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ANALYSIS OF THE EFFICIENCY OF A POWER GENERATING PLANT OPERATING ON THE BASIS OF THE BRAYTON THERMODYNAMIC CYCLE AND ENERGY RECUPERATION

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The thermal scheme of a power generating plant with a remote heat exchanger operating according to the Brayton cycle with energy recuperation is considered. It is assumed that the plant will work on non-certified (cheap) biofuel. It is shown that, in contrast to the usual Brayton cycle, in the cycle with energy recuperation, the greatest influence on the thermal efficiency is the heating temperature of the working medium and the internal efficiency of the main components of the plant, such as the compressor and the turbine. Also, in contrast to the usual Brayton cycle, a higher efficiency of the plant is achieved with smaller degrees of pressure reduction (increase) in the turbine (compressor). It was established that even at a relatively low temperature of the working medium heating (500 ºC), with high efficiency of the compressor and turbine, it is possible to achieve good characteristics of the power plant as a whole. At a temperature of up to 850 ºC, a thermal efficiency of 40% is achieved, but in this case the cost of materials and production increases. For a final conclusion about the possibility of using the proposed plant and its efficiency, it is necessary to conduct additional studies, in particular, of its main elements, such as a compressor, turbine, heat exchanger and others.

Keywords: *thermal scheme, power generating plant, Brayton cycle, energy recuperation, thermal efficiency, turbine, compressor, efficiency.*

Introduction

Today, the global trend of the so-called "green" transition and decarbonization of the energy industry is extremely relevant [1–3]. Green energy primarily refers to wind and solar generation, the potential of which, particularly for Ukraine, is significant [3, 4]. The main disadvantage of wind and solar energy is considered to be their instability associated with changes in the weather, seasons and even daily changes. To ensure their functioning and, in particular, the unified energy system as a whole, it is necessary to have additional maneuvering and reserve capacities or highly efficient energy storage systems [5, 6]. Hydropower, of course, is also "green", but despite all that, it is more traditional and largely developed, so the potential for its expansion, including at the expense of small rivers, is significantly limited [6, 7]. According to modern classifications, nuclear energy is also classified as "green", which, taking into account the development of technologies of small modular reactors and the orientation towards the transition to thermonuclear fusion in the future, is the most effective and promising [8, 9].

Another source of "green" energy is biofuels [10, 11]. They are usually relatively low-quality and low-calorie (uncertified) fuels. Their use for the production of electrical energy with the help of piston engines or gas turbines requires the introduction of expensive processing technologies for higher quality (certified) fuels (biomethane, liquid fuels, pyrolysis gasification, and others) [12, 13]. To reduce the cost of using uncertified biofuels, the creation of highly efficient heat generators has gained intensive development [14, 15]. Most of them are designed to generate heat for heating or other technological processes, for example, drying grain crops in elevators [14]. Research is also being conducted on the development of technologies for converting this thermal energy into electrical energy [16]. The most suitable for these purposes are steam turbine plants operating on the thermodynamic Rankine cycle [16, 17]. Their disadvantages include the high cost of heating equipment and operating costs, as well as relatively large unit electrical power. Economically feasible steam turbine plants should have an electrical capacity starting from 1 MW, but preferably 5–6 MW and above. Plants on low-boiling working mediums (ORC turbines) can be designed for lower power, but at the same time they are less efficient compared to conventional steam turbines and have a higher price per unit of installed capacity [18, 19]. As noted earlier, piston engines and gas turbines require certified fuels.

In [20], a highly efficient power plant operating on the Brayton thermodynamic cycle and energy recuperation is proposed. The paper considers the possibility of its application and adjustment to the conditions provided by a heat generator fueled by uncertified biofuel. This plant has a relatively simple design, which gives reasons to hope for a low cost per unit of installed capacity.

In the future, it is planned to develop a highly efficient turbine, compressor and heat exchange equipment for this plant with the help of modern design methodologies and the experience of the authors of the paper [21, 22, 23].

