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DECREASING THE MASS INDICES OF GAS TURBINE ENGINES REGENERATORS BY MEANS OF CHOOSING RATIONAL PARAMETERS

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Отримані спрощені залежності між геометричними параметрами поверхні теплообміну, ефективністю та втратами тиску у регенераторі, ККД газотурбінного двигуна. Показано, що раціональним вибором ефективності та втрат тиску можливо добитись істотного зниження маси регенератору при постійному ККД. Розроблено алгоритм пошуку раціональних параметрів регенератору. Проведено аналіз параметрів регенераторів газотурбінних двигунів різних схемних рішень

Ключові слова: газотурбінний двигун, регенератор, коефіцієнт корисної дії, маса регенератору, ефективність регенератору

Получены упрощенные зависимости между геометрическими параметрами поверхности теплообмена, эффективностью и потерями давления в регенераторе, КПД газотурбинного двигателя. Показано, что рациональным выбором эффективности и потерь давления можно добиться существенного снижения массы регенератора при постоянном КПД. Разработан алгоритм поиска рациональных параметров регенератора. Проведен анализ параметров регенераторов газотурбинных двигателей разных схемных решений

Ключевые слова: газотурбинный двигатель, регенератор, коэффициент полезного действия, масса регенератора, эффективность регенератора

1. Introduction

Under conditions of constant growth of fuel prices, reduction of fuel stocks and possible interruption in deliveries, one of the main tasks in the process of creating plants for the generation of mechanical and thermal energy is to decrease the consumption of energy resources [1]. A strategic task is the development of the stationary and transportation plants with performance efficiency (PE) at the level of 50 % and higher, with high parameters of reliability, functionality, and maintainability [2].

In the global energy sector, the introduction of gas turbine plants (GTP) with gas turbine engines (GTE) is considered to be one of the promising directions of increasing the efficiency of fuel consumption in the process of production of mechanical and thermal energy. GTE have the following advantages over other heat engines: high energy capacity, small mass and dimensions indices, short terms of putting into operation, possibility of maximum proximity to the consumer, high maneuverability, ease of repair and maintenance [3]. However, an important shortcoming of GTE is relatively low performance efficiency (PE), which is currently 34–38 % [2, 3].

Simultaneously with the development of the traditional way of increasing PE by increasing the initial temperature of gases in the cycle (t_3), the transition to the thermodynamically complex cycles of GTE is becoming important in the world science and practice. In the energy sector, combined gas-to-vapor turbine plants (PE is 45–52 %), GTE with a vapor injector and of the "Vodoley" type (PE is 42–45 %), with intermediate cooling, overheating and regeneration of heat from exhaust gases (PE is 40–45 %) are becoming increasingly common [2–13].

From the specified variants, regeneration is the easiest way to improve PE of GTE. High PE is achieved at $t_3 \leq 900$–950 °C, which allows us to do without cooling the turbine stages. With current level of gas temperatures, regeneration allows achieving PE at the level of 40–42 %, which, for a simple cycle, corresponds to t_3=1300–1400 °C and asguires the use of new systems for cooling turbines and heat resisting materials [3, 5, 12]. However, the introduction of regenerator plant leads to an increase in mass and costs, as well as an increase in the start-up time and time of transition from mode to mode, and complicates the structure of GTE. Given this, an important task is to decrease mass of the regenerator and enhance its compactness [2–5, 10, 12, 13].

The main characteristics of the regenerator are a degree of heat regeneration (r) or regenerator effectiveness (ε) and pressure losses ($\delta P_{R\Sigma}$), which have a significant impact on both PE and capacity of GTE [14, 15], and mass and dimensions of the regenerator [10, 14]. Non-rational choice of effectiveness value at the stage of the calculation of GTE cycle in the course of designing a regenerator may lead to overestimating its mass by several times [10]. An attempt to decrease mass of the regenerator by selecting lower values of effectiveness can lead to a decrease in PE of the GTE below the level listed in economic calculations or State programs [1–5].

Taking the aforementioned into account, a relevant problem is determining such levels of effectiveness and pressure losses that would ensure minimum mass of the regenerator at the asguired PE.

2. Literature review and problem statement

At present, in the process of designing regenerative GTE, there has developed the practice of consistent (independent) calculation of the cycle and the regenerator. In the beginning of designing, parameters of the GTE and the regenerator are chosen regarding the existing experience [3, 5], then calculations of the cycle and the search for its optimal parameters are performed [12, 14], the results of which are used at the following stages of designing a regenerator [16, 17]. When necessary, by the results of regenerator designing, initial data for the calculation of cycle are corrected and it is repeated [12]. Since the methods of calculating GTE and the regenerator, used in the process of performing such tasks, are complicated, they take quite a long period of time.

The main shortcoming of this approach to designing regenerative GTE is that the regenerator effectiveness and pressure losses are selected at the stage of conceptual designing, when the main parameters of GTE are defined. In this case, their essential impact on the mass and dimensions of the regenerator are not taken into account, as a result of which a slight modification in the given magnitudes may lead to an increase in mass by several times [10]. In this case, structural, technological and transportation restrictions, imposed on the regenerator [18, 19] are not considered either, but complying with them may lead to substantial growth of its mass.

When analyzing different structural solutions of regenerative GTE, authors usually focus on the GTE efficiency [9, 20–26] and do not take into account the influence of parameters of the regenerator on its mass and the mass and costs of the plant. As a result, non-rational choice of parameters of the regenerator may lead to such increase in mass so that the "optimal" parameters of the plant, obtained in the analysis of the cycle, cannot be implemented in practice. This approach is particularly critical when using regeneration in transportation GTE [23], in which tight restrictions are imposed on the mass of the plant.

On the other hand, when analyzing the structures of regenerators, choosing their geometrical parameters and the method for heat exchange intensification, authors focus on energy parameters or on the mass and do not consider the influence of parameters of the regenerator on GTE. Thus, in paper [27], a comparison of intensified heat transfer surfaces was conducted by the mass of the obtained regenerators, but it was not taken into account that different pressure losses in them would lead to different values of efficiency and capacity of the engine.

An attempt of simultaneous analysis of influence of parameters of regenerator both on its mass (cost) and on the effectiveness of GTE, was performed in [28]. However, all considered variants had different mass of regenerators and effectiveness of plants, which made their correct comparison impossible, and recommendations for selecting parameters of regenerators were not obtained.

It should be noted that the techniques of calculations that were used in the examined papers are complicated, they take into account a large number of parameters. The advantage of such techniques is their high accuracy. The disadvantage is the need to perform a significant amount of cumbersome calculations, which considerably complicates the choice of parameters of the regenerator at the initial stages of designing.

Thus, there are practically no approaches to a comprehensive analysis of parameters of regenerative GTE and their regenerators and the choice of their rational values at the initial stages of designing, at which the efficiency will be maximal, and the mass-dimensional and cost indices will be reasonable. To speed up this process, it would be advisable to use simplified models and techniques for determining parameters of regenerators of GTE.

3. The aim and tasks of the study

The aim of the study is to develop a method for choosing rational energy and geometrical parameters of regenerators of GTE to obtain minimum mass and dimensions of the heat exchanger at the given values of fuel efficiency of GTE.

To ahieve this aim, the following tasks were to be solved:

– establishing the relationship between efficiency of GTE, energy parameters of the regenerator, geometrical parameters of the heat exchange surface and the mass and dimensions indicators of the regenerator;

– development of the algorithm for choosing rational parameters of GTE and the regenerator at the initial stages of designing, taking into account the mass and dimensions of the regenerator;

– analysis of effectiveness of regenerative GTE of different structural solutions and the choice of rational parameters of the cycle and the regenerator.

4. Determining the interrelation between parameters of the regenerator, its mass, efficiency and GTE capacity

4. 1. Influence of regenerator parameters on efficiency of regenerative GTE

In a general case, GTE of a complex scheme with heat regeneration can structurally consist of the following elements:

– compressors (C, in the number of n_C);

– combustion chambers (CC, in the number of n_{CC}), one of which is primary and is placed before the start of expansion process, the others (additional) are placed between turbines or turbines stages;

– turbines (T, in the number of n_T), one of which (power turbine) is designed for the rotation of mechanical power user (propeller, generator, pump), the others (turbine of compressors, TC) are used to drive compressors, the schemes without a separate power turbine are possible, when the shaft of user is connected to one of the turbocompressors;

– regenerators (R), in which the heat of hot gases after expansion in the turbine (turbines) is used for heating the air before combustion chambers (with some CC – before the primary one). Regenerator can be located both after turbines (final regeneration) and between them (intermediate regeneration).

