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HIGHLY EFFICIENT COGENERATION POWER PLANT WITH DEEP REGENERATION BASED ON AIR BRAYTON CYCLE

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Today, an urgent scientific problem is the development of highly efficient, environmentally friendly, mobile, low-power cogeneration power plants that have small size and weight characteristics and use renewable resources as fuel. Potential consumers of generated energy are enterprises located in settlements that are far from combined heat and power plants (CHPP) or thermal power plants (TPP). Supplying heat and networks to such settlements from large power facilities is difficult, and transport charges for fuel delivery are very high. A concept of creating a highly efficient cogeneration power plant based on gas turbine technologies is proposed. A thermodynamic analysis of air, simple, and regenerative Brayton cycles is carried out. On the basis of its results, in a wide varying range of operating parameters, we determined the cycle implementation conditions providing high energy efficiency. A peculiarity of the proposed design solution is the use of air as a turbine working fluid to obtain useful capacity. In this case, the heat of the air leaving the turbine is used in the combustion process in a boiler. The proposed installation can be used with any heat source. Its main advantages compared to traditional gas turbine installations are as follows: energy advantages – the mounting of the combustion chamber of a solid fuel boiler downstream of the air turbine allows using the heat of the air leaving the air turbine, thereby reducing fuel consumption in the combustion chamber and, accordingly, increasing its efficiency; technological advantages – the turbine operates on pure air, and is protected from the formation of sludge on the surfaces of its blades or their erosion if the working fluid is dirty. It does not require that external turbine cooling systems be used, which greatly simplifies its design; environmental benefits – the turbine can operate on gas produced as a result of the thermal treatment of municipal solid waste. In addition, the boiler combustion chamber operates at almost atmospheric pressure with a lower emission of harmful substances into the atmosphere.

***Keywords**: direct Brayton cycle, regeneration, air turbine, cogeneration power plant.*

Introduction

With rising prices for fuel and electricity in this country, the problem of increasing the efficiency of power plants that provide the combined production of heat and electricity has become even more urgent. Today, a significant part of the electric and thermal energy is generated at large combined heat and power plants (CHPP) or thermal power plants (TPP). The efficiency of the operating power units is at the level of 35 to 37%.

In addition to relatively low energy efficiency, the operation of large CHPPs and TPPs is associated with large losses of energy during its transportation to the consumer, emissions of harmful substances into the environment as well as greenhouse gases. The economic aspect is also important. The use of mostly imported natural gas and coal at CHPPs and TPPs makes the cost of the generated electric and heat energy substantially dependent on the price of the primary fuel.

One of the progressive solutions to the existing energy problems is the introduction of the concept of distributed generation based on the construction of small thermal power plants (mini-CHPPs) with cogeneration power plants, which are cost-effective and environmentally friendly. Mini-CHPPs, which are capable of changing the power being generated in a wide range (from 0 to 100%) for short periods of time, are especially relevant because they can be used to regulate the balance in the United Energy System. In Europe, governments are encouraging the construction of energy centers based on mini-CHPPs and their connection to distribution networks in every way [1, 2].

Distributed Generation of Electric and Thermal Energy: Problems and Challenges

Distributed generation of electric and thermal energy is in great demand in this country today, especially for geographically separate enterprises, municipal facilities located in the areas where centralized energy supply systems are missing or destroyed, for agro-industrial enterprises, units of the armed forces, and state emergency services.

Despite the fact that today a fairly wide range of energy-generating equipment is presented on the market, there is a narrow range of choice of so-called small cogeneration power plants (with a capacity of 20 to 300 kW) operating on various types of fuel with high electric and total efficiency. The creation of such facilities is consistent both with global trends in the development of distributed energy and the provisions of the New Energy Strategy of Ukraine until 2035, which provides for a significant increase in the use of renewable fuel resources.

For cogeneration power plants operating in distributed generation systems, there can be used local fuel resources, for example, peat, oil shale, straw, waste from woodworking enterprises. The possibility of changing traditional hydrocarbon fuels for biogenic ones can reduce the payback period of such an installation and, to some extent, contribute to solving this country's energy independence problem. It is known that using cogeneration power plants in parallel with an external distribution network is most appropriate. In this case, it is possible to transfer surplus electricity to a distribution network, as well as receive it from the network in the case when the electricity generated does not cover its own needs (in the case of a stop for preventive work or during peak loads).

Now, many companies in the world produce small cogeneration power plants with capacities from several hundred kW and below [1–3]. Since the late 90s, there began the serial production of micro gas turbines (MGT) by such companies as Capstone Turbine Corporation (CPST), Toyota Turbine And Systems Inc., Calnetix Technologies, Bowman Power Group, Wilson Turbo Power Inc., Turbec R & DAB, Nissan Motors Co., UTC Power and others. Manufacturers offer their original technical solutions both for stationary and mobile cogeneration power plants. An analysis of commercially available MGTs showed that they differ in design, operational characteristics, power, types of fuel used. The greatest successes in terms of energy efficiency have been achieved for installations based on internal combustion engines [4]. The electrical efficiency of these installations can reach 40 to 45%. However, they require the use of expensive liquid or gaseous fuels. Among the disadvantages of such systems is their non-universality in the type of fuel, that is, impossibility of using different fuels simultaneously. For installations developed on the basis of gas turbine equipment, the electrical efficiency is somewhat lower (35 to 37%), but they also require the use of expensive natural gas. The small cogeneration power plants, that are now being created, operate on various types of biofuel, and in terms of their energy efficiency are still at a much lower level. For biogas plants based on gas turbine equipment, the efficiency is 21 to 27% [4].

In general, it can be stated that today there is an urgent need for small cogeneration power plants operating on various types of fuel, including renewable ones (wood, leaves, waste from wood and agricultural production, municipal solid waste, etc.), which have a fairly high electrical and total efficiency [5, 6].

Thus, an urgent scientific problem is the development of a concept of creating highly efficient, environmentally friendly, mobile, low-power cogeneration power plants that have small size and weight characteristics, and use various types of primary fuel, including renewable ones.

Thermodynamic Analysis of Ideal and Real Air Brayton cycle

At the first stage, to evaluate the possible efficiency of the power plant being designed, a thermodynamic analysis of the Brayton cycle was performed.

Simple Ideal Brayton Cycle

The simple ideal Brayton cycle in *T-s* coordinates is shown in Fig. 1. In adiabatic process 1–2, the working fluid is compressed from the parameters of point 1 to the parameters of point 2. In isobaric process 2–3, the working fluid is supplied with an amount of heat from the source q_1. In adiabatic process 3–4, the working fluid expands to the initial pressure $p_4=p_1$, and in isobaric process 4–1, it is reduced to the parameters of point 1 with the removal of heat q_2 to the heat receiver [7].

