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Improvement Performance of Gas Turbine Plants Due to Heat Recovery with a Gas Bypass

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Abstract**—** *Mathematical models and algorithms for the system analysis of a gas turbine plant have been developed, taking into account the efficiency of the air heater and the distribution of local thermo-hydraulic parameters in it. For a stationary gas turbine unit, design options for a tubular regenerator have been developed and analyzed. The maximum effective efficiency of the installation, which can be obtained using a regenerator of the selected design, has been determined.*

Keywords**—** *Air heater, power plant, gas turbine, effective efficienc, ratios of pressure, degree of regeneration.*

I. INTRODUCTION

Recently, much attention of scientists and practitioners in the field of energy has been paid to the use of highly maneuverable units, which are gas turbine plant. Gas turbine plant may be an integral part of modern energy efficient plants: combined cycle and gas-steam installations, cogeneration and trigeneration units, pumping stations. However, the efficiency of stationary gas turbines with a simple thermal circuit based on the Brighton cycle (without improvements) is not high, and amounts to 24% at low ratios of pressure increase ($\pi_k \leq 7$); at higher degrees of pressure increase (π_k = (10-22) – (28-32)%. Another feature of stationary gas turbines is the relatively low temperature of gases in front of the turbine, usually 700 ÷ 900 °C, due to the use of cheaper and less heat-resistant materials, which causes their lower efficiency compared to gas turbine aircraft. It is generally known that in order to increase the efficiency of such plants, it is advisable to use heat recovery (especially at low degrees of pressure increase). For its implementation, heat exchangers, regenerators and air heaters are used. With their use, the thermal load on the environment is also reduced, and the emissions of harmful exhaust gases are reduced. Thus, the development of new and improvement of existing gas turbine plants by introducing or increasing the use of heat recovery is an urgent task of increasing the energy efficiency of the entire energy industry.

II. PROBLEM STATEMENT

The thermal efficiency of air heaters in the power plants is determined by a dimensionless temperature parameter, the degree of regeneration σ. With an increase in the degree of regeneration, the thermal efficiency of the power plants is unambiguously increased. However, in this case, it is necessary to increase the heat exchange surface or intensify the heat transfer process in the apparatus, which will undoubtedly lead to an increase in the cost of the heat exchanger. On the other hand, the aerodynamic resistance increases, which the heat exchanger introduces into power plant path. To overcome the air resistance, a part of the useful power generated by the turbine is consumed, and the resistances along the exhaust path reduce this power. All these factors will affect the decrease in both the effective power of the power plant and the efficiency, which, as a result, may even be lower than in the basic installation without a regenerator. Another problem in the use of regeneration is the low operational reliability of air heaters. Typical surface cracking during operation is caused by high temperatures of heat transfer fluid, heat resistance of materials, thermal stresses, variable loads. These processes are typical for both conventional tubular air heaters and high-performance and compact plate apparatus. Therefore, in this work, the task is posed to improve the efficiency of a stationary power gas turbine plant, taking into account the modeling of thermo-hydraulic processes in the regenerator-air heater and analysis of their influence on the efficiency of the plant as a whole.

III. RESEARCH METHODOLOGY

Air heaters for gas turbine units have a wide range of designs and layouts. Heat exchangers can be both conventional tubular and more efficient - tubular-finned, plate-type, plate-finned, etc. The main requirement for air heaters for transport gas turbines is to ensure the specified efficiency with high compactness and minimum weight and dimensions. In stationary power gas turbines, tubular heat exchangers with cross-flow or mixed flow of coolants are widely used, which are composed of smooth pipes or highly efficient finned tubes. Such devices have low compactness, large weight and dimensions, but are cheaper and have less aerodynamic resistance compared to others. As an object of research in this work, a stationary power gas turbine of the GT 35 Hz brand is considered, having the following nominal parameters [1]: As an object of research in this work, a stationary power gas turbine of the GT 35 Hz brand is considered, having the following nominal parameters [2]: electric power 32 MW; Efficiency 23.2%; the degree of pressure increase $\pi\kappa$ = 6.5; air consumption 215 kg / s; power shaft rotation speed 3000 rpm; gas temperature in front of the turbine 780 ^{o}C; exhaust gas temperature 430 ^{o}C; there is no heat recovery. For the analysis, standard environmental parameters were taken, as well as a pressure loss at the inlet and outlet of 1500 Pa. As can be seen from the characteristics of this power plant, it has all the prerequisites for the introduction of regeneration, namely: high temperature of exhaust gases, low $\pi\kappa$ and, most importantly, low efficiency (23.2%). Consider as a regenerator the simplest, cheapest and affordable option - a tubular heat exchanger.