The structure of GTE may include air coolers: intermediate (IC, in the number of n_{IC}), installed between compressors or stages of one compressor; input, which cools air at the inlet to GTE; and the final, which is placed at the end of the compression process before the regenerator. A generalized cycle of regenerative GTE of complex scheme is presented in Fig. 1, *a*, *b*.

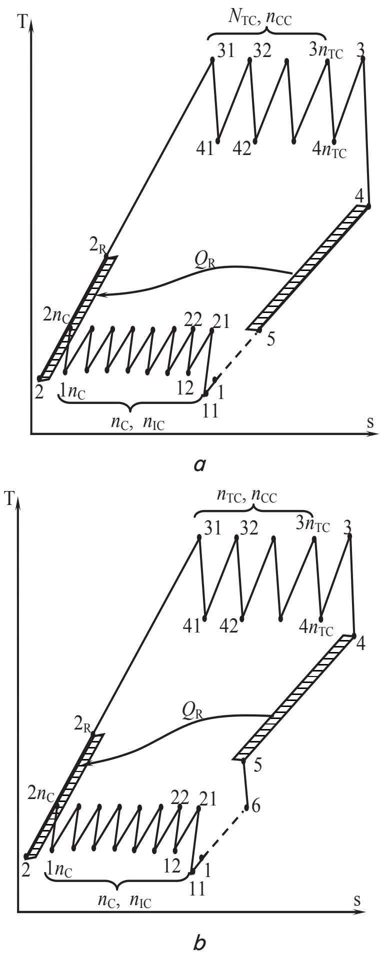


Fig. 1. Generalized cycle of GTE of complex scheme with heat regeneration: a – with final regeneration; b – with intermediate regeneration; 1–11 – air cooling in input cooler, 11–21, ..., $1n_C$ – $2n_C$ – air compression in compressors; 21–12, ..., $2n_C$ $1n_C$ – air cooling in intermediate cooler; $2n_C$–2 – air cooling in final cooler; 2–2_R – air heating in the process of regeneration; 2_R–31 – heating in the main combustion chamber; 41–32, ..., $4n_{C-1}$–$3n_C$ – heating in the intermediate combustion chambers, 31–41, ..., $3n_C$–$4n_C$ – expansion in the turbines of compressors; 3–4 – expansion in the turbines before regenerator; 4–5 – cooling of the working medium in regenerator; 5–6 – expansion in the turbines with intermediate regeneration

A scheme of the plant may involve the injection of water into coolers, water or vapor into combustion chambers to decrease the emissions of nitrogen oxides and to increase the GTE efficiency, intakes of compressed air (for cooling the blades of turbine, for the GTE lubrication systems, for external users, if any), intakes of exhaust gases (to utilization boilers, for recycling, for external heat consumers).

The GTD efficiency is determined as [14]

$$\eta = \frac{N_e}{Q_{CC}}, \tag{1}$$

where N_e is the capacity of GTE; $Q_{CC\Sigma}$ is the total amount of heat supplied to GTP cycle, which is equal to the sum of fuel heats, emitted in all combustion chambers.

In the regenerative GTE, amount of heat $Q_{CC\Sigma}$, which is needed for heating the working medium to the asguired temperature, is less than the correspondent amount of heat in the GTE without regeneration ($Q'_{CC\Sigma}$) by magnitude ΔQ_R, and capacity N_e is lower than the capacity of GTE without regeneration (N'_e) by magnitude ΔN_{eR}, hence:

$$\eta = \frac{N'_e - N_{eR}}{Q_{CC} - Q_R} = \eta' \frac{1 - \delta N_{eR}}{1 - Q_R / Q_{CC}}, \tag{2}$$

where $\delta N_{eR} = \Delta N_{eR} / N'_e$ is the relative loss of capacity of GTE as a result of the introduction of regeneration; η' is the efficiency of GTE without heat regeneration (here and further parameters without a stroke refer to regenerative GTE, with a stroke refer to GTE without regeneration).

It follows from Fig. 1 that heat regeneration mostly influences the process of heating the working medium in the first (main) combustion chamber. The amount of heat, brought to CC to the working medium, will equal

$$Q_{CC1} = G_f \cdot Q_w^{low} \cdot \eta_{CC} = G_{31} \cdot c_{p\,0-31} \cdot t_{31} - G_{2R} \cdot c_{p\,0-2R} \cdot t_{2R} \tag{3}$$

and without regeneration:

$$Q'_{CC1} = G'_f \cdot Q_w^{low} \cdot \eta_{CC} = G'_{31} \cdot c_{p\,0-31} \cdot t_{31} - G'_2 \cdot c_{p\,0-2} \cdot t_2, \tag{4}$$

where G, kg/s is the consumption of working medium, $c_{p\,t1-t2}$, J/(kg×K) is the specific average mass isobar thermal capacity of the working medium in the range of temperatures t_1 and t_2, G_f, kg/s is the fuel consumption, Q_w^{low}, kJ/kg is the lower working heat of fuel consumption; η_{CC} is the efficiency of combustion chamber.

The total amount of heat Q_R, which is transferred in the GTE regenerator, according to Fig. 1, can be represented in the form:

$$Q_R = G_2 \cdot c_{p\,0-2R} \cdot t_{2R} - G_2 \cdot c_{p\,0-2} \cdot t_2. \tag{5}$$

Considering it, we will receive:

$$\Delta Q_R = Q'_{CC1} - Q_{CC1} = \left(G'_{31} \cdot c_{p\,0-31} - G_{31} \cdot c_{p\,0-31}\right) \cdot t_{31} + Q_R. \tag{6}$$

Based on the material balance of CC [14] (the heat of fuel that has not been combusted is not taken into account due to its small share in the gas mixture at the outlet of CC), heat capacity of gas at the outlet from it can be represented as follows:

$$c_{p\,0-31} = \left(G_п \cdot c_{pп\,0-31} + G_{пз} \cdot c_{pпз\,0-31}\right) / G_{31}, \tag{7}$$

where $G_{pp} = G_f \eta_{CC}(L_0+1)$ kg/s is the amount of pure products of fuel consumption as a result of fuel combustion, $G_a = G_{31} - \eta_{CC} G_f L_0$ kg/s is the amount of the remained air, L_0, kg/kg is theoretically necessary amount of oxidizer for combusting 1 kg of fuel, α_{CC} is the coefficient of excessive air.

For the cycle without regeneration, dependence will be similar.

After substitution of (7) in (6) and transformations, we will obtain:

$$\Delta Q_R = C_t \cdot \Delta Q_R, \tag{8}$$

where

$$C_t = \left(1-\left(\left(L_0+1\right)\cdot c_{p\,pp\,0-31} - L_0\cdot c_{p\,a\,0-31}\right)\cdot t_{31} / Q_w^{low}\right)^{-1}, \quad (9)$$

C_t is the coefficient that considers the influence of change in consumption of the working medium, which follows from the main CC, on the GTE efficiency.

The heat amount, which is transmitted in regenerator, can be defined as [14, 17]:

$$Q_R = W_{Rh}\cdot\left(t_4-t_5\right) = W_{Rc}\cdot\left(t_{2R}-t_2\right), \quad (10)$$

where $W_{R\,h}=G_4 \times c_{p\,5\text{-}4}$ is the water equivalent of a hot heat carrier, $W_{R\,c}=G_2 \times c_{p\,2\text{-}2R}$ is the water equivalent of a cold heat carrier.

According to [14], the heat amount transmitted in regenerator as a common heat exchanger, can be represented in the form

$$Q_R = \varepsilon\cdot Q_{max}, \quad (11)$$

where ε is the thermal effectiveness, Q_{max} is the maximum amount of heat that can be transferred from one heat carrier to another:

$$Q_{max} = W_{min}\cdot\left(t_4-t_2\right), \quad (12)$$

where W_{min} is the least of the two water equivalents $W_{R\,h}$ and $W_{R\,c}$.