Thermal scheme, thermodynamic cycle, analysis

Fig. 1 shows the thermal scheme, and Fig. 2 – thermodynamic Brayton cycle of a power plant with energy recuperation. The plant consists of the following main elements: a compressor (C), a turbine (T), a heat exchanger, a heat generator (HG) and an electric generator (G). The compressor and turbine usually are located on the same shaft (turbo-compressor), and the electric generator can be connected to the turbo-compressor shaft either directly or through a reducer.

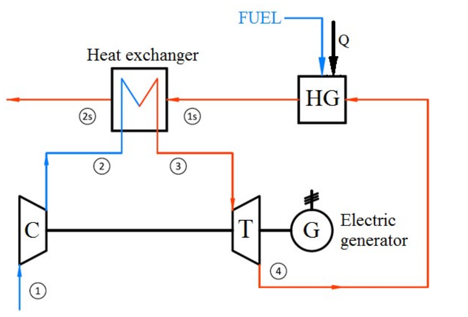

Fig. 1. Thermal scheme of the plant with energy recuperation

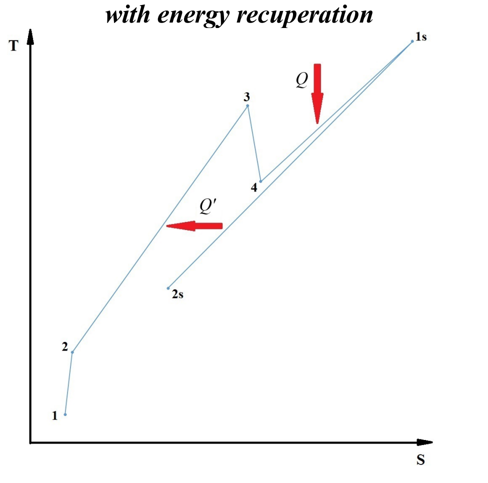

Fig. 2. Scheme of Brayton's thermodynamic cycle with energy recuperation

According to the thermal scheme and thermodynamic cycle shown in Figs. 1 and 2, the principle of operation of the plant is as follows.

Air is sucked in by the compressor from the environment (point 1), compressed in it and then supplied to the inner part of the heat exchanger (point 2). At this stage, the mechanical energy supplied to the compressor is transferred to the air. The real compression process occurs with an increase in entropy (points 1–2), while the gas-dynamic efficiency of the compressor is evaluated relative to the "ideal" (isentropic) process. Then the isentropic efficiency of the compressor can be calculated according to the formula

$$\eta_c = \frac{h_{2iz} - h_1}{h_2 - h_1} \cdot 100\% , \qquad (1)$$

where h is the total (inhibited) enthalpy; iz is the isentropic process, other indices correspond to the points in Figs. 1 and 2.

In the inner part of the heat exchanger due to the introduction of heat Q', the air is heated and fed to the turbine inlet (point 3). In the heat exchanger, the air loses its full pressure due to aerodynamic resistance. Losses of total pressure are calculated by the formula

$$\overline{P}_{\text{in}}^{\text{total}} = \frac{P_2 - P_3}{P_2} \cdot 100\%, \quad (2)$$

where P is the full pressure, the indices correspond to the points in Figs. 1 and 2.

In the process of passing through the turbine (points 3–4), the air expands, and its energy is converted into mechanical energy on the turbine shaft. The mechanical energy from the turbine is used to rotate the rotors of the compressor and the electric generator. Just like in the compressor, the real process in the turbine is not isentropic, its efficiency is determined by

$$\eta_{\text{t}} = \frac{h_3 - h_4}{h_3 - h_{4iz}} \cdot 100\%, \quad (3)$$

where h is the total (inhibited) enthalpy; iz is the isentropic process, other indices correspond to the points in Figs. 1 and 2.

If after the turbine (point 4) the air is discharged into the atmosphere, it will be a plant that works according to the classic Brayton cycle. However, unlike other power plants (gas turbines, aviation gas turbines, and others), in it, the energy to the working medium after the compressor is supplied not to the internal combustion chamber (when the air from the compressor is directly used as an oxidizer during fuel combustion), but to the external heat exchanger. Such an energy supply scheme has both disadvantages and advantages. The main disadvantages of using an external heat exchanger are the need for additional heating equipment (a heat exchanger and an external heat generator) and an increase in the dimensions of the plant. Among the advantages, the most significant are: the possibility of using non-certified fuels, including solid ones and biofuels, as well as the organization of effective energy recuperation. Figs. 1 and 2 show one of the methods of effective energy recuperation organization.