Accepting the assumption that the isobaric heat capacity of the working fluid c_p=const, and taking into account that $T_2/T_1=T_3/T_4=\pi_k^{(k-1)/k}$, the thermal efficiency of an ideal cycle can be represented as

$$\eta_{\text{thrm}} = 1 - 1/\pi_k^{(k-1)/k}, \tag{1}$$

where $\pi_k=p_2/p_1$ is the degree of air pressure increase in the compressor; k is the adiabatic exponent [7].

In an ideal cycle, the thermal efficiency depends only on the degree of pressure increase π_k, and does not depend on the turbine inlet air temperature T_3, and the greater the degree of pressure increase in the compressor, the greater the thermal efficiency of the cycle. It should be noted that even when the ideal cycle is analyzed, this condition is valid only in the first approximation. In the future, additional factors must be taken into account, which will be discussed below.

Let us analyze the effect of π_k on the thermal efficiency of the ideal cycle.

The compressor inlet pressure and air temperature are assumed to be constant (T_1=const; p_1=const), and so is the turbine inlet air temperature T_3 (T_3=const). Under such conditions, π_k can take values from 1 to $\pi_k^{\max}$. When $p_2=p_1$, $\pi_k=1$ (and at the maximum pressure $p_2^{\max}$ $\pi_k=\pi_k^{\max}$) then as a result of compression, the temperature reaches the maximum value $T_2=T_3$. The corresponding change in cycle configuration is shown in Fig. 2 [7].

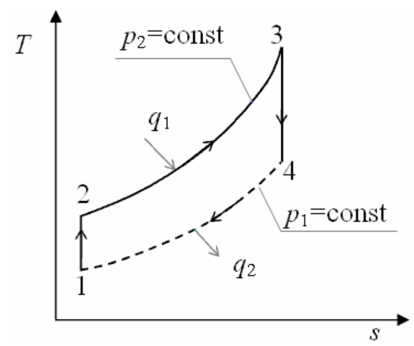


Fig. 1. Ideal Brayton cycle in the T-s diagram

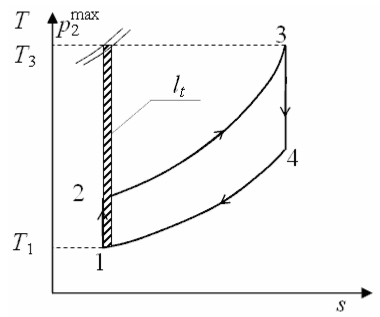


Fig. 2. Change in the cycle configuration at the maximum T_3 and p_2^{max}

At $\pi_k=1$, the cycle work $l_t = q_1 - q_2$ is zero, and therefore $\eta_{\text{thrm}}=0$. At $\pi_k=\pi_k^{\max}$, the efficiency η_{thrm} has its maximum value, since $T_2=T_3$. In this case, the thermal efficiency is determined from the expression $\eta_{\text{thrm}} = 1 - \dfrac{T_1}{T_3}$, and corresponds to the efficiency of the ideal Carnot cycle. As can be seen from Fig. 2, in the *T-s* diagram, at $\pi_k=\pi_k^{\max}$, the cycle changes its configuration, and degenerates into a straight vertical line. Thus, there exists a paradoxical situation when, at the maximum value of the thermal efficiency, the cycle operation is missing (l_t=0), i.e. all turbine operation is spent on the compressor drive [7].

On the other hand, at a fixed value of π_k, an increase in the turbine inlet air temperature T_3 leads to an increase in both the average heat supply temperature T_1 and average heat removal temperature T_2, their ratio remaining constant and equal to the T_2/T_1 ratio as a result of isobar equidistance [7]. In this case, the thermal efficiency of the cycle does not change, either, when the turbine inlet air temperature T_3 changes.

Figure 3 shows the dependencies of the thermal efficiency and operation of the ideal cycle on the degree of pressure increase.

As it can be seen, the choice of the pressure increase degree value of π_k should be made not only by the thermal efficiency of the cycle (Fig. 3, a), but also by the maximum cycle work value $l_t^{\max}$ (Fig. 3, b).

To find the conditions for reaching $l_t^{\max}$, we use the necessary condition for the extremum [7]

$$\left(\frac{\partial l_t}{\partial \pi_k}\right) = 0,$$

as a result of which we obtain the optimal compression ratio equal to

$$\pi_k^{\text{opt}} = \beta^{\frac{k}{2(k-1)}}, \tag{2}$$

where β is the ratio of temperatures T_3/T_1.

Thus, the efficiency of the ideal cycle is influenced both by the compressor inlet temperature T_1 and the turbine inlet working fluid temperature T_3. Fig. 4 shows the combined influence on both the thermal efficiency of the cycle and turbine operation of both the degree of pressure increase in the cycle and temperature T_3, where π_k^{opt1}, $\pi_k^{\max 1}$ are the optimal and maximum degrees of pressure increase at β_1=3.5, and π_k^{opt2}, $\pi_k^{\max 2}$, at β_2=3.8. It can be seen that with an increase in the temperature ratio $\beta=T_3/T_1$ from β_1=3.5 to β_2=3.8, it increases from $\pi_k^{\max 1}$=80 to $\pi_k^{\max 2}$=105. It can also be seen that the optimal compression ratio π_k^{opt} depends on the value of β, which corresponds to dependence (2).

As an additional indicator of the efficiency of the gas turbine cycle, the work coefficient is also used, which is the ratio [7]

$$\varphi = 1 - l_{\text{compr}} / l_{\text{turb}}$$

where l_{compr}, l_{turb} are the technical works of reversible adiabatic processes of air compression in the compressor and air expansion in the turbine.

Given that for an ideal cycle $l_{\text{compr}}=c_p(T_2-T_1)$ and $l_{\text{turb}}=c_p(T_3-T_4)$, and $T_2/T_1=T_3/T_4=\pi_k^{(k-1)/k}$, $T_3/T_1=\beta$, the exponent φ can be expressed as a function of the degree of pressure increase and temperature ratio β as

$$\varphi = 1 - \frac{\pi_k^{\frac{k-1}{k}}}{\beta}. \tag{3}$$

Fig. 5 shows the dependence of φ both on β and π_k.