Considering (8) and (11), efficiency of GTE with regeneration can be represented as follows:

$$\eta = C_\eta\cdot\eta', \quad (13)$$

where $C_\eta=\eta'(1-\delta N_{eR})/(1-C_t\cdot C_Q\cdot\varepsilon)$ is the relative increase in the GTE efficiency as a result of introduction of heat regeneration; $C_Q=Q_{max}/Q'_{CC1}$ is the coefficient that indicates what maximum part of the heat that is supplied to the CC can be transferred with full regeneration of heat (ε=1) in the regenerator.

To establish the connection between the work losses and pressure losses in the regenerator, we will introduce the following magnitudes:

– degree of pressure reduction in turbines from the first (main) CC to the last CC – π_{T1};

– degree of pressure reduction in turbines after the last CC before the regenerator – π_{T2};

– degree of pressure reduction in turbines after the regenerator – π_{T3}.

If the degree of pressure reduction in turbines of GTE without regeneration is designated as π'_T, then the total degree of pressure reduction in GTE turbines with regeneration:

$$\pi_T = \nu_R\cdot\pi'_T, \quad (14)$$

where ν_R is the coefficient of pressure losses in regenerator, which is defined as

$$\nu_R = \left(1-\delta P_{Rc}\right)/\left(1-\delta P_{Rh}\right), \quad (15)$$

where $\delta P_{Rc}=\Delta P_{Rc}/P_{Rc\,in}$ and $\delta P_{Rh}=\Delta P_{Rh}/P_{Rh\,in}$ are the relative losses of pressure of hot and cold heat carrier in the regenerator, ΔP_{Rc} and ΔP_{Rh}, Pa are the absolute losses of pressure of hot and cold heat carrier in the regenerator, $P_{Rc\,in}$ and $P_{Rh\,in}$, Pa is the pressure of hot and cold heat carrier at the inlet of the regenerator.

The point of introduction of heat regeneration will be assigned via degree of pressure reduction in turbines after the regenerator π_{T3} (we accept $\pi_{T3}=\pi'_{T3}$). Then the degree of pressure reduction in turbines from the last CC to the point of introduction of regeneration to the cycle for non-regenerative cycle

$$\pi'_{T2} = \frac{\pi'_T}{\pi'_{T1}\cdot\pi_{T3}} \quad (16)$$

and regenerative:

$$\pi_{T2} = \frac{\pi_T}{\pi_{T1}\cdot\pi_{T3}} = \nu_R\cdot\pi'_{T2}. \quad (17)$$

Useful work of the GTE cycle can be represented as follows:

$$N_e = N_{T\Sigma} - N_{C\Sigma}, \quad (18)$$

where $N_{T\Sigma}$ is the total capacity of turbines; $N_{C\Sigma}$ is the total capacity consumed by compressors considering mechanical losses.

The total capacity of turbines can be represented as

$$N_{T\Sigma} = N_{T1} + N_{T2} + N_{T3}, \quad (19)$$

where N_{T1} is the capacity of expansion processes between the first and the last combustion chambers (with degree of pressure reduction π_{T1}), N_{T2} is the capacity of expansion processes between the last combustion chamber and regenerator (with degree pressure reduction π_{T2}), N_{T3} is the capacity of expansion processes after the regenerator (with degree of pressure reduction π_{T3}).

Since heat regeneration does not affect the compression process, capacities of the compressors will be the same for regenerative and non-regenerative cycles ($N_{C\Sigma}=N'_{C\Sigma}$), hence the equal powers of turbines that set compressors into motion. Taking into account that additional CC are commonly placed between the turbines of compressors [14] or immediately after them, it is possible to accept $N_{T1}=N'_{T1}$ and $\pi_{T1}=\pi'_{T1}$.

Magnitude N_{T2} for regenerative GTE can be represented as follows:

$$N_{T2} = G_{3n}\cdot c_{p\,4n-3n}\cdot T_{3n}\cdot\left(1-\pi_{T2}^{\;-m_{T2}}\right)\cdot\eta_{T2}, \quad (20)$$

where η_{T2} and $m_{T2}=R_g/c_{p\,4n-3n}$ are the efficiency and index of polytrope of conditional process of expansion of gas from the point of outlet from the last CC to the outlet from the last stage of turbine before the regenerator; R_g, J/(kg×K) is the gas constant.

Similarly for the cycle without regeneration

$$N'_{T2} = G'_{3n}\cdot c'_{p\,4n-3n}\cdot T_{3n}\cdot\left(1-\pi'^{\;-m'_{T2}}_{T2}\right)\cdot\eta_{T2}. \quad (21)$$

Next in calculations we will consider that as a result of an insignificant change in temperature and in composition of the working medium, properties of the working medium for the cycle without regeneration and with regeneration can be considered identical and can be defined as:

$$\eta_{T2} = N'_{T2} / \left(G'_{3n} \cdot c'_{p\,4n-3n} \cdot T_{3n} \cdot \left(1 - \pi'^{\,-m'_{T2}}_{T2} \right) \right). \qquad (22)$$

After substitution from (17) and (22) to (20) and transformations we will obtain

$$N_{T2} = N'_{T2} - \Delta N_{R\upsilon} - \Delta N_{Rg2} - \Delta N_{R\upsilon g}, \qquad (23)$$

where

$$\Delta N_{R\upsilon} = N'_{T2} \cdot \left(\nu_R^{-m_{T2}} - 1 \right) / \left(\pi'^{\,-m_{T2}}_{T2} - 1 \right)$$

– a change in capacity from pressure losses in the regenerator;

$$\Delta N_{Rg2} = N'_{T2} \cdot \left(G'_f - G_f \right) / G'_3$$

– a change in capacity due to a decrease in consumption of the working medium in regenerative cycle;

$$\Delta N_{R\upsilon g} = N'_{T2} \cdot \left(\nu_R^{-m_{T2}} - 1 \right) / \left(\pi'^{\,-m_{T2}}_{T2} - 1 \right) \cdot \left(G'_f - G_f \right) / G'_3$$

– a change in capacity, which depends on the influence of pressure losses and decrease in consumption of the working medium, unconsidered above.

Magnitude N_{T3} for regenerative GTE can be represented as follows:

$$N_{T3} = G_5 \cdot c_{p\,6-5} \cdot T_5 \cdot \left(1 - \pi_{T2}^{-m_{T2}} \right) \cdot \eta_{T2}, \qquad (24)$$

where η_{T3} and $m_{T3}=R_g/c_{p\,6-5}$ are the efficiency and index of polytrope of conditional process of gas expansion from the point of outlet from the regenerator to the outlet form the last turbine stage (defined similarly to η_{T2} and m_{T2}).

For the cycle without regeneration:

$$N'_{T3} = G'_4 \cdot c_{p\,6-4} \cdot T_4 \cdot \left(1 - \pi_{T3}^{-m_{T2}} \right) \cdot \eta_{T3}, \qquad (25)$$

From the expression of thermal balance of regenerator (10), considering (11)

$$T_5 = T_4 - \varepsilon \cdot Q_{max} / W_{R\,h}. \qquad (26)$$

We can define temperature at the point of introduction of regeneration as temperature at the end of the conditional expansion process after the last CC and obtain either from T-s diagram, or as

$$T_4 = T_3 \cdot \left[1 - \left(1 - \pi'^{\,-m_{T2}}_{T2} \right) \cdot \eta_{T2} \right]. \qquad (27)$$

After substitution in (24) of value η_{T3} from (25) and appropriate transformations, magnitude N_{T3} can be represented as

$$N_{T3} = N'_{T3} - \Delta N_{R\varepsilon} - \Delta N_{Rg3} - \Delta N_{R\varepsilon g}, \qquad (28)$$

where

$$\Delta N_{R\varepsilon} = N'_{T3} \cdot \varepsilon \cdot Q_{max} / \left(W_{R\,h} T_4 \right)$$

– a change in capacity due to temperature reduction in the process of expansion after the regenerator;

$$\Delta N_{Pg3} = N'_{T3} \cdot \left(G'_f - G_f \right) / G'_3$$

– a change in capacity due to consumption reduction of the working medium in the process of expansion after the regenerator;

$$\Delta N_{R\varepsilon g} = N'_{T3} \cdot \varepsilon \cdot Q_{max} / \left(W_{R\,h} T_4 \right) \cdot \left(G'_f - G_f \right) / G'_3$$

– a change in capacity that depends on the unconsidered above influence of changes in temperature and consumption of the working medium in the process of expansion after the regenerator.