If the air after the turbine is (fully or partially) directed to the heat generator, where it is used as an oxidizer in the process of fuel combustion, then due to the fact that the air behind the turbine always has a greater internal energy compared to the ambient air (based on the temperature difference between the points 4 and 1), this energy is returned to the cycle, that is, less fuel must be used to heat the working medium in the heat generator (points 4–1s). In the heat generator, heat Q is supplied to the working medium. Total pressure losses in the heat generator are determined by the formula

$$\overline{P}_{\text{hg}}^{\text{total}} = \frac{P_4 - P_{1s}}{P_4} \cdot 100\%, \quad (4)$$

where P is the full pressure, the indices correspond to the points in Figs. 1 and 2.

After the heat generator, the working medium is directed to the outer part of the heat exchanger (point s), where it gives off heat Q' to the air from the compressor, and then goes outside (point 2s). Due to energy recuperation, heat Q' is greater compared to heat Q (Fig. 2). Losses of total pressure outside of the heat exchanger are determined by the formula

$$\overline{P}_{\text{out}}^{\text{total}} = \frac{P_{1s} - P_{2s}}{P_{1s}} \cdot 100\%, \quad (5)$$

where P is the full pressure, indices correspond to points in the Figs. 1 and 2.

The efficiency of the power plant without taking into account mechanical losses and the efficiency of the electric generator can be expressed as

$$\eta = \frac{N_y}{Q} \cdot 100\%, \quad (6)$$

where Q is the heat supplied to the heat generator as a result of fuel combustion; $N_y = N_{\text{t}} – N_{\text{c}}$ is the useful power of the turbocharger; N_{t} is the turbine power; N_{c} is the compressor power.

A power plant can be considered adiabatic, i.e. one in which there is no heat exchange with the environment, when

$$N_{\text{c}} = (h_2 - h_1) \cdot G_{\text{c}}; \; N_{\text{t}}(h_3 - h_4) \cdot G_{\text{t}}; \; Q = (h_4 - h_{1s}) \cdot G_{\text{hg}}; \; Q' = (h_{1s} - h_{2s}) \cdot G_{\text{hg}} = (h_3 - h_2) \cdot G_{\text{c}}, \quad (7)$$

where G_c is the mass flow rate of air in the compressor; G_t is the mass flow rate of air in the turbine; G_{hg} – is the mass flow rate of the working medium at the output of the heat generator.

Since the goal of the paper is a qualitative assessment of the power plant efficiency, to simplify the analysis, some assumptions that do not significantly affect the accuracy of the calculations can be made:

1. The mass flow rates of air in the compressor, the internal part of the heat exchanger and the turbine are the same.

2. All the air after the turbine is sent to the heat generator. In case if $T_4>T_2$, this condition, from a theoretical point of view, ensures the greatest recuperation efficiency. If $T_4<T_2$, then it is more rational to send only part of the air after the turbine to the heat generator.

3. The amount of fuel that is added to the heat generator is considered small and is not taken into account, therefore the mass flow rate of the working medium in the heat generator and the external part of the heat exchanger is the same.

4. The properties of the working medium in the entire tract of the power plant correspond to the equation of state of a perfect gas for air (γ=1.4; R=287.3 J/(kg·K)).

Taking into account the assumptions, we have

$$G = G_c = G_t = G_{hg}\,;\; Q = Q' - G(h_4 - h_{2s})\,;\; h = c_p T\,, \tag{8}$$

where c_p is the coefficient of specific isobaric heat capacity; T – temperature.

