ENHANCEMENT OF THE ECONOMIC
AND ECOLOGICAL CHARACTERISTICS
OF GAS-TURBINE PLANTS BY MEANS
OF THERMOCHEMICAL RECUPERATION

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We have developed a new scheme of gas-turbine plant with thermochemical recuperation by means of natural gas reforming in combustion products. Thermodynamic calculations show that this scheme enables one to enhance the efficiency of plant by 5–5.5% as compared with the traditional scheme of air recuperation. In addition, the new scheme leads to a substantial decrease in the emission of NOx.

Keywords: Gas-turbine plant; Efficiency; Thermochemical and air recuperation; Harmful emissions

During the last years, a stable tendency is observed toward a fast growth of the price of natural gas. Therefore, the search for ways of enhancing the efficiency of its use in various thermal plants attracts the attention of researchers in different countries. In addition, more and more rigid ecological demands lead to the necessity of decreasing the amount of harmful substances (in particular, nitrogen oxides) thrown into the atmosphere.

To enhance the efficiency of using natural gas in a thermal plant, it is necessary, one way or another, to utilize the sensible heat of combustion products (CPs) at the plant exit. One of the possible variants is connected with the heating of natural gas (in what follows, we assume that it consists of pure methane), but its amount is not great, and, furthermore, it begins to decompose at fairly low temperatures. Another possibility is connected with the heating of air, fed to combustion, but this way also has two shortcomings, namely, the amount of air and its specific heat are less than the corresponding parameters of CPs, and, with increase in the air temperature, the emission of NOx grows substantially.

At the same time, there is a different, in principle, variant of enhancement of the efficiency of using natural gas that has no such shortcomings, i.e., this is methane reforming at the expense of sensible heat of CPs, which is called ther-
mochemical recuperation (Nosach, 1989). In this case, we obtain a new fuel with a higher calorific value than methane and a significant amount of molecular hydrogen and carbon monoxide, whose combustion is accompanied by quite low emissions of NO\(_x\) (Sigal, 1988). Hence, thermochemical recuperation (TCR) enables one to solve ecological problems as well.

As is known, the efficiency of present-day gas-turbine plants (GTPs) is on the order of 36–38%. Probably, the most reasonable way of increasing their efficiency is connected with TCR. However, two difficulties arise here: first, the degree of methane reforming decreases with decrease in the temperature of CPs used as a heat carrier in the thermochemical reformer, but the temperature of exhaust gases of GTPs is usually quite low; second, GTPs work with a high excess-air coefficient, but the degree of reforming falls sharply if the reacting mixture fed to the thermochemical reformer contains oxygen. These difficulties can be overcome if one applies the following measures proposed at the Institute of Engineering Thermophysics:

1) The thermochemical reformer is placed not after the turbine, but at the break, i.e., between its high- and low-pressure cylinders (so-called intermediate recuperation).

2) Excess-air coefficient decreases to unity, but for decreasing the temperature of CPs before the turbine to an acceptable level, a certain part of CPs is fed to the combustion chamber.

Furthermore, for more complete utilization of the sensible heat of CPs, it seems reasonable to use here a combined recuperative system, namely, TCR + air heating. In what follows, we describe some results of the development of a combined recuperative system for GTPs and calculations of its efficiency.

In Figure 1, we show a schematic diagram of a GTP with TCR. The main element of this system is the reformer/heat exchanger 4, where CPs after the high-pressure cylinder 3 serve as a hot heat carrier. After expansion of CPs in the low-pressure cylinder 5, the corresponding part of them is thrown into the atmosphere, and the others, after cooling in heat exchanger 6, are compressed in compressor 7 to the initial pressure \(p_1\) (here and below, subscripts correspond to numbers of points in Fig. 1). Further, the flow of compressed CPs is divided into two parts, which are marked by A and B in Figure 1. Flow A, representing the stoichiometric amount of CPs in the reaction of methane reforming, i.e., \((\text{CO}_2 + 2\text{H}_2\text{O} + 7.52\text{N}_2)/3\), is mixed with CH\(_4\) in the mixer 8 and then comes to the reformer/heat exchanger 4, where the reaction of reforming is realized. Another flow of CPs (B), consisting of \(\beta(\text{CO}_2 + 2\text{H}_2\text{O} + 7.52\text{N}_2)/3\), is directed to the apparatus 4. Furthermore, the stoichiometric amount of air \((2\text{O}_2 + 7.52\text{N}_2)\) after
compressor 2 is also heated in the reformer/heat exchanger 4. Afterward, these three flows are fed to the combustion chamber 1. Naturally, the coefficient $\beta$ is chosen in such a way that the temperature of CPs before the cylinder 3 should be equal to the assigned value.

As follows from Figure 1, the useful work of this plant is equal to the sum of works of cylinders 3 and 5 ($A_3, A_5$) minus the sum of works of compressors 2 and 7 ($A_2, A_7$). The plant efficiency is

$$\eta = \frac{A_3 + A_5 - A_2 - A_7}{Q},$$

where $Q$ is the calorific value of methane.

In the thermodynamic calculations, we varied the values of $p_1, T_1, T_8$ (in all variants, we took $T_{11} = T_0 = T_8$). For simplification, the following assumptions were accepted:

1) The residence time of reacting mixture in the reformer/heat exchanger is sufficient for reaching the equilibrium composition of reformed fuel (RF).
2) The losses of heat and pressure in all apparatus and pipelines are negligible.