After substitution of the obtained expressions for N_{T1}, N_{T2} and N_{T3} in (19) we will receive:

$$N_{T\Sigma} = N'_{T\Sigma} - \Delta N_{R\upsilon} - \Delta N_{R\varepsilon} - \Delta N_{Rg} - \Delta N_{R\upsilon g} - \Delta N_{R\varepsilon g},$$

where $\Delta N_{Rg}=\Delta N_{Rg2}+\Delta N_{Rg2}$ W is the total change in capacity due to the change in consumption of the working medium.

After substitution in (18) we will obtain:

$$N_e = N'_e - \Delta N_{R\upsilon} - \Delta N_{R\varepsilon} - \Delta N_{Rg} - \Delta N_{R\upsilon g} - \Delta N_{R\varepsilon g}.$$

An analysis of calculation results of GTE cycles with heat regeneration at different values T_3, ε, ν_R demonstrated that in the zone of optimal values π_C:

– value $\Delta N_{R\varepsilon}$, which may amount to 8...10 % of cycle efficiency has the most influence on efficiency;

– value $\Delta N_{R\nu}$ may amount to 5 % of cycle effficiency;

– values ΔN_{Rg}, $\Delta N_{R\varepsilon g}$, $\Delta N_{R\nu g}$ do not exceed 1 %, 0,1 % and 0,05 % of cycle efficiency and can be neglected with the accuracy sufficient for practical calculations.

Thus, the magnitude of relative losses of work at the introduction of regeneration

$$\delta N_e = \delta N_{R\upsilon} + \delta N_{R\varepsilon}, \qquad (29)$$

where $\delta N_{R\nu}=\Delta N_{R\nu}/N'_e$, $\delta N_{R\varepsilon}=\Delta N_{R\varepsilon}/N'_e$, are the relative changes in the GTE capacity at the introduction of heat regeneration due to pressure losses in regenerator and temperature reduction in the expansion process after the regenerator, respectively.

Magnitude $\delta N_{R\,\nu}$ can be represented as follows:

$$\delta N_{R\upsilon} = C_\upsilon \cdot \left(\nu_R^{-m_{T2}} - 1 \right), \qquad (30)$$

where

$$C_\upsilon = 1 / \left(\pi'^{\,-m_{T2}}_{T2} - 1 \right) \cdot N'_{T2} / N'_e, \qquad (31)$$

coefficient that characterizes the influence of pressure losses on the capacity of regenerative GTE.

If we use dependence [0]

$$\nu_R^{-m_{T2}} = 1 - m_{T2} \cdot \delta P_{R\Sigma}, \qquad (32)$$

we will receive:

$$\delta N_{R\upsilon} = C_P \cdot \delta P_{R\Sigma}, \qquad (33)$$

where

$$C_P = m_{T2} \cdot C_\upsilon, \qquad (34)$$

$\delta P_{R\Sigma}=\delta P_{Rc}+\delta P_{Rh}$ are the total relative pressure losses in regenerator.

Magnitude $\delta N_{R\varepsilon}$ can be written down as:

$$\delta N_{R\varepsilon}=C_{\varepsilon}\cdot C_Q\cdot\varepsilon, \tag{35}$$

where coefficient

$$C_{\varepsilon}=N'_{T3}/\left(\eta'\cdot W_{R\,h}\cdot T_4\right), \tag{36}$$

connects the heat that is transferred in intermediate regenerator with losses of efficiency of useful work.

After substitution of expressions (30) and (35) in (29), and then in (13), we will obtain:

$$C_{\eta}=\frac{1-\left[C_{\upsilon}\cdot\left(\nu_R^{\,-m_{T2}}-1\right)+C_{\varepsilon}\cdot C_Q\cdot\varepsilon\right]}{1-C_t\cdot C_Q\cdot\varepsilon}=$$

$$=\frac{1-\left[C_P\cdot\delta P_{R\Sigma}+C_{\varepsilon}\cdot C_Q\cdot\varepsilon\right]}{1-C_t\cdot C_Q\cdot\varepsilon}. \tag{37}$$

The obtained dependences (13) and (37) indicate that efficiency of a new regenerative cycle can be expressed through the efficiency of the cycle without regeneration and coefficient C_{η}, which considers the influence of heat regeneration on efficiency. This allows us to calculate the efficiency of regenerative GTE without performing complex and labour-consuming calculations only by two independent magnitudes ε and δP_R, which characterize the processes taking place in the regenerator. Other magnitudes η', N'_e, C_t, C_Q, C_{ε} can be determined from the calculation of the GTE cycle without heat regeneration. The resulting formula is characterized by simplicity and the possibility to study influence of parameters of the regenerator on the GTE efficiency separately from other parameters of the plant elements.

4. 2. Establishing interrelation between regenerator effectiveness, parameters of heat exchange surface and heat carriers

According to [17], efficiency of heat exchange device can be calculated by formula

$$\varepsilon=\frac{2\left(\exp\left(z_t\cdot NTU\right)-1\right)}{\left[z_t+\left(W_r+1\right)\right]\cdot\exp\left(z_t\cdot NTU\right)-\left[z_t-\left(W_r+1\right)\right]}, \tag{38}$$

where

$$z_t=\sqrt{\left(W_r+1\right)^2-4\cdot p_e\cdot W_r}, \tag{39}$$

p_e is the index of countercurrent flow of the scheme, which takes into account the efficiency of heat transfer, $W_r=W_{min}/W_{max}$ is the ratio of the minimum and maximum water equivalents of heat carriers, which move along the channels of regenerator, NTU is the number of units of heat transfer, which is calculated by formula:

$$NTU=k\cdot F/W_{min}, \tag{40}$$

where k, Wt/(m^2 K) is the coefficient of heat transfer, F, m^2 is the area of heat exchange surface.

The advantage of formula (38) over other similar dependences [29] is that its structure is identical for the different variants of flows of heat carrier. However, within ranges, characteristic for modern regenerators (at NTU>2 and $W_r\to1$), the error of calculating ε using it significantly increases (up to 6 % compared with the values given in [29]). In this case, for six-pass regenerator of GTE at W_r=0,98, the error of 1 % in determining ε will correspond to the error of defining the area of heat transfer surface of 9 % (at ε=0,85) and 26 % (with ε=0,9), which led to the need to refine dependence (38).

After analyzing results of the calculation of efficiency of heat exchangers with different flow schemes, it was found that magnitude p_e can be represented in the form of quadratic dependence on W_r which allowed us to record values of z_t in the form

$$z_t=\sqrt{p_{e1}\cdot W_r^{\,3}+p_{e2}\cdot W_r^{\,2}+p_{e3}\cdot W_r+1}, \tag{41}$$

and formula (38) in the form:

$$\varepsilon=\frac{2\left(\exp\left(z_t\cdot NTU\right)-1\right)}{\left[z_t+\left(p_{e4}\cdot W_r+1\right)\right]\cdot\exp\left(z_t\cdot NTU\right)-\left[z_t-\left(p_{e4}\cdot W_r+1\right)\right]}, \tag{42}$$

where $p_{e1\ldots4}$ are the coefficients that depend on the flow scheme of heat carriers.

The introduction of dependences (41), (42) allowed decreasing the maximum error of calculation ε to 2 %, which quickly decreases with increasing the number of passes. We also obtained coefficients for the calculation of multi-pass scheme where both heat carriers are not mixed, for which the data are missing in [17].

The product of the area of heat transfer surface and coefficient of heat transfer that is included in (40) can be represented in the form [17]:

$$kF=\left(kF\right)_{Wmin}=\left(kF\right)_{Wmax}=k_F\cdot\left(\left(\alpha\cdot F\right)_{Wmin}^{-1}+\left(\alpha\cdot F\right)_{Wmax}^{-1}\right)^{-1}, \tag{43}$$

where α, Wt/(m^{2}K) is the coefficient of heat transfer between the heat carrier and the wall from the side of correspondent heat carrier; k_F is the coefficient of surface reserve.

Indices "Wmin" and "Wmax" refer to the heat carrier with minimum and maximum water equivalents, respectively.