By substituting (2) into (1) and (3) we can obtain the dependence for finding the optimal thermal efficiency of the ideal cycle, as well as the work coefficient φ

$$\eta_{\text{thrm}}^{\text{opt}} = 1 - \frac{1}{(\pi_k^{\text{opt}})^{\frac{k-1}{k}}} = 1 - \frac{1}{\sqrt{\beta}}, \quad \varphi^{\text{opt}} = 1 - \frac{1}{\sqrt{\beta}}.$$

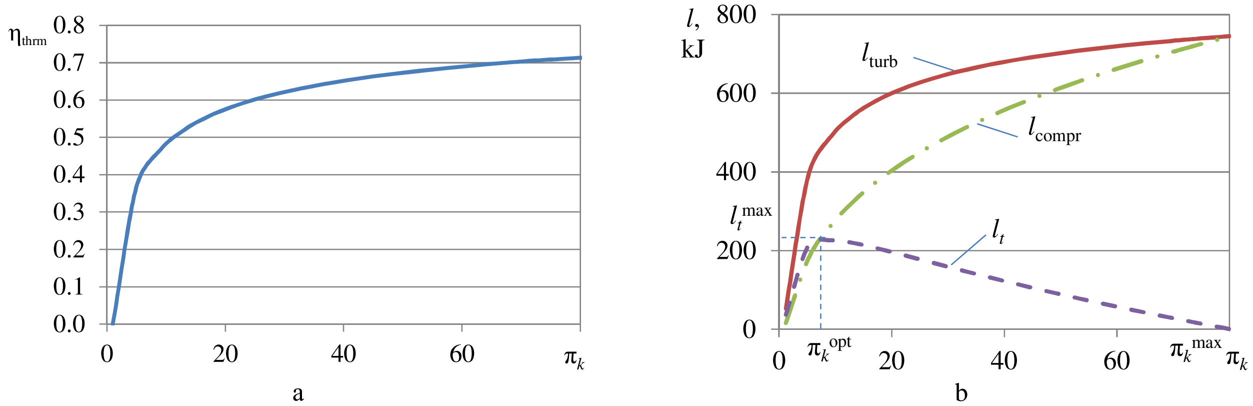


Fig. 3. Dependencies on the degree of pressure increase:
a – of the thermal efficiency of the ideal cycle; b – the compression work l_{compr}, the expansion work l_{turb}, and the cycle work l_t

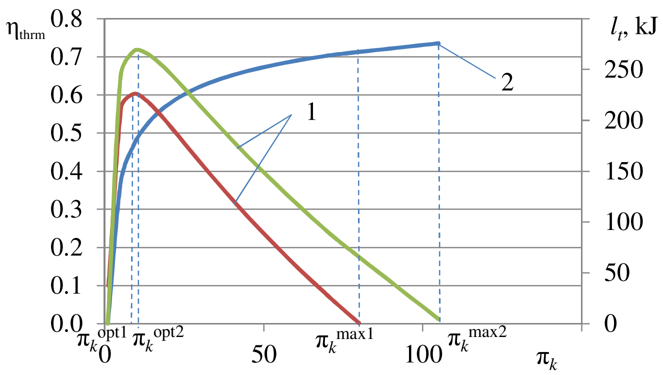


Fig. 4. Combined influence on both the thermal efficiency of the cycle and turbine operation of both the degree of pressure increase in the cycle and the temperature T_3:
1 – cycle work; 2 – cycle thermal efficiency

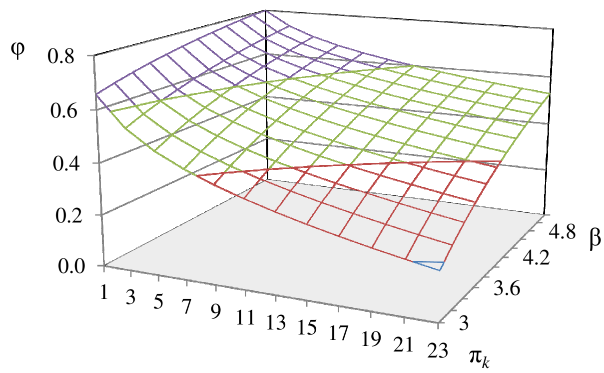


Fig. 5. Combined influence of β and π_k on the work coefficient φ

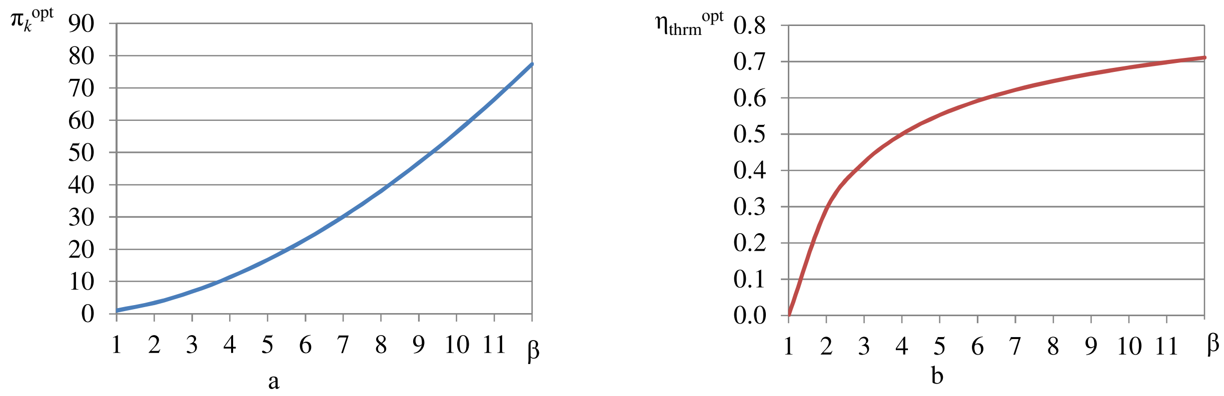


Fig. 6. Dependence of the optimal values of π_k^{opt} (a) and η_{thrm}^{opt} (b) on β

The analysis of the equations allows us to conclude that the higher T_3 and lower T_1, the higher the efficiency of the ideal Brayton cycle. The optimal, in terms of maximum specific work, values of π_k^{opt}, $\eta_{\text{thrm}}^{\text{opt}}$, and φ^{opt} depend only on the ratio $\beta=T_3/T_1$, $\eta_{\text{thrm}}^{\text{opt}}$ and φ^{opt} being equal for an ideal cycle. Fig. 6 shows the dependencies of the optimal compression ratio and optimal thermal efficiency of the ideal cycle on β.

The analysis of the ideal cycle at optimal $\pi_k^{\text{opt}} = f(\beta)$ showed that the condition for the maximum cycle work is satisfied if the compressor outlet temperature T_2 is equal to the turbine outlet temperature T_4.