Then equation (6) taking into account (7–8) can be written in the form

$$\eta = \left(1 - \frac{T_{2s} - T_1}{T_{1s} - T_4}\right) \cdot 100\%\,. \tag{9}$$

At the same time, the fulfillment of the second law of thermodynamics is ensured under the conditions $T_{2s} \geq T_2$, $T_{1s} \leq T_3$. It is interesting to consider the efficiency of the "ideal" Brayton cycle with recuperation. For this, let's assume that $T_{2s}=T_2$, $T_{1s}=T_3$. The efficiency of the compressor and turbine is 100% (equations (1), (3)), and the pressure losses are equal to 0 (equations (2), (4), (5)). Then equation (9) will take the form [21]

$$\eta = \left(1 - \frac{T_1}{T_3}\pi^{\frac{\gamma-1}{\gamma}}\right) \cdot 100\%\,. \tag{10}$$

where $\pi = \pi_c = \pi_t = \dfrac{p_2}{p_1} = \dfrac{p_3}{p_4}$ is the degree of pressure increase/decrease in the compressor / turbine. In case when the limit option is set as π=1, which is impossible in reality, then the thermal efficiency of this cycle (equation (10)) will be equal to the efficiency of the ideal Carnot cycle [24].

The efficiency of the ideal Brayton cycle without recuperation [25] has the form

$$\eta = \left(1 - \frac{1}{\pi^{\frac{\gamma-1}{\gamma}}}\right) \cdot 100\%\,. \tag{11}$$

From the analysis of equations (10) and (11), it can be seen that in a cycle without recuperation, the efficiency does not depend on temperature, but increases only with an increase in π. The cycle with recuperation, on contrary, depends on cycle temperatures and decreases with increasing π.

In general, it can be concluded that in the ideal Brayton cycle with recuperation at relatively small values of π and the heating temperature (T_3), the thermodynamic efficiency will be significantly higher compared to the ideal classical Brayton cycle. For comparison, at π=2, T_3=500 ºC and T_3=15 ºC , the thermal efficiency of ideal Brayton cycles without and with recuperation will be 18% and 54%, respectively. To achieve a similar value of efficiency in a conventional ideal Brayton cycle, it is necessary to provide $\pi \approx 15$, which is a difficult technical task. Of course, for real power plants, their characteristics will differ significantly from ideal ones.

Results and discussion

In order to evaluate the efficiency of a real power plant operating on the Brayton thermodynamic cycle with energy recuperation, several options with different technical characteristics of the equipment components were developed. Calculations were performed according to approximate formulas (1)–(5), (9). The efficiency of the compressor and turbine, pressure loss, as well as the air heating temperature in the inner part of the heat exchanger (T_3) and the environment temperature (T_1) were considered as initial data. For the assumptions made earlier, the following condition are met:

$$\Delta T = T_{1s} - T_3 = T_{2s} - T_2 .$$

The value ΔT is set for calculations, it shows the temperature difference between the working medium in the internal and external parts of the heat exchanger.

Calculations were made under the following conditions:

1. ΔT=50 ºC. Such a condition is quite difficult for technical implementation, first of all for the heat exchanger, but at the same time it is quite achievable.

2. Pressure losses in the internal part of the heat exchanger, total pressure losses in the heat generator ($\overline{P}_{\text{hg}}^{\text{total}}$) and the outer part of the heat exchanger ($\overline{P}_{\text{out}}^{\text{total}}$) are 7 %.

3. Two compressor and turbine efficiency options are η_c=80%, η_t=85% and η_c=85%, η_t=90%.

4. Ambient temperature T_1=20 ºC, and the heating temperature T_3 of the working medium in the inner part of the heat exchanger was considered in the range from 500 to 850 ºC. The minimum value of the temperature range is limited by the acceptable value of the minimum efficiency of the plant, and the maximum is limited by the cost of the material and production of the heat exchanger.

5. The degree of pressure reduction in the turbine π_t was considered in the range from 2 to 4, while the degree of pressure increase in the compressor π_c was calculated based on the pressure losses according to p. 2. The relatively small maximum value of π_t was taken while bearing in mind the conclusion about the negative impact of its growth on the thermal efficiency according to the equation (10).

Table and Fig. 3 show the results of calculations of a power plant operating on the Brayton thermodynamic cycle with energy recuperation, depending on the conditions described above.