Heat transfer coefficients for cold and hot heat carriers are possible to calculate from the known criterion equations [16, 17, 29]. Most often it is considered that, in the channels of regenerators of GTE, turbulent motion of gas heat carriers is observed, for which criterion equations of heat transfer are presented in the form:

$$Nu=A\cdot Re^a\cdot Pr^b\cdot\varphi_l\cdot\varphi_t, \tag{44}$$

where $Nu=\alpha l/\lambda$ is the Nusselt criterion; $Re=wl/\upsilon$ is the Reynolds criterion; $Pr=\mu\times c_p/\lambda$ is the Prandtl number; l, m is the characteristic size of the heat transfer surface from the side of a corresponding heat carrier, by which the appropriate criterion is calculated; w, m/s is the characteristic flow rate; λ, Wt/(mK) is the thermal conductivity of the heat carrier; μ, Pa×s is the dynamic viscosity of heat carrier; φ_l is the coefficient, which takes into account geometrical parameters of heat transfer matrix; φ_t is the coefficient that takes into account the change in the properties of heat carrier due to differences in temperatures between the wall of channel and flow nucleus (usually represented in the form $(Pr/Pr_{wall})^{0,25}$), for heat carriers used in GTE we can accept φ_t=1; A, a, b are the coefficients, which, depending on formula's form, can be constant values or be determined through geometrical parameters of the heat exchange surface.

Considering (44), as well as determining the Nusselt, Reynolds and Prandtl numbers, the product of coefficient of heat transfer and the surface area of heat exchange for each heat carrier can be represented as follows:

$$\alpha \cdot F = (S/W)^{-a}\left(A \cdot F \cdot l^{-(1-a)} \cdot \varphi_l\right) \cdot \left(\varphi_t \cdot \lambda^{(1-a)} Pr^{(1-a)}\right). \quad (45)$$

Thus, taking into account (43) and (44), NTU can be represented as follows:

$$NTU = k_F\left(\left(s_q^{-a}\left(A \cdot f_q \cdot l^{-(1-a)}\right) \cdot \left(\varphi_t \lambda^{(1-a)} Pr^{-(a-b)}\right)\right)_{W\min}^{-1} + \right.$$
$$\left. + W_r \cdot \left(s_q^{-a}\left(A \cdot f_q \cdot l^{-(1-a)}\right) \cdot \left(\varphi_t \lambda^{(1-a)} Pr^{-(a-b)}\right)\right)_{W\max}^{-1}\right)^{-1}, \quad (46)$$

where s_q=S/W and f_q=F/W, $m^2/(Wt/K)$ are the areas for passage of heat carrier and heat exchange surface, referred to water equivalent, or to the heat amount that is necessary to transmit in the regenerator for heating the heat carrier by 1 degree.

Using dependences [29], which connect the area of heat exchange surface and the volume of matrix (V_M)

$$F_c / \psi_c = F_h / \psi_h = V_M, \quad (47)$$

and the volume of the matrix with its mass (M_M) and the mass of the heat exchanger (M_{HE})

$$M_{HE} = k_M M_M = k_M \rho_M V_M, \quad (48)$$

expression (46) for calculating NTU can be presented as

$$NTU = \frac{k_F m_q}{k_M \rho_M}\left(\left(s_q^{-a}\left(A \cdot \psi \cdot l^{-(1-a)} \cdot \varphi_l\right)\left(\varphi_t \cdot \lambda^{(1-a)} Pr^{-(a-b)}\right)\right)_{W\min}^{-1} + \right.$$
$$\left. + \left(s_q^{-a}\left(A \cdot \psi \cdot l^{-(1-a)} \cdot \varphi_l\right)\left(\varphi_t \cdot \lambda^{(1-a)} Pr^{-(a-b)}\right)\right)_{W\max}^{-1}\right)^{-1}, \quad (49)$$

where $m_q = M_{HE}/W_{min}$, kg/(Wt/K) is the specific mass of the heat exchange surface ψ, m^2/m^3 is the coefficient of compactness of the heat exchange surface on the side of corresponding heat carrier, $\rho_M = \rho_{st} V_{st}/V_M$, kg/m^3 is the conditional density of heat exchange matrix, which depends on the geometry of channels and density of material matrix, V_{st}, m^3 is the volume of matrix except the volume of channels for passage of heat carriers, ρ_{st}, kg/m^3 is the density of material of which heat-exchange matrix is made, k_M is the ratio between the mass of matrix and the mass of the entire heat exchanger (takes into account a lot of additional structural elements), which may be defined based on the characteristics of prototypes of regenerators, similar in design, or by results of constructive works.

After substitution to (42) and (49) and transformations, we will obtain

$$m_q = \frac{k_M \rho_M}{z_t k_F} \cdot \ln\left(\frac{2 + \varepsilon\left[z_t - \left(p_{e4} W_r + 1\right)\right]}{2 - \varepsilon\left[z_t + \left(p_{e4} W_r + 1\right)\right]}\right) \times$$
$$\times\left(\left(s_q^{-a}\left(A \cdot \psi \cdot l^{-(1-a)} \cdot \varphi_l\right)\left(\varphi_t \cdot \lambda^{(1-a)} Pr^{-(a-b)}\right)\right)_{W\min}^{-1} + \right.$$
$$\left. + \left(s_q^{-a}\left(A \cdot \psi \cdot l^{-(1-a)} \cdot \varphi_l\right)\left(\varphi_t \cdot \lambda^{(1-a)} Pr^{-(a-b)}\right)\right)_{W\max}^{-1}\right). \quad (50)$$

Thus, the resulting dependence that connects the mass of regenerator and its efficiency, parameters of heat carriers and geometric parameters, was obtained.

4. 3. Determining the dependence of pressure losses in regenerators on the parameters of heat exchange surface and heat carriers

The total change in static pressure of heat carrier ΔP when passing through the heat exchanging device is usually divided into [17] friction resistance ΔP_{fr}, local resistances ΔP_{loc}, hydrostatic resistance ΔP_h and pressure losses caused by the change in dynamic pressure, ΔP_d:

$$\Delta P = \sum_{i=1}^{n_{fr}} \Delta P_{fr\,i} + \sum_{j=1}^{n_{loc}} \Delta P_{loc\,j} + \Delta P_h + \Delta P_d, \quad (51)$$

where n_{fr} is the number of sections by the progress of heat carrier, where pressure losses are connected with friction, n_{loc} is the number of local resistances on the progress of heat carrier.

For each section of the tract of heat exchanger, pressure losses from friction and local resistances[17] are defined as:

$$\Delta P_{fr} = C_{L/d} \xi \frac{\rho w^2}{2}, \quad (52)$$

$$\Delta P_{loc} = \zeta \frac{\rho w^2}{2}, \quad (53)$$

where ξ is the coefficient of resistance of friction, ζ is the coefficient of local resistance, $C_{L/d}$ is the coefficient, which depends on the ratio of channel length L to its hydraulic diameter d_g.

Coefficient of resistance of friction is usually determined by dependences of the form:

$$\xi = C \cdot Re^{-p}, \quad (54)$$

where C, p are the magnitudes that, depending on the channel type, may be permanent, or depend on its geometrical parameters.

Hydrostatic resistance and losses related to the change in dynamic pressure are calculated for the entire heat exchanger [17]:

$$\Delta P_h = \rho_{mean} \cdot g \cdot H, \quad (55)$$

$$\Delta P_d = \frac{\rho_{in} w_{in}^2}{2} - \frac{\rho_{out} w_{out}^2}{2}, \quad (56)$$

where H is the height of the heat carrier rise in regenerator.

Indices "in" and "out" correspond to the parameters of heat carrier at the inlet and outlet of regenerator; "mean" are the mean parameters of heat carrier in regenerator. The values of rate and density of flow are calculated for each element of tract of regenerator by the parameters of heat carrier at its inlet.

Performed calculations of regenerators of different structures demonstrated that magnitudes ΔP_h and ΔP_d, due to low density of heat carrier, typically do not exceed 5 % of the total loss, so to simplify the calculations, they can be neglected.

The rate of heat carrier in the matrix channels of regenerator can be determined from equation of consumption:

$$w_M = G / \left(\rho_{mean} S\right). \quad (57)$$

At the stage of preliminary designing, to determine pressure losses in regenerator, mean density of heat carrier is used, which is defined from equation of state [17]:

$$\rho_{mean} = P_{mean} / \left(R \cdot T_{mean}\right), \quad (58)$$

where P_{mean}, Pa and T_{mean}, K are the mean pressure and heat carrier temperature in regenerator.