Ideal Brayton Cycle with Heat Regeneration

One of the ways of increasing the thermodynamic efficiency of the Brayton cycle, that is, bringing it closer to the generalized Carnot cycle, is to use regeneration [7, 8]. Fig. 7 shows an ideal cycle with the regeneration of the working fluid heat in the *T*-*s* diagram.

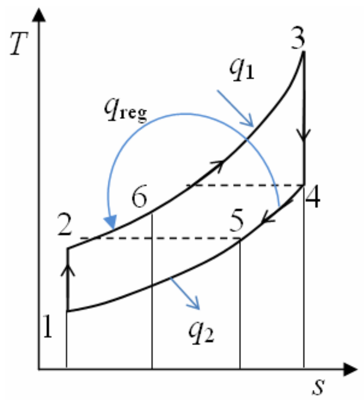


Fig. 7. Ideal Brayton regenerative cycle

Fig. 7 presents the following processes: 1–2 – adiabatic air compression in the compressor; 2–6 – isobaric air heating in the heat exchanger-regenerator; 6–3 – isobaric supply of heat; 3–4 – adiabatic expansion on turbine blades; 4–5 – isobaric heat removal from gases to the air in the heat exchanger-regenerator; 5–1 – isobaric cooling of the working fluid. With complete regeneration, the heat from section 4–5 is transferred reversibly to section 2–6. As a result, the heat q_1 from an external source is supplied to the working fluid at a higher average temperature, and the heat q_2 is removed to an external source at a lower average temperature.

To characterize the regenerative heat transfer process, we used the degree of regeneration σ

$$\sigma = \frac{q_{\text{reg}}}{q_{\text{reg}}^{\max}} = \frac{T_4 - T_5}{T_4 - T_2} = \frac{T_6 - T_2}{T_4 - T_2},$$

where q_{reg} is the actually regenerated heat; $q_{\text{reg}}^{\max}$ is the maximum possible regenerated heat.

The complete regeneration (σ=1) occurs when $T_6=T_4$, and $T_5=T_2$. In practice, the condition $T_6<T_4$ and $T_2<T_5$ is fulfilled.

Given that $T_2/T_1=T_3/T_4=\pi_k^{(k-1)/k}$ and $T_3/T_1=\beta$, the thermal efficiency of the regenerative cycle can be expressed as

$$\eta_{\text{thrm}}^{\text{RC}} = \frac{T_3 - T_4 - T_2 + T_1}{T_3 - T_2 - \sigma \cdot q_{\text{reg}}^{\max}} = \frac{\beta - \pi_k^{\frac{k-1}{k}} - \pi_k^{\frac{1-k}{k}}\beta + 1}{\beta - \pi_k^{\frac{k-1}{k}} - \sigma\left(\pi_k^{\frac{1-k}{k}}\beta - \pi_k^{\frac{k-1}{k}}\right)}.$$

and with the complete regeneration (σ=1), as

$$\eta_{\text{thrm}}^{\text{RC}}=1-\frac{\pi_k^{\frac{k-1}{k}}-1}{\beta\left(1-\pi_k^{\frac{1-k}{k}}\right)}.$$

Fig. 8 shows the change in the thermal efficiency of the regenerative cycle, with the change depending on π_k, σ, β.

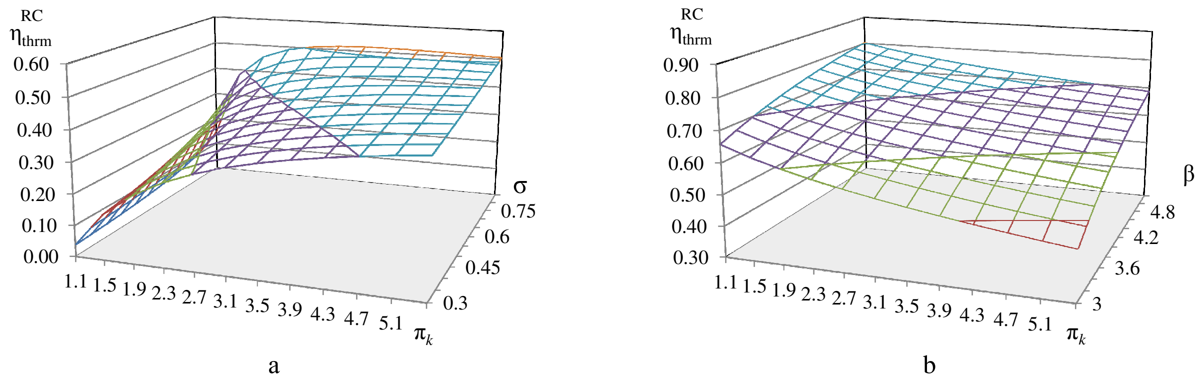


Fig. 8. Dependencies of the thermal efficiency of the regenerative cycle on:
a – π_k and σ at β=3.5; b – π_k and β at σ=1

Fig. 9 shows the change in the thermal efficiency of the regenerative cycle with the complete heat recovery in the cycle, and Fig. 10, at various degrees of regeneration for variable temperature ratios β.

The highest efficiency is observed at β=4. So, if T_1=25 °C and β=4, then T_3=920 °C, and at T_1=15 °C, T_3=880 °C. Fig. 9 shows that if the regeneration is complete (σ=1), which is impossible in a real cycle, then the maximum possible thermal efficiency of the cycle will be, for example, at π_k=2 and β=4, $\eta_{\text{thrm}}^{\text{RC}}$=0.69 (Fig. 9), and for the same values of π_k and β, but at σ=0.75, $\eta_{\text{thrm}}^{\text{RC}}$ will be equal to 0.43 (Fig. 10, a), and at σ=0.93, $\eta_{\text{thrm}}^{\text{RC}}$ will be equal to 0.5 (Fig. 10, b). If we go over to absolute values, then the temperatures corresponding to the regime with σ=0.93, π_k=2 and β=4 can be T_1=25 °C; T_2=90.3 °C; T_3=919.5 °C; T_4=705 °C; T_5=130 °C; T_6=665 °C.

