasing the fuel efficiency of gas turbines can appear. Further calculations are required to assess these effects.

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