Mean pressure of heat carrier in regenerator can be represented in the form:

$$P_{mean} = P_{in} - \Delta P / 2. \quad (59)$$

Magnitude of coefficient $C_{L/d}$ depends on the geometry of the channel [14, 17, 29]: in motion inside plain pipes $C_{L/d}$= =L/d, at flowing over chess beams of plain pipes $C_{L/d}$= =5,4+3,4×L/(ds$_2$).

The pass length of heat carrier in the heat exchange matrix (L), which is included in $C_{L/d}$, may significantly differ from the pass length (L_M) in the active part of the matrix that takes part in heat exchange. Thus, in the heat exchange matrix there are return bends or passages, along which heat carrier transfers from one passage to another, bends, by which it is brought to collectors or is removed from them. These elements do not take part in heat transfer, however, they increase the length of motion of heat carrier in the matrix and resistance that can be considered by introducing magnitude k_L=L/L_M.

Using dependence [29]

$$d_g = 4 \cdot L_M \cdot S / F, \quad (60)$$

magnitude $C_{L/d}$=$C_{L/d}$(L/d_g) can be represented as $C_{L/d}$= =$C_{L/d}$($k_L L_M/d_g$)=$C_{L/d}$(k_L/4×F/S).

After substitution (52)–(59) to (51) and transformations, we will receive dependence that connects relative pressure losses of the regenerator with parameters of heat carriers and geometry of the matrix:

$$\delta P = 1 - \sqrt{1 - \left[C \cdot C_{L/d}\left(\frac{k_L}{4}\frac{\psi}{k_M \rho_M}\frac{m_q}{s_q}\right)^{1-p} \cdot \left(\lambda Pr\right)^p \cdot s_q^{-p} + \zeta_\Sigma\right] \cdot s_q^2 \cdot \frac{R}{c_p^2} \cdot \frac{T_{mean}}{P_{in}}}, \quad (61)$$

where $\zeta_\Sigma=\Sigma\zeta_j\times(S/S_j)$ is the sum of coefficients of local resistances (ζ_j), reduced to the rate of heat carrier in the channels of matrix, S and S_j, m^2, is the area for passing heat carrier in channels of heat-exchange matrix and in certain element of the tract of regenerator.

Expressions (50), (61), written down for each heat carrier, form a system of transcendental equations, after solution of which by assigned effectiveness and pressure losses with the use of a number of methods, it is possible to calculate the mass of regenerator. Taking into account recommendations, worked out in [18, 19], with the help of the given dependences, it is possible to find such values of geometrical characteristics of the heat exchange surface, which would ensure the minimum mass of regenerator.

5. Selection of rational parameters of GTE regenerators

5. 1. Selection of rational values of effectiveness of GTE regenerator and pressure losses

Obtained dependences (37), (50) and (61) describe relationships between effectiveness, pressure losses, mass of regenerator and efficiency of GTE by mathematical functions of different types, as a result of which the same change in ε and $\delta P_{R\Sigma}$ differently affects m_q, and C_η (Fig. 2).

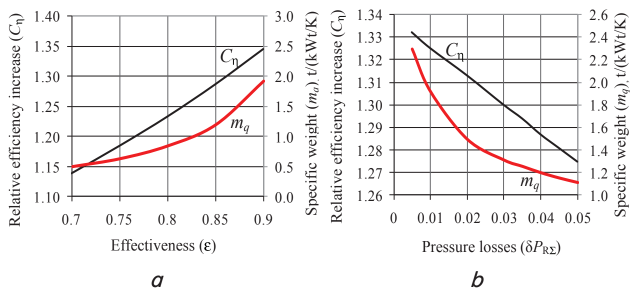


Fig. 2. Dependence of relative increase in efficiency (C_η) and specific mass of regenerator (m_q): *a* — effectiveness; *b* — pressure losses

Calculations are performed by parameters of the plant GTU-16R, designed by the state enterprise "Gas Turbine Scientific Production Complex "Zorya"–"Mashproekt" (Mykolayiv, Ukraine) [10, 12, 13]. Regenerator (Fig. 3) consists of two parts, which are installed in parallel. The scheme of motion of heat carriers in regenerator is a multiple cross-flow with a common counterflow. Heat transfer surface 1 is made of flat tubular beams, executed in the form of coilpipes, placed in a rectangular case, which is a part of gas duct of GTE. The coilpipes are constructed from thin-walled pipes of length L_M with external diameter d_{ex} and wall thickness δ_{wall}, placed in a checkerboard pattern (cross step s_1, longitudinal s_2). Exhaust gases flow over tubes from outside, the air makes several passes (n_p) inside the tubes and is supplied (removed) by round collectors. Displacers are installed between the passes along the perimeter of the gas duct. The tubes are connected to the collectors by bents. The passage of air from one pass to another is made by return bends. Design features of regenerator are declared in the patent for the utility model UA 78601 U.

Based on Fig. 2, it is possible to make an assumption that by choosing values ε and $\delta P_{R\Sigma}$ at the same efficiency of the plant, it is possible to obtain several variants of regenerators, from which the one that has the minimum mass may be chosen.

The relationship between ε and $\delta P_{R\Sigma}$, which provide for constant efficiency, may be obtained from (37):

$$\delta P_{R\Sigma} = \frac{1 - C_\eta + \left[C_\eta \cdot C_t - C_\varepsilon\right] C_Q \cdot \varepsilon}{C_P}. \quad (62)$$

If for the GTE of simple cycle at t_3=1000 °C we accept π_C=6 as calculated value, then C_η will be equal to 1,7. Next, by formula (62), we define a set of pairs of values ε and $\delta P_{R\Sigma}$ at constant C_η (Fig. 4).

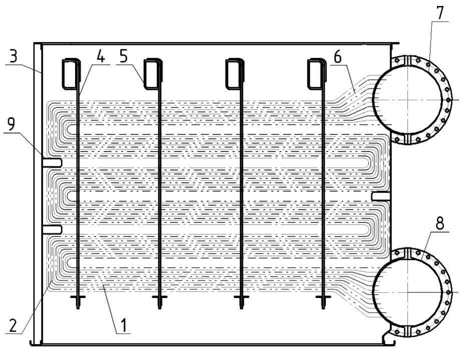


Fig. 3. Section of regenerator for GTU-16R (SE GTSPC "Zorya"—"Mashproekt": 1 — heat exchange package of flat coilpipes; 2 — return bends; 3 — case; 4 — distance spacer; 5 — beams; 6 — bends; 7 — distributing air collector; 8 — collecting air collector; 9 — displacer

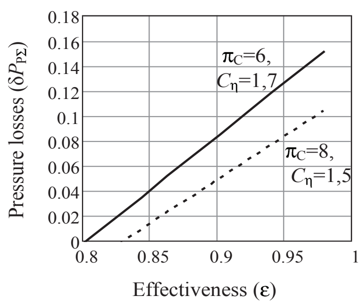


Fig. 4. Relationship between ε and $\delta P_{R\Sigma}$ at constant value C_η

Calculations of coil plain-pipe regenerator demonstrated that the minimum mass will be achieved at ε=0,844 and $\delta P_{R\Sigma}$=0,034 (Fig. 5). For π_C=8 and C_η=1,5, minimum mass will be achieved at ε=0,864 and $\delta P_{R\Sigma}$=0,024.

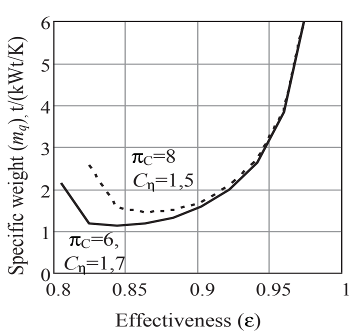
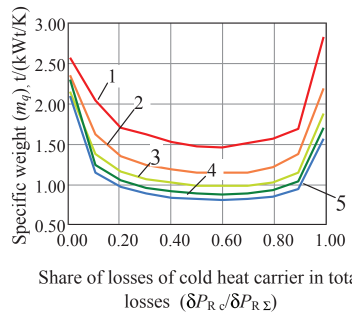


Fig. 5. Change in the mass of regenerator from ε (values $\delta P_{R\Sigma}$ are taken from Fig. 4)

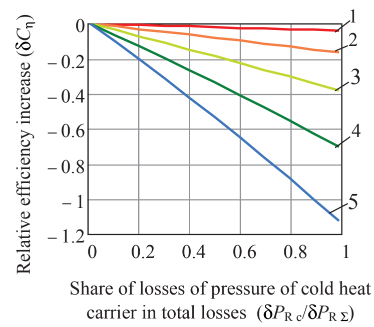
Thus, due to the rational choice of ε and $\delta P_{R\Sigma,}$ it is possible to achieve a decrease in the mass of regenerator at constant efficiency.