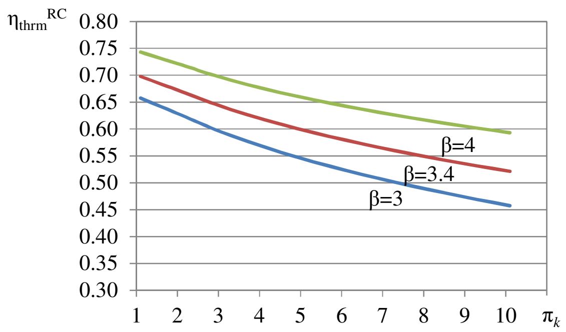


Fig. 9. Dependence of η_{thrm}^{RC} on π_k for various values of β with the complete regeneration (σ=1)

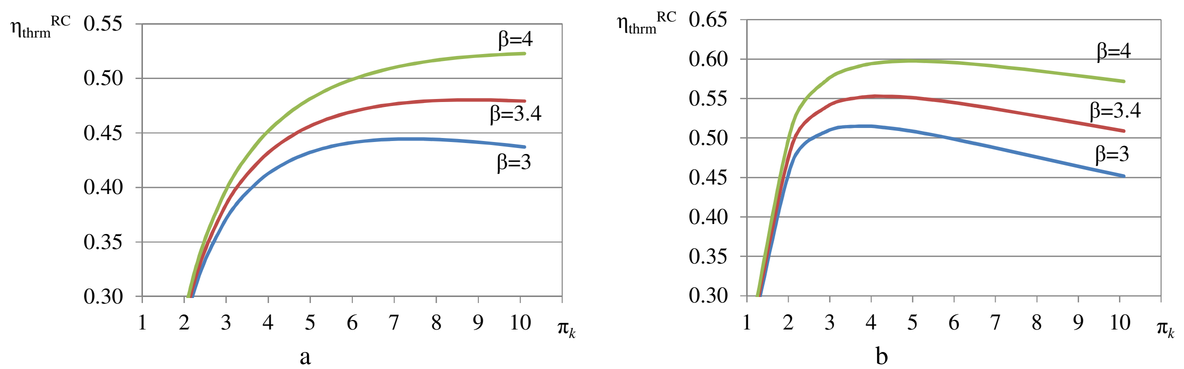


Fig. 10. Dependence of η_{thrm}^{RC} on π_k for various values of β:
a – at the regeneration degree σ=0.75; b – at the regeneration degree σ=0.93

Thus, the analysis of the ideal Brayton cycle both without and with heat regeneration allowed us to draw the following conclusions:

– the optimal degree of pressure increase in the compressor, which corresponds to the maximum cycle work for a fixed value of the temperature ratio $T_3/T_1=\beta$, corresponds to the cycle parameters $T_2=T_4$. Accordingly, this ideal cycle regime without regeneration will correspond to such a regime with regeneration when $\eta_{\text{thrm}} = \eta_{\text{thrm}}^{\text{RC}}$, and $q'_{\text{reg}} = 0$. Thus, π_k^{opt} of an ideal cycle without regeneration is a limitation for a cycle with regeneration;

– to increase the thermal efficiency of the cycle with heat regeneration, $\eta_{\text{thrm}}^{\text{RC}}$, it is necessary to increase $T_3/T_1=\beta$ and, as much as possible, the degree of regeneration.

The Scheme of Implementation of the Air Brayton Cycle with the Original Method of Regeneration

In the power installation proposed, the working fluid is pure air, and its peculiarity is that the air leaving the turbine is supplied to the boiler as an oxidizing agent for the combustion process. Such an approach provides the utilization of the heat leaving the turbine air. The schematic diagram of the power installation is shown in Fig. 11.

The power installation contains an air compressor with a drive, an air turbine with an electric generator, a combustion chamber of a boiler operating on various types of fuel, a fuel source, a heat exchanger with an air and a gas circuit, as well as an air pipeline. The compressor and turbine can be made either as single-stage or multi-stage devices of various types (axial, radial, radial-axial, etc.). The electric generator can be located either on a common shaft with a compressor and a turbine, or on a separate shaft with a power turbine.

The installation operates as follows.

Atmospheric air is absorbed by the compressor, where its temperature and pressure increase as a result of the compression process. Further, the air is supplied for heating to the regenerative heat exchanger HE-1 included in the boiler. After passing the air circuit of the heat exchanger, the heated air enters the air turbine, where it expands to near atmospheric pressure, with the air being accompanied by a decrease in its temperature and generation of the mechanical energy that is transmitted to the power consumer. Part of the useful power produced by the turbine goes to the compressor drive, and the other part is used to drive the electric generator. From the air turbine, the air is supplied to the combustion chamber of the boiler, where the fuel source supplies fuel for combustion. At the stages of ignition of the boiler, with the help of a blower, atmospheric air is supplied to the boiler. In the boiler combustion chamber, the mixture of fuel and air burns with releasing heat, and the exhaust gas enters the gas circuit of the heat exchanger HE-1, on passing which it is released into the atmosphere. Also, in the heat exchanger HE-2 of the solid fuel boiler, the water circulating from a heat point is heated.

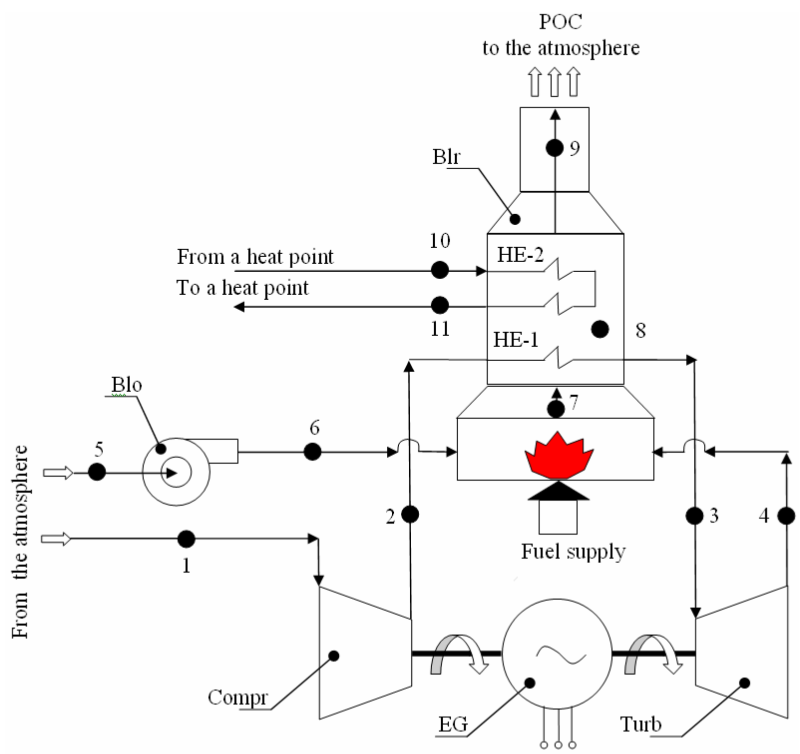


Fig. 11. Power installation diagram:
Blr – boiler with a heat exchanger; Compr – compressor; Turb – turbine;
EG – electric generator; Blo – blower;
HE-1 – recuperative heat exchanger for air heating;
HE-2 – heat exchanger for water heating

Power installation advantages:

– energy – the mounting of the combustion chamber of a solid fuel boiler downstream of the air turbine allows using the heat of the air leaving the air turbine, thereby reducing fuel consumption in the combustion chamber and, accordingly, increasing its efficiency;

– technological – the turbine operates on pure air, and is protected from the formation of sludge on the surfaces of its blades or their erosion if the working fluid is dirty. It does not require that external turbine cooling systems be used, which greatly simplifies its design;

– environmental – waste management. The turbine can operate on gas produced as a result of the thermal treatment of municipal solid waste. In addition, the boiler combustion chamber operates at almost atmospheric pressure with a lower emission of harmful substances into the atmosphere;

– economic – it can generate its own electricity and heat.