In tubular regenerators, air is often passed through tubes, and gases flow around the tubes from the outside [3]. This provides the following advantages [4]: the body of the regenerator comes out much lighter, since it is calculated for the pressure of the flue gases, which is close in magnitude to the atmospheric pressure; the heating surface on the gas side is easier to clean from carbon deposits and soot. However, for reasons of reducing aerodynamic resistance, increasing the strength and ease of cleaning the surface, tubular regenerators are also widely used, where hot gases are inside the pipes, and air is outside [5]. In regenerators, the body, tube sheets and tubes themselves are often made of carbon or stainless steel [6]. But for tubes, alloys are often used, for example, aluminum bronze [7]. To intensify heat transfer in tubular regenerators, external and also internal finning and turbulators are sometimes made [8,9]. For the regenerator under consideration, smooth pipes made of steel 20 were selected as the heat exchange surface. To prevent high-temperature corrosion, increase heat resistance and facilitate cleaning, the diameter of the pipes was increased to 57 mm with a wall thickness of 3.5 mm and a roughness of 0.06 mm. The split of the beam is staggered, along an equilateral triangle with a split step of 80 mm. The numbers of pipes and the formation of the size of the bundle were determined based on the recommended velocities of air (10–20 m/s) and gases (15–40 m/s) [10]. The maximum velocities were taken: for air-15 m/s, for gases-40 m/s. The analysis was carried out for fundamentally different, but as mentioned earlier, encountered two variants of such heat exchangers [11]:

1) the gases outside wash the bundle of pipes, making several passes, the air moves inside the pipes;
2) the gases move inside the pipes, the outside air washes the bundle of pipes, making several passes.

The layout options differ significantly, since the air density in the studied installation exceeds the density of gases by more than 8 times, and the mass flow rates of the media are close to each other. For the option where the gases are inside the pipes, it is accepted:

the number of pipes in a row - 104 pcs.,
the number of rows of pipes along the air flow - 50 pcs.,
the number of pipes in one pass - 5200 pcs.,
the length of pipes in the course is 1.75 m;
outer surface area - 1637.4 m^2.

For the option where gases outside the pipes are accepted:
number of pipes in a row - 131 pcs.,
number of rows of pipes in the air flow - 16 pcs.,
the number of pipes in one pass - 2096 pcs.,
the length of pipes in the course is 3.5 m;
the area of the outer surface is 1318.6 m^2.

For the analysis, two mathematical models and algorithms were formed, which are directly related to each other:

1) Calculation of the parameters and efficiency of the gas turbine unit;
2) Calculation of the parameters and efficiency of the air heater, taking into account the distribution of local thermo-hydraulic parameters and operating conditions.

The mathematical model and algorithm for calculating the regenerator provides for the breakdown of the heat exchanger into discrete elements (micro heat exchangers) [12, 13]. The number of breakdown elements on the length of pipes of one stroke is taken equal to 10 (which provides sufficient calculation accuracy [14, 15]). The properties of heat transfer fluid; the parameters of heat transfer and heat transfer [16] in each micro-heat exchanger are different and depend on the features of the apparatus layout, initial sections and contamination. The flow rates and velocities of the medium inside each row of pipes were determined by a specially developed hydraulic calculation algorithm using graph theory, where hydraulic and local resistances were taken into account [17].

IV. Results and Discussion

The results of the analysis of the dependence of the degree of regeneration and the efficiency of the gas turbine plant on the number of strokes in the regenerator are shown in Figure one. As can be seen from Figure 1 for the selected design parameters of the strokes (sections), the efficiency of the regenerators, i.e. the degree of regeneration does not differ. The effective efficiency of a gas turbine unit first increases due to an increase in the degree of regeneration, and then decreases due to an increase in pressure losses in the regenerator. Moreover, the maximum efficiency in the variant where the gases are outside was 27.83% in a four-section apparatus with an area of 5274.7 m^2, in the variant where the gases are inside the pipes was 29.08% in a six-section apparatus with an area of 9824.2 m^2. Figure 2 shows a similar dependence of the degree of regeneration and the effective efficiency of the gas turbine unit, but already on the total surface area of the apparatus. Figure 2, it can be seen that the option where the gases are outside is more efficient in terms of heat transfer, and the option where the gases are inside is better in terms of thermodynamic efficiency. This is explained by the more significant influence of the absolute values of pressure losses at the exhaust on the efficiency of the gas turbine cycle in comparison with losses after the compressor. The next stage of the study was to determine the effect of hot gases bypassing past the regenerator. At the same time, the pressure loss along the exhaust path of the power plant decreases, which has a positive effect on its efficiency, however, the efficiency of the air heater itself (the degree of regeneration) also decreases, which will reduce the efficiency of the power plant. Thus, the present optimization problem is done. Figure 3 shows the results of the study for the basic (selected) and other options for the layout of the regenerator. As can be seen from the analysis results, in the base case, where the gases are inside the pipes, the maximum efficiency of the installation coincides with the bypass fraction $\varphi = 0$ (i.e., there is no optimum). In variants where gases are outside the pipes, the maximum values are present with a bypass fraction $\varphi = 0.2\text{-}0.3$; moreover, the highest efficiency of the installation is 28.0% in the five-way version. It should be noted that this increase in efficiency is negligible. Further, the influence of the layout on the resulting dangerous wall temperature differences (at the junction of the passages at one