Depending on the chosen regenerator structure at the same values of total pressure losses, the change in ratio $\delta P_{Rc}/\delta P_{R\Sigma}$ (redistribution of total losses of pressure between the hot and cold sides of the regenerator) can lead to substantial (by 3 times) change in the mass of the heat exchanging device (Fig. 6).

It follows from (15) and (37) that at constant value of $\delta P_{Rc}/\delta P_{R\Sigma}$, a change in ratio between δP_{Rc} and δP_{Rh} will lead to changes in efficiency of the cycle. For the cycle with simple regeneration (Fig. 6, *a*), an increase in the share of losses from the cold side ($\delta P_{Rc}/\delta P_{R\Sigma}$) decreases C_η by 0,1 to 0,12 % (relative).



a



b

Fig. 6. Influence of change in ratio between pressure losses of heat carrier on the parameters of regenerator and GTE (t_3=1000 °C, π_C=4 and ε=0,85): *a* — is the mass of regenerator; *b* — is the relative increase in efficiency; 1 — $\delta P_{R\Sigma}$=2 %; 2 — $\delta P_{R\Sigma}$=4 %; 3 — $\delta P_{R\Sigma}$=2 %; 4 — $\delta P_{R\Sigma}$=2 %; 5 — $\delta P_{R\Sigma}$=10 %

In a wide range of change in $\delta P_{Rc}/\delta P_{R\Sigma}$ (under these conditions at $\delta P_{Rc}/\delta P_{R\Sigma}$=0,4...0,8), the mass of regenerator is only slightly different from the minimum, which makes it possible to select such value of $\delta P_{Rc}/\delta P_{R\Sigma}$ (under given conditions $\delta P_{Rc}/\delta P_{R\Sigma}$=0,4), which with a slight increase in mass will lead to a small decrease in efficiency (in this case, to 0,4 % relative).

Thus, due to the rational distribution of the value of total pressure losses in the regenerator between the hot and cold sides, it is possible to ensure a substantial decrease in its mass at almost constant efficiency.

5. 2. Development of the algorithm for choosing rational parameters of GTE regenerators

Based on the findings obtained in chapter 5.1, we propose the following procedure for finding rational parameters of the regenerator and GTE, which ensure minimum mass of regenerator at the asguired level of efficiency (Fig. 7):

– the cycle without regeneration is calculated and N'=f(π_C), η'=f(π_C) are determined (Fig. 7, *a*);

– by the obtained values of η' and assigned value of η_{asg}, magnitudes $C_{\eta\,asg}=\eta_{asg}/\eta'$ are calculated (Fig. 7, *b*);

– first for each t_3, the lower limit of change in efficiency of regenerator (ε_{min}) is established, for this we accept $\delta P_{R\Sigma}$=0, and for each π_C by formila (62), minimum values of ε_{min} are defined, which ensure the obtained values of $C_{\eta\ asg}$ (Fig. 7, *c*). From the diagram it is seen that there are such values of π_C, at which it is impossible to achieve the assigned efficiency ($\varepsilon_{min} \geq 1$);

– for each value of t_3 in the range from ε_{min} to 1, some values of ε are accepted and the values of $\delta P_{R\Sigma}$, correspondent to them, are determined by formula (62), which ensure obtained values of $C_{\eta\ asg}$ with selected value of ε (Fig. 7, *d*, the results are given only for t_3=1000 °C);

– since (Fig. 2), with an increase in $\delta P_{R\Sigma}$ the mass of regenerator decreases, the smallest mass (at other equal parameters) will be obtained in the point of maximum value of total pressure losses $(\delta P_{R\Sigma})_{max}$. Based on this, for each value ε, according to Fig. 7, *d*, we determine $(\delta P_{R\Sigma})_{max}$ and the value of $\pi_{C\ opt}$ correspondent to it;

– for each group of the obtained values of ε, $(\delta P_{R\Sigma})_{max}$ and $\pi_{C\ opt}$, we performed the calculation of regenerator with simultaneous choice of values δP_{Rc} and δP_{Rh}, which ensure the minimum mass of regenerator at a constant value of δP_{Rc} and δP_{Rh}. The choice of rational geometrical parameters of the heat exchange surface in order to obtain the minimum mass of the heat exchanger is shown in [18, 19];

– the obtained structural variants of regenerators are compared, the variant with a minimum mass of the heat exchanger is selected, and the corresponding values of ε, $\delta P_{R\Sigma}$, δP_{Rc} and δP_{Rh}, $\pi_{C\ opt}$ and geometrical parameters are considered the ones that ensure minimum cost of the heat exchanger at the given value of efficiency.

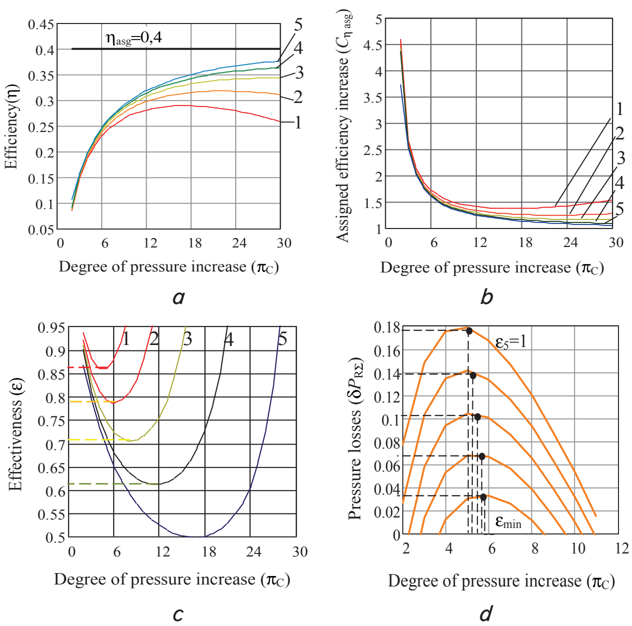


Fig. 7. Results of calculations when searching for rational values of parameters of the regenerator of GTE: *a* — efficiency of cycle without regeneration; *b* — assigned efficiency increase $C_{\eta\ asg}$; *c* — minimum effectiveness of regenerator ε_{min}; *d* — pressure losses $\delta P_{R\Sigma}$ (for t_3=1000 °C, η_{asg}=0,4); 1 — t_3=900 °C; 2 — t_3=1000 °C; 3 — t_3=1100 °C; 4 — t_3=1200 °C; 5 — t_3=1300 °C

Thus, the proposed method for determining parameters of regenerator of GTE allows ensuring minimum mass of the heat exchanger and reaching the specified level of efficiency at the initial stages of designing.

6. Discussion of results of rational parameters of regenerators of GTE of different structural solutions

Consider the use of developed dependences and algorithm for searching for rational parameters of GTE of different structural design with plain-pipe regenerator, the design of which is shown in Fig. 3. The calculations are performed for GTE with regeneration (R), with intermediate cooling and regeneration (R and IC), with intermediate heating and regeneration (R and IH), with intermediate cooling, overheating and regeneration (R, IC and IH). Results of the calculation of specific mass and effectiveness of regenerator depending on the assigned engine efficiency are given in Fig. 8, 9.

The mass of regenerator with an increase in efficiency increases by curve, close to the exponent, and the rate of this increase dramatically grows after ε=0,8...0,9 (with an increase in temperature, zone of abrupt change in the mass shifts to the area of high values of ε). These values can be considered to be the limits for ε when using the plain-pipe heat exchangers.