In Fig. 11 are accepted the following designations: 1 – compressor inlet; 2 – compressor outlet; 3 – turbine inlet; 4 – turbine outlet; 5 – blower inlet; 6 – blower outlet; 7 – boiler combustion chamber; 8 – in the boiler downstream of HE-1; 9 – boiler outlet downstream of HE-2; 10 – water from a heat point; 11 – water to a heat point.

Calculation of Real Power-Plant Cycle Characteristics

To choose the parameters of the air Brayton cycle with the original method of regeneration, we will further take into account the real cycle irreversibility caused by the presence of friction in adiabatic compression processes 1–2´ and expansion processes 3–4´ of the air both in the compressor and air turbine (Fig. 12).

The irreversibility of the adiabatic process of air compression in the compressor characterizes the adiabatic efficiency of the compressor

$$\eta_{\text{compr}} = \frac{l_{\text{compr}}}{l'_{\text{compr}}} = \frac{c_{p\,\text{air}}(T_2 - T_1)}{c_{p\,\text{air}}(T'_2 - T_1)},$$

and the irreversibility of the adiabatic process of air expansion in the turbine is the internal relative efficiency of the turbine

$$\eta_{\text{turb}} = \frac{l'_{\text{turb}}}{l_{\text{turb}}} = \frac{c_{p\,\text{air}}(T_3 - T'_4)}{c_{p\,\text{air}}(T_3 - T_4)},$$

where $l'_{\text{compr}}, l'_{\text{turb}}$ are the works of both compression in the compressor and expansion in the turbine in a real cycle.

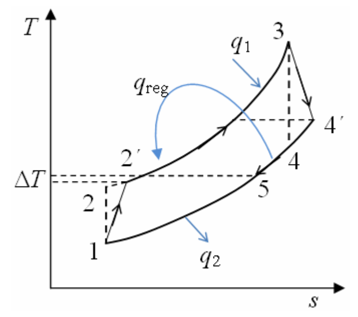


Fig. 12. Real Brayton cycle with heat regeneration

As it is for an ideal cycle, the assumption is made that the isobaric heat capacity of air $c_{p\,\text{air}}$ is variably weak. This allows us to represent the internal absolute efficiency of the actual regenerative cycle through the relative quantities π_k, β and σ' as

$$\eta_i^{\text{RC}} = \frac{l'_{\text{turb}} - l'_{\text{compr}}}{q_1 - \sigma' q'^{\max}_{\text{reg}}} = \frac{(T_3 - T'_4) - (T'_2 - T_1)}{(T_3 - T'_2) - \sigma'(T'_4 - T'_2)} =$$

$$= \frac{\beta\left(1 - \pi_k^{\frac{1-k}{k}}\right)\eta_{\text{т}} - \left(\pi_k^{\frac{k-1}{k}} - 1\right)\dfrac{1}{\eta_{\text{compr}}}}{\beta - \left(1 + \dfrac{1}{\eta_{\text{compr}}}\left(\pi_k^{\frac{k-1}{k}} - 1\right)\right) - \sigma'\left(\beta - \left(1 + \dfrac{1}{\eta_{\text{compr}}}\left(\pi_k^{\frac{k-1}{k}} - 1\right)\right) - \beta\left(1 - \pi_k^{\frac{1-k}{k}}\right)\eta_{\text{turb}}\right)},$$

where the actual cycle regeneration degree is

$$\sigma' = \frac{q'_{\text{reg}}}{q'^{\max}_{\text{reg}}} = \frac{T'_4 - T_5}{T'_4 - T'_2}.$$

It should be noted that for the proposed scheme (Fig. 11), the temperature difference ΔT (Fig. 12) will occur only at the outlet of the heat exchanger HE-1 from the side of the exit gases T_5, because at the inlet, the heat exchanger HE-1 receives a mixture of combustion products with the air temperature T'_4, such a mixture having a temperature that is higher than the turbine inlet temperature T_3.

We studied the influence of the degree of regeneration (Fig. 13) and the degree of increase in pressure at β=3.5 (Fig. 14) on the efficiency of the actual regenerative cycle. Fig. 14 shows the results of the calculations carried out at different values of the adiabatic efficiency of both the compressor and turbine.

Figs. 13 and 14 show that the efficiency values at σ´=0 are equal to the efficiency value of the actual cycle without regeneration. With regeneration at σ´=0.99, the temperature difference $\Delta T = (T_5 - T_2')$ is 5 °C.

Fig. 15 shows the dependencies of the efficiency of the actual Brayton cycle both without and with regeneration (σ´=0.9) at given η_{compr}=0.88 and η_{turb}=0.9.

Fig. 15 shows that the efficiency of both a simple actual cycle and regenerative one significantly depend both on the degree of air pressure increase in the compressor and the ratio of the turbine and compressor inlet air temperatures. The higher β, the higher the efficiency in both cycles. However, the nature of the dependence of the efficiency on π_k for simple and regenerative real cycles varies significantly. So, for a regenerative cycle, the maximum values of η_i^{RC} are observed at π_k<4, while for a simple one, the efficiency increases at large values of the air pressure increase degree (π_k>4). In addition, in a regenerative cycle, the presence of maximum values of η_i^{RC} is observed in the range 2>π_k>2.6 with a change in β from 3 to 4.8.