It can be seen from the above results that the maximum temperature of the working medium T_3 has the most significant effect on the thermal efficiency. The internal efficiency of the compressor and turbine also significantly affects the thermal efficiency of the plant. To increase the thermal efficiency, it is necessary to strive to increase the temperature T_3 and the internal efficiencies of the compressor and turbine. The degree of pressure reduction/increase in the turbine/compressor has a less significant effect on the thermal efficiency, and the given dependences, as well as for the ideal thermal efficiency (equation (10)), show that the efficiency mainly increases with reduced degrees of pressure reduction / increase in the turbine / compressor.

The specific mass consumption of the working medium (relative to the produced power) affects the mass-dimensional characteristics of the power plant and, accordingly, its cost. The value of the specific mass flow rate is also most affected by the temperature T_3 and the internal efficiency of the turbine and compressor. Moreover, this effect is greater at lower temperatures T_3, which indicates the impossibility of reducing it below 500 ºC. The degree of pressure reduction (increase) in the turbine (compressor) has a less significant effect on the specific consumption of the working medium.

Table. Energy characteristics of the power generating plant

T, °C	Parameter	η_c=80%, η_t=85%			η_c=85%, η_t=90%		
	π_t	2	3	4	2	3	4
500	η, %	12.9	10.5	6.5	19.6	18.2	15.0
	G, kg/s	4.58	4.17	5.77	2.90	2.30	2.40
600	η, %	20.2	18.7	15.4	26.4	25.8	23.1
	G, kg/s	2.69	2.13	2.21	1.97	1.47	1.40
700	η, %	26.3	25.5	22.7	32.2	32.1	29.8
	G, kg/s	1.90	1.43	1.37	1.49	1.08	0.99
800	η, %	31.6	31.3	28.9	37.1	37.4	35.5
	G, kg/s	1.47	1.07	0.99	1.20	0.86	0.77
850	η, %	33.9	33.9	31.6	39.3	39.8	38.0
	G, kg/s	1.32	0.96	0.87	1.09	0.78	0.69