Using the given graphs for GTE of different structural solutions, it is possible to determine the value of initial temperature of gases and the regenerator effectiveness, which are necessary to achieve the assigned level of efficiency with moderate mass of the regenerator. For example, for GTE with regeneration capacity of 16 MW at the limitation of regenerator mass (M_{HE}) by value of 100 t

– it is possible to achieve 40 % efficiency at t_3=1000 °C and ε=0,84;

– it is possible to achieve 45 % efficiency at t_3=1150 °C and ε=0,9;

– it is possible to achieve 50 % efficiency at t_3=1300 °C and ε=0,9.

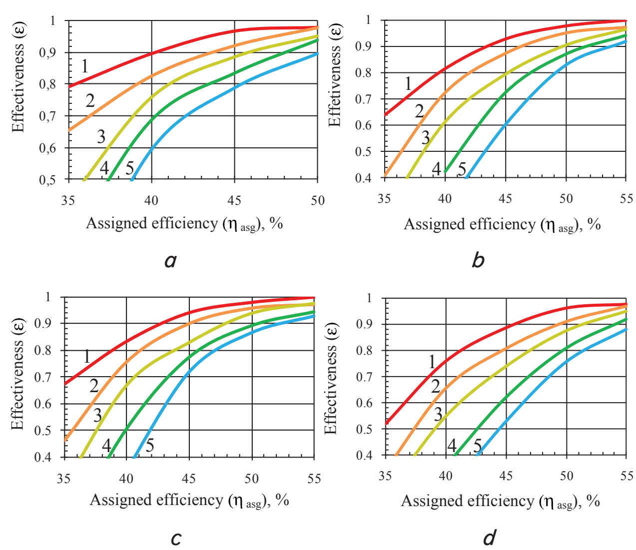


Fig. 8. Effectiveness of regenerator, which ensures assigned efficiency: *a* — GTE with R; *b* — GTE with R and IC; *c* — GTE with R and IH; d — GTE with R, IC and IH; 1 — t_3=900 °C; 2 — t_3=1000 °C; 3 — t_3=1100 °C; 4 — t_3=1200 °C; 5 — t_3=1300 °C

The range of the recommended values of total pressure losses in regenerator at effectiveness ε=0,5...0,95 is 0,03...0,08. Thus, the level of economically viable pressure losses, given in [14], which for degree of heat regeneration r=0,6–0,8 was 0,03–0,07, was extended. The range of recommended values of magnitude $\delta P_{Rc}/\delta P_{R\Sigma}$ is 0,3...0,8.

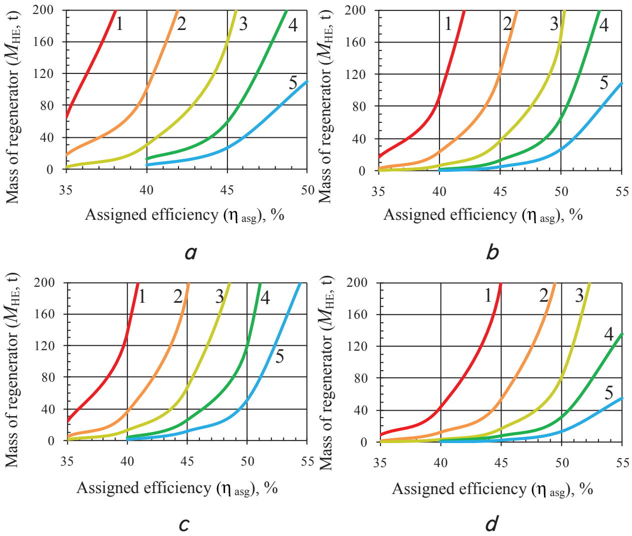


Fig. 9. Minimum mass of regenerator (M_{HE}): *a* – GTE with R; *b* – GTE with R and IC; *c* – GTE with R and IH; *d* – GTE with R, IC and IH; 1 – t_3=900 °C; 2 – t_3=1000 °C; 3 – t_3=1100 °C; 4 – t_3=1200 °C; 5 – t_3=1300 °C

With an increase in the initial temperature of gas in the cycle, the degree of regeneration that is needed to achieve the assigned efficiency decreases, which at constant efficiency leads to a substantial decrease in the mass of the regenerator (approximately by 2...3 times for every 100 °C).

At equal values of initial temperature of gases and efficiency:

– introduction of intermediate cooling to the structure of the GTE leads to a decrease in the mass of regenerator by 3–5 times;

– of intermediate overheating – by 1,5...2 times;

– of simultaneous intermediate overheating and cooling – by 8...10 times.

Thus, the use of intermediate cooling is more efficient, because at equal values of efficiency, the mass of the generator of GTE with intermediate cooling will be 1,5...3 times less than that with intermediate overheating.

If efficiency and specific mass of the regenerator is expressed through a relative increase in efficiency ($C_{\eta\ asg}$), the the graphs of dependences $\varepsilon=f(C_{\eta\ asg})$ and $m_q=f(C_{\eta\ asg})$ for the considered cycles and temperatures will be close enough (Fig. 10).

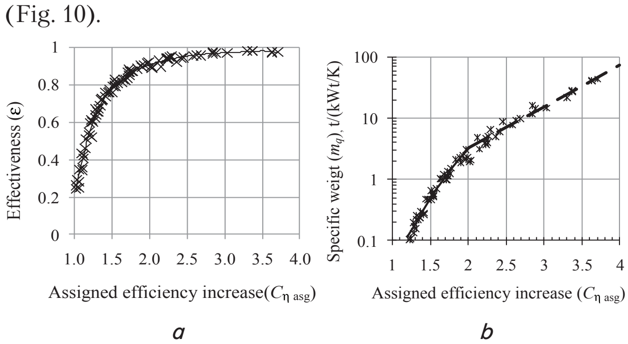


Fig. 10. Dependence of parameters of regenerator on the assigned increase in efficiency ($C_{\eta\ asg}$): *a* – effectiveness of regenerator; *b* – specific mass

Dependence of recommended effectiveness on the assigned relative increase in efficiency (Fig. 10, *a*) can be represented as follows:

$$\varepsilon = -0{,}0414\cdot C^{6}_{\eta\ asg} + 0{,}6516\cdot C^{5}_{\eta\ asg} - 4{,}228\cdot C^{4}_{\eta\ asg} + \\ +14{,}49\cdot C^{3}_{\eta\ asg} - 27{,}75\cdot C^{2}_{\eta\ asg} + 28{,}40\cdot C^{3}_{\eta\ asg} - 11{,}39. \quad (63)$$

Dependence of specific mass on the assigned relative increase in efficiency (Fig. 10, *b*) in the range of values $C_{\eta\ asg}$ from 1,2 to 2,1 can be approximated by formula:

$$m_q = 35{,}0\cdot C^{6,5}_{\eta\ asg}, \quad (64)$$

and at $C_{\eta\ asg}$ larger than 2,1

$$m_q = 140{,}79\cdot \exp\left(1{,}5654\cdot C_{\eta\ asg}\right). \quad (65)$$

The shape of the curves is selected in such a way that would ensure minimum error in the approximation of data, coefficients are calculated by the method of least squares.

The obtained dependences allow us to estimate the mass of plain-pipe regenerator of GTE at the initial stages of design.

7. Conclusions

As a result of conducted studies into the influence of parameters of the regenerator (effectiveness and pressure losses) on the efficiency of GTE and the mass of regenerator, it was found that due to the choice of rational values of effectiveness of the regenerator and pressure losses, it is possible to ensure a significant decrease in the mass of the regenerator at a constant value of efficiency of GTE. Relationship between parameters of regenerator and efficiency of regenerative GTE was described by functional dependence where energy parameters of regenerator are collected in a single complex. It was established that the value of effectiveness of the regenerator is related to efficiency by inversely proportional function, and pressure losses – by linear function. The effectiveness of the regenerator is connected with the mass of the heat exchanger and the geometry of heat exchange surface by exponential dependence, and relative pressure losses – by algebraic irrational function.

Based on these dependences, we developed the algorithm for choosing rational parameters of regenerator at the initial designing stages, the use of which will ensure minimum mass of the regenerator.

The obtained results of calculations of mass of regenerators of GTE with different structural solutions and rational magnitudes of effectiveness of regenerator, pressure losses and initial temperature of gas allow a designer engineer to choose rational values of effectiveness of regenerator, pressure losses and initial temperature of gas at the initial stages of designing regenerative GTE.

Thus, the possibility of obtaining the minimum mass of the regenerator at the assigned efficiency of GTP by means of selecting its rational parameters is demonstrated.

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