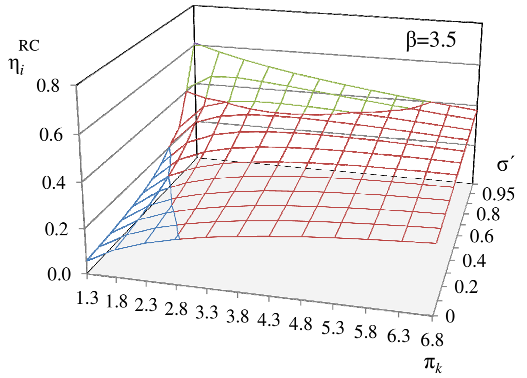


Fig. 13. Dependence of the efficiency of the actual regenerative cycle with varying π_k and σ´
at β=3.5, η_{compr}=0.88 and η_{turb}=0.9

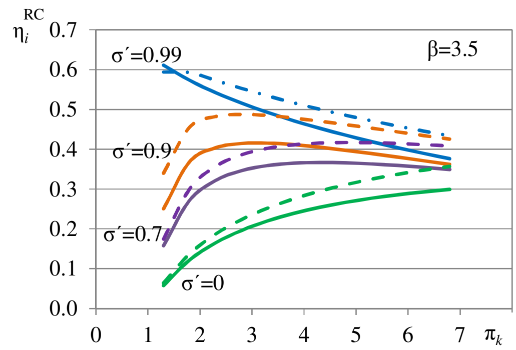
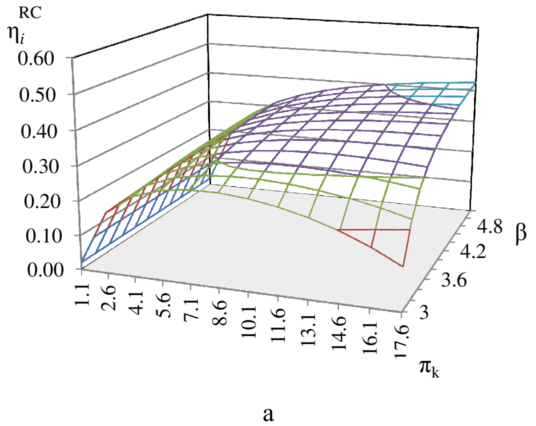


Fig. 14. Dependence of the actual regenerative cycle efficiency with varying π_k and σ´
at β=3.5 and at η_{compr}=0.88 and η_{turb}=0.9 (solid line);
and at η_{compr}=0.92 and η_{turb}=0.95 (dash-dot line)



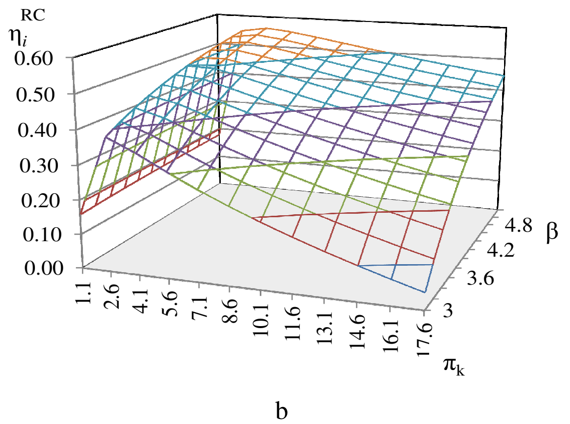


Fig. 15. Dependence of the efficiency with varying π_k and β:
a – for the simple actual cycle; b – for the regenerative cycle

Next, we consider the effect of various cycle parameters on the flow characteristics of the installation. The required air flow rate for a turbine with a power W_{el}=100 kW can be found as [5]

$$m_{\rm air} = \frac{W_{\rm el}}{\left[c_{p_{\rm air(3\text{-}4')}}\left(T_3 - T_4'\right)\cdot\eta_{\rm MT} - c_{p_{\rm air(2'\text{-}1)}}\left(T_2' - T_1\right)\cdot\dfrac{1}{\eta_{\rm MC}\cdot\eta_{\rm CD}}\right]\eta_{\rm EG}},$$

where $c_{p_{\rm air}}$ is the average isobaric heat capacity of air in the corresponding temperature range; $\eta_{\rm MT}$, $\eta_{\rm MC}$ are the mechanical turbine and compressor efficiencies taken equal to 0.95; $\eta_{\rm CD}$, $\eta_{\rm EG}$ are the compressor drive and electric generator efficiencies taken equal to 0.97; T'_2, T'_4 are the actual compressor and turbine outlet air temperatures, which are defined as

$$T_2' = T_1\left\{1 + \frac{1}{\eta_{\rm compr}}\left[\left(\pi_k\right)^{\frac{k-1}{k}} - 1\right]\right\}, \qquad T_4' = T_3\left\{1 - \eta_{\rm turb}\left[1 - \left(\frac{p_3}{p_4}\right)^{\frac{1-k}{k}}\right]\right\}.$$

The heat exchanger outlet air pressure for heating the air is determined as follows:

$$p_3 = p_2\left(1 - \Delta p_{\rm HE}^{\rm air}\right),$$

where $\Delta p_{\rm HE}^{\rm air}$ is the generalized coefficient of pressure loss in HE-1, which is taken to be 0.05.

The pressure at the turbine outlet is $p_4 = p_1$.

Fig. 16 shows the change in the required air flow to a 100 kW turbine with varying T_3, π_k, $\eta_{\rm compr}$ and $\eta_{\rm turb}$.

The total installation efficiency is determined by the following function:

$$\eta_{\rm instl}^{\rm RC} = \frac{\left[c_{p_{\rm air(3\text{-}4')}}\left(T_3 - T_4'\right)\cdot\eta_{\rm MT} - c_{p_{\rm air(2'\text{-}1)}}\left(T_2' - T_1\right)\cdot\dfrac{1}{\eta_{\rm MC}\cdot\eta_{\rm CD}}\right]\eta_{\rm EG}}{c_{p_{\rm air(3\text{-}2')}}\left(T_3 - T_2'\right) - c_{p_{\rm air(2'\text{-}4')}}\left(T_2' - T_4'\right)}.$$

Fig. 17 shows the influence on the total efficiency of a power installation with the regeneration of the parameters π_k, $\eta_{\rm compr}$ and $\eta_{\rm turb}$, where the solid line indicates the dependence obtained at T_3=800 °C; and the dot-and-dash one, at T_3=700 °C.

In the calculations, $\Delta T = \left(T_5 - T_2'\right)$ was taken equal to 15 °C, which corresponds to the change in the regeneration coefficient in the range from 0.95 to 0.97.

In addition to the estimates presented in this paper, it is necessary to carry out a large amount of research related to the assessment of the design as well as technical and economic characteristics of the installation. After setting out the fundamental aspects of designing a power plant operating on the basis of the air Brayton cycle, which are generally applicable to any heat source, there should be assessed the possibility of using different types of fuel.