All parameters of reformed fuel (its composition, calorific value, and enthalpy) were calculated according to the procedure described by Nosach and
Shraiber (2009). For each variant, the pressure $p_2$ was selected in such a way that the temperature of CPs after the cylinder 3 be the same: $T_2 = 1130$ K. The internal efficiency of both cylinders of the turbine was taken to be equal to 0.92, and that of both compressors to be 0.88. Obviously, the efficiency of the system under consideration will grow with an increase in $T_8$, but at the same time, temperature $T_3$ after the reformer/heat exchanger decreases. Therefore, for each variant of calculations, we imposed the following limitations on $T_8$:

$$\min (T_3 - T_6, T_3 - T_7, T_3 - T_{10}) \geq 20 \text{ K.}$$ (2)

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**Figure 2** Increment in efficiency versus $T_8$: (1, 3) $T_1 = 1400$ K, (4, 6) $T_1 = 1500$ K, (7, 9) $T_1 = 1600$ K, (1, 4, 7) $p_1 = 2.5$ MPa, (2, 5, 8) $p_1 = 2$ MPa, (3, 6, 9) $p_1 = 1.5$ MPa.

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**Figure 3** Maximal increment in the efficiency of GTP vs. $T_1$: (1) $p_1 = 2.5$ MPa, (2) $p_1 = 2$ MPa, (3) $p_1 = 1.5$ MPa.
## TABLE 1 Parameters of working media

<table>
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<tr>
<th>No. of point in Figure 1</th>
<th>Parameters</th>
<th>No. of variant</th>
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<td>H(_2)O</td>
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<td>1009.3</td>
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<td>883.4</td>
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<td>129.3</td>
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<td>Efficiency of the plant, %</td>
<td>57.63</td>
<td>56.74</td>
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<td>56.74</td>
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Note: pressures at points 6–11 are the same as at point 1; temperatures at points 9 and 11 are the same as at point 8.
The value of \( T_8 \) corresponding to equality in (2) is denoted by \( T_8' \) (naturally, this is the maximum possible \( T_8 \)).

Some results of two variants are presented in Table 1 (here, \( p_1 = 1.5 \) and 2 MPa, \( T_8 = 1010.2 \) and 1009.3 K). We see that the proposed scheme at fairly high \( T_8 \) enables one to obtain a very great efficiency of a gas-turbine plant. Table 2 illustrates the dependence of \( \eta \) calculated by formula (1) on the values of \( T_1 \) and \( T_8 \) for \( p_1 = 1.5 \) MPa (two last columns correspond to \( T_8' \)). In Table 3, we present the values of \( T_8' \) for different \( p_1 \) and \( T_1 \).

To reveal the advantages of using the TCR method, we also calculated the efficiency of GTPs with recuperative air heating only (it was denoted by \( \eta^o \)) for the same temperatures \( T_1 \) and \( T_2 \). The dependence of \( \Delta \eta = \eta - \eta^o \) on \( p_1 \), \( T_1 \), and \( T_8 \) is displayed in Figure 2. As follows from here, the value of \( \Delta \eta \) grows appreciably with a decrease in the initial pressure \( p_1 \) or an increase in \( T_1 \), other conditions being equal. The greatest \( \Delta \eta \) corresponds to variant 1 in Table 1 (\( p_1 = 1.5 \) MPa, \( T_1 = 1600 \) K, \( T_8 = T_8' = 1010.2 \) K); here, \( \Delta \eta = 5.53\% \). At the same time, a low initial temperature gives the maximal \( \Delta \eta = 3.67\% \), and middle temperature, \( \Delta \eta = 4.72\% \). In Figure 3, we show the maximum possible \( \Delta \eta \) for different \( p_1 \) and \( T_1 \).

It seems also interesting to compare a GTP-TCR and a GTP with air recuperation (GTP-AR) under conditions of equal efficiencies, \( \eta = \eta^o \) (obviously, we should take a higher \( T_1 \) value for GTP-AR). If, for example, \( p_1 = 1.5 \) MPa, \( T_1(\text{GTP-TCR}) = 1400 \) K, and \( T_8 = 1000 \) K, we should take \( T_1(\text{GTP-AR}) \approx 1520 \) K, i.e., this increment in the initial temperature is equal to \( \delta T \approx 120 \) K. For higher temperature levels, the \( \delta T \) value will be greater. Thus, TCR enables one to obtain high efficiency without increasing the initial temperature, which is very important from the viewpoint of the cost of materials for GTP.

### Table 2 Efficiency of the gas-turbine plant (%)

<table>
<thead>
<tr>
<th>( T_1, ) K</th>
<th>( T_8, ) K</th>
<th>900</th>
<th>950</th>
<th>1000</th>
<th>1010.2</th>
<th>1021</th>
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<td>1500</td>
<td>48.4</td>
<td>51.0</td>
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</tr>
<tr>
<td>1600</td>
<td>50.0</td>
<td>53.0</td>
<td>56.8</td>
<td>57.6</td>
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### Table 3 Maximal values of \( T_8 \)

<table>
<thead>
<tr>
<th>( T_1, ) MPa</th>
<th>( p_1, ) 1.5</th>
<th>2</th>
<th>2.5</th>
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<tbody>
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<td>1400</td>
<td>1033.2</td>
<td>1030.2</td>
<td>1027.8</td>
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<tr>
<td>1500</td>
<td>1021</td>
<td>1019.3</td>
<td>1017.7</td>
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<tr>
<td>1600</td>
<td>1010.2</td>
<td>1009.3</td>
<td>1008.5</td>
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**Nomenclature**

A  work  
I  enthalpy  
J  total enthalpy  
p  pressure  
Q  calorific value  
T  temperature  
\( \beta \)  coefficient  
\( \eta \)  efficiency  
AR  air recuperation  
CPs  combustion products  
GTP  gas-turbine plant  
NG  natural gas  
RF  reformed fuel  
TCR  thermochemical recuperation

**References**