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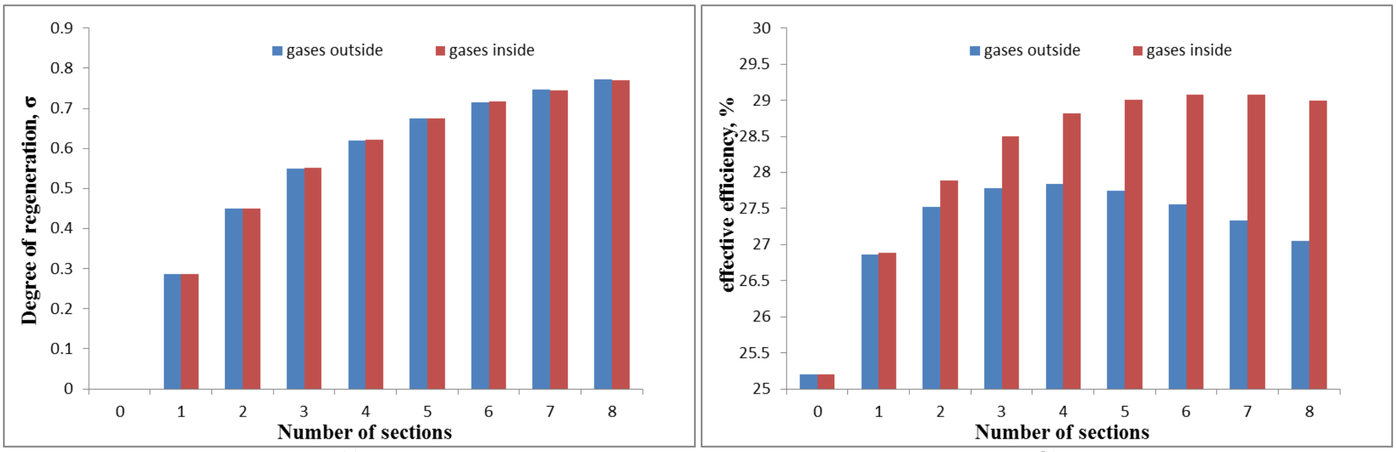
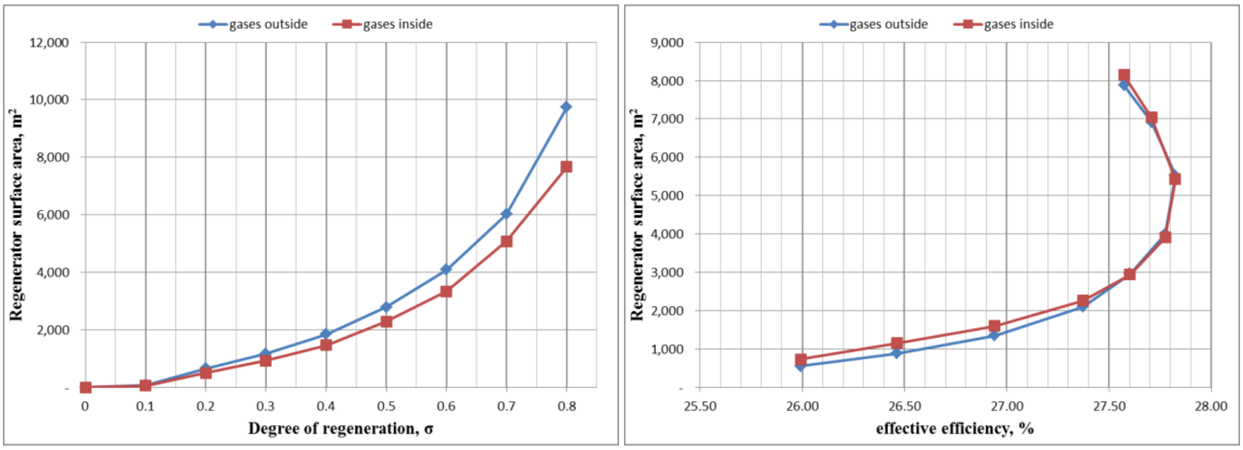
point on one pipe) and on the maximum wall temperature was investigated. As shown by the analysis of the results of discrete calculations, the maximum temperature differences in the basic variants were 28.9-33.6 ^{o}C at the maximum wall temperatures of 394.4-396.2 ^{o}C. In embodiments with gas bypass, the maximum temperature differences increase by 10 ^{o}C with a bypass fraction φ = 0.2-0.3 (just where the maximum efficiency is present). Further, with an increase in the bypass fraction, the maximum temperature differences decrease, and the maximum wall temperatures also decrease.