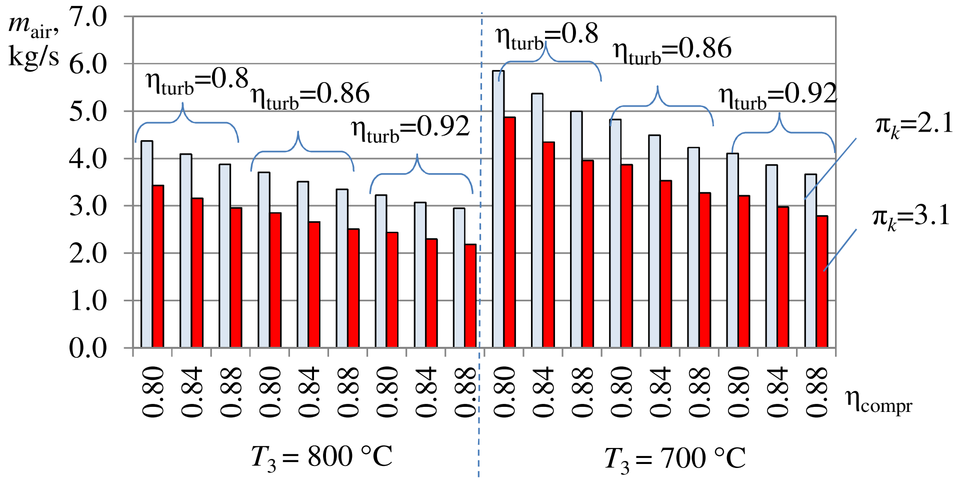


Fig. 16. Change in m_{air} for a 100 kW turbine with varying T_3, π_k, η_{compr} and η_{turb}

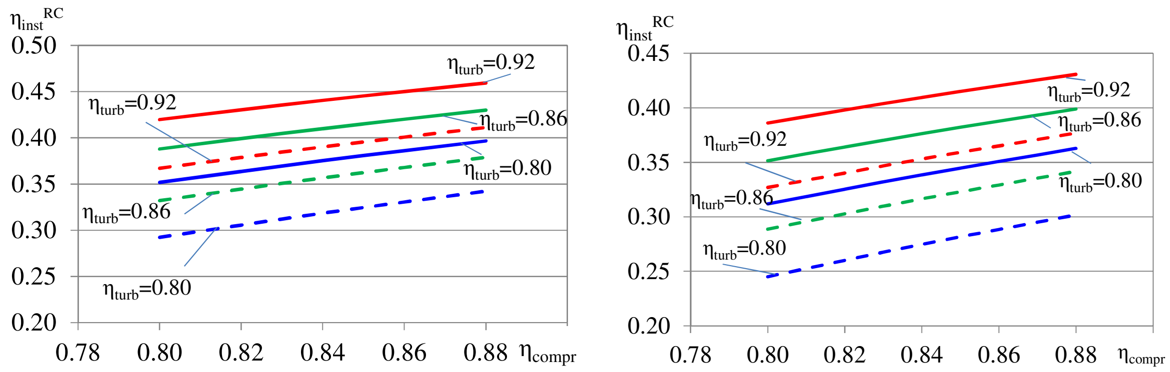


Fig. 17. Change in η_{instl}^{RC} for a 100 kW turbine with varying η_{compr} and η_{turb}:
a – at π_k=2.1; b – at π_k=3.1

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Conclusions

A concept of creating a highly efficient mobile power plant with deep regeneration based on air Brayton's cycle is proposed. The original method of regeneration allows us to increase the coefficient of heat regeneration in the cycle up to 0.95 to 0.97, which makes it possible to increase the power plant efficiency up to 45%.

Based on the results of the thermodynamic analysis of air, simple and regenerative Brayton cycles in a wide range of different varying parameters, we determined the cycle implementation conditions that provide high energy efficiency.

It was established that in the compressor, there is an optimal compression ratio at which the maximum cycle efficiency is achieved. Its value depends on the ratio of the turbine inlet temperature to the compressor inlet temperature. Thus, if the ratio of these temperatures is 3.5, then the maximum 50% efficiency of the actual cycle is observed with a compression ratio of 2.1. It should be noted that under the same conditions, taking into account real losses in the cycle, the efficiency of the installation will decrease by 5% compared to the ideal version without losses.

The analysis of the influence of the adiabatic efficiency of both the compressor and turbine on the efficiency of the installation showed that when the former vary in the range from 80 to 92%, the values of the latter vary from 25 to 45%. Thus, the results obtained make it possible to establish rational operating parameters of the installation at the initial design phase.

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Високоефективна когенераційна енергоустановка з глибокою рекуперацією на основі повітряного циклу Брайтона

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На сьогодні актуальною науковою проблемою є розробка високоефективних, екологічно чистих маневрених когенераційних енергетичних установок малої потужності з невеликими масогабаритними характеристиками, що використовують як паливо поновлювані ресурси. Потенційними споживачами енергії, що виробляється, є підприємства у віддалених від теплоелектроцентралей (ТЕЦ) і теплових електростанцій (ТЕС) населених пунктах, куди підвести тепло і електромережі від великих енергооб'єктів важко, а транспортні витрати з доставки палива дуже великі. Запропоновано концепцію створення високоефективної когенераційної енергоустановки на базі газотурбінних технологій. Проведено термодинамічний аналіз повітряного, простого і такого, що використовує регенерацію, циклів Брайтона, за результатами якого в широкому діапазоні варіювання режимних параметрів визначені умови реалізації циклу, які забезпечують високу енергетичну ефективність. Особливість запропонованого схемного розв’язку полягає у використанні повітря як робочого тіла в турбіні для отримання корисної потужності. За таких умов теплота повітря, що виходить з турбіни, використовується в процесі горіння в котлі. Установка, що пропонується, може використовуватися з будь-яким джерелом тепла. Її основні переваги порівняно з традиційними газотурбінними установками такі: енергетичні переваги – установка камери згоряння твердопаливного котла за повітряною турбіною дозволяє використовувати тепло повітря, що виходить з повітряної турбіни, і тим самим зменшити витрату палива в камері згоряння та відповідно збільшити коефіцієнт корисної дії; технологічні переваги – турбіна працює на чистому повітрі і захищена від утворення осадів на поверхні лопаток або їх ерозії при використанні «брудного» робочого тіла, не потребує застосування зовнішніх систем охолодження турбіни, що значно спрощує її конструкцію; екологічні переваги – можливість роботи установки на газі, що отримується в результаті термічної обробки твердих побутових відходів. Крім того, камера згоряння котла працює практично за атмосферного тиску з меншим викидом шкідливих речовин в атмосферу.

***Ключові слова**: прямий цикл Брайтона, регенерація, повітряна турбіна, когенераційна енергоустановка.*

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