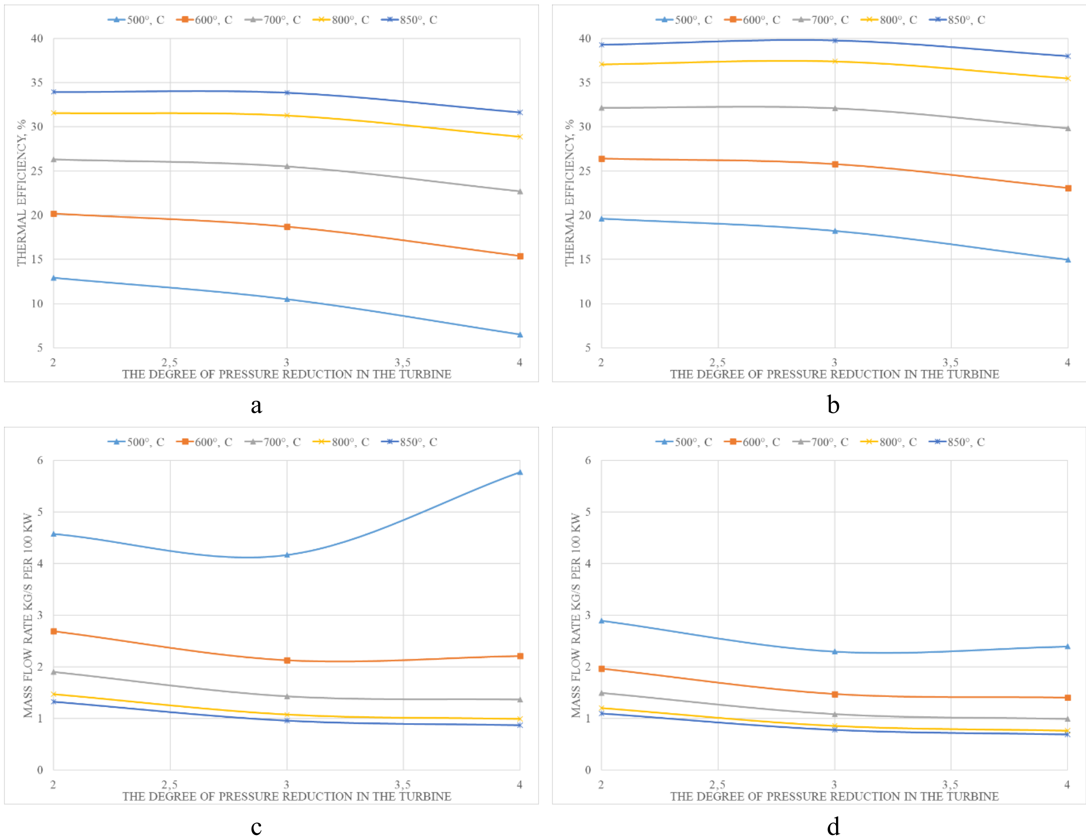

Fig. 3. Characteristics of the plant with energy recuperation:
a, b – thermodynamic efficiency; c, d – mass consumption of the working medium per 100 kW of power;
a, c – η_c=80%, η_t=85%; b, d – η_c=85%, η_t=90%

Thus, it can be concluded that even at a relatively low temperature T_3 (500 ºC), under the condition of high efficiency of the compressor and turbine (η_c=85%, η_t=90%), it is possible to achieve good characteristics of the power plant as a whole (thermal efficiency is almost 20%). Taking into account the fact that this plant is intended for operation on non-standard (cheap) fuel, this level of its efficiency is quite acceptable. When the temperature T_3 increases to 850 ºC, which is not too high, a thermal efficiency of 40% is achieved, but in this case the cost of materials and production increases.

Conclusions

The thermal scheme of a power generating plant with a remote heat exchanger operating according to the Brayton cycle with energy recuperation is considered. It is assumed that the plant will work on non-standard (cheap) biofuel.

It is shown that, unlike the usual Brayton cycle, in the cycle with energy recuperation, the heating temperature of the working medium and the internal efficiency of the main components of the plant, such as the compressor and the turbine, have the greatest influence on the thermal efficiency. Also, unlike the usual Brayton cycle, the higher efficiency of the plant is achieved with smaller degrees of pressure reduction (increase) in the turbine (compressor).

It has been established that even at a relatively low temperature of the working medium heating (500 ºC), under the condition of high efficiency of the compressor and turbine (η_c=85%, η_t=90%), it is possible to achieve good characteristics of the power plant as a whole (thermal efficiency of almost 20%). When the heating temperature of the working medium is increased to 850 ºC, which is not too high, a thermal efficiency of 40% is achieved, but in this case the cost of materials and production increases.

For a final conclusion about the possibility of using the proposed plant and its efficiency, it is necessary to conduct additional studies, in particular, of its main elements, such as the compressor, turbine, heat exchangers and others.

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Аналіз ефективності електрогенеруючої установки, що працює на основі термодинамічного циклу Брайтона та рекуперації енергії

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Розглянуто теплову схему електрогенеруючої установки з виносним теплообмінником, що працює за циклом Брайтона з рекуперацією енергії. Передбачається, що установка працюватиме на несертифікованому (дешевому) біопаливі. Показано, що на відміну від звичайного циклу Брайтона у циклі з рекуперацією енергії найбільший вплив на термічний ККД має температура нагріву робочого тіла і внутрішній ККД основних складових установки, таких, як компресор і турбіна. Також на відміну від звичайного циклу Брайтона більш висока ефективність установки досягається при менших ступенях зниження (підвищення) тиску в турбіні (компресорі). Встановлено, що навіть при відносно низькій температурі нагріву робочого тіла (500 ºC), за умови високої ефективності компресора і турбіни, можна досягти гарних характеристик енергетичної установки в цілому. При температурі до 850 º C досягається термічний ККД на рівні 40%, але у цьому випадку збільшується вартість матеріалів і виробництва. Для остаточного висновку про можливість застосування запропонованої установки й її ефективність необхідно провести додаткові дослідження, зокрема, таких її основних елементів, як компресор, турбіна, теплообмінник та інші.

***Ключові слова**: теплова схема, електрогенеруюча установка, цикл Брайтона, рекуперація енергії, термічний ККД, турбіна, компресор, ефективність.*

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