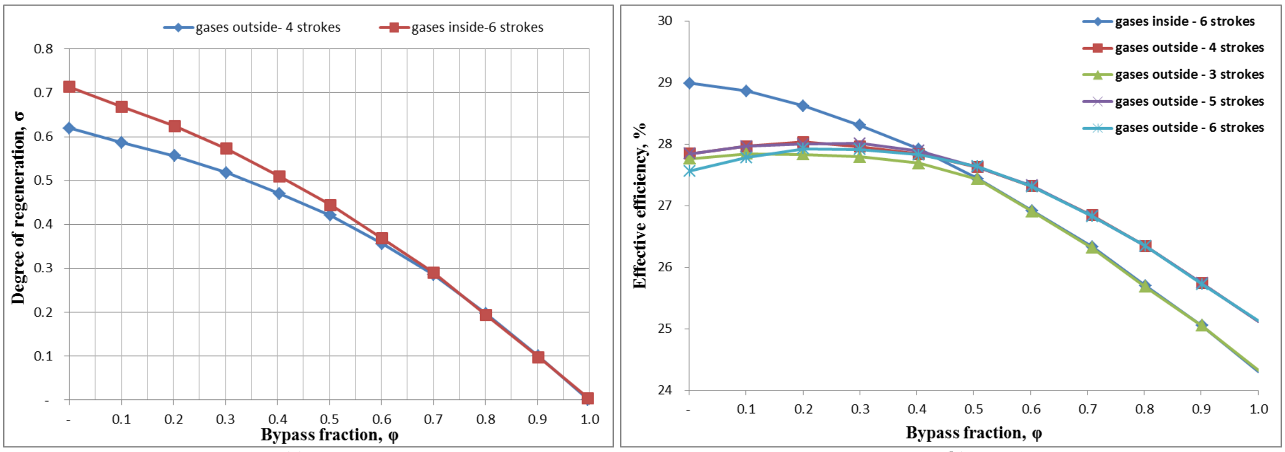
(a) (b)

Fig. 1. Indicators of efficiency of the regenerator and installation depending on the number of strokes (sections) in the regenerator: a - the degree of regeneration; b - effective efficiency of the gas turbine power plant



(a) (b)

Fig. 2. Indicators of efficiency of the regenerator and power plant depending on the surface area of the regenerator: a - the degree of regeneration; b - effective efficiency of the gas turbine power plant



(a) (b)

Fig. 3. Indicators of the efficiency of the regenerator and the power plant, depending on the proportion of hot gases bypass past the regenerator: a - the degree of regeneration; b - effective efficiency of the gas turbine power plant

V. Conclusions

Mathematical models, methods and algorithms for the system analysis of a gas turbine plant have been developed, taking into account the efficiency of the regenerator-air heater and the distribution of local thermo-hydraulic parameters in it. The design of a tubular regenerator has been developed for the power plant gas turbine GT 35 Hz. calculation is made in two ways: moving gases outside the tubes or inside them. The maximum effective efficiency, which can be obtained using a regenerator of the selected design, was found, which in the variant where the gases are outside was 27.83% (4 strokes, the area of 5274.7 m^2), in the variant where the gases are inside the pipes, it was 29.08 % (6 passes, area 9824.2 m^2). Thus, we can conclude that the option where the gases are outside is more efficient in terms of heat transfer, and the option where the gases are inside is better in terms of the thermodynamic efficiency of the installation. The study of the expediency of using the bypass of gases past the regenerator from the point of view of increasing its reliability and efficiency of the installation has been carried out. It seems expedient to carry out optimization calculations in the future, taking into account the balance between fuel economy and the cost of manufacturing devices.

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