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# MODEL OF GREENHOUSE GAS EMISSION MINIMIZATION UNDER VARIABLE LOAD OF A STEAM BOILER

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The article investigates the problem of reducing greenhouse gas emissions during variable load of steam generating energy equipment. The aim of the research is to develop a method and model that allow reducing greenhouse gas emissions by using a mixture of regulated methane and alternative low-calorific gas. The scientific and practical significance of the work lies in the creation of regulation methods that ensure the ecological and energy efficiency of the equipment without its modernization. The research methodology is based on mathematical modeling of the combustion process of a gas mixture using chemical kinetics equations for the isoenthalpic process. A model has been developed within the study that allows accurately determining the adiabatic combustion temperature of a fuel-air mixture, as well as the quantitative and qualitative composition of the combustion products formed under different ratios of methane and low-calorific gas. A relationship between the fuel mixture consumption, its composition, and the stability of the combustion products volume has been identified. It has been shown that maintaining constant exhaust gas flow can be achieved by adjusting the fuel mixture composition, which influences the equipment power and the temperature of the exhaust gases. The effect of maintaining isoenthalpic process conditions on the amount of energy released has been separately considered, which allowed determining the patterns for regulating the equipment power without significant losses in energy efficiency. The value of the study lies in the development of new approaches to energy efficiency management and greenhouse gas emissions reduction. The practical significance lies in the possibility of applying the obtained results to regulate equipment power without modernization, while maintaining a constant volume of combustion products.
**Keywords:** power regulation, low-calorific gas, greenhouse gases, chemical kinetics, isoenthalpic process.

**Introduction.** One of the directions of the international decarbonization program for industry and energy is the development of renewable energy sources. "Green" energy requires significant capital investments. Therefore, it is fully accessible only to a limited number of industrially developed countries. However, even these countries, in order to ensure the dispatchability of the energy supply, retain the majority of their generation based on the combustion of extractable fuels. Furthermore, the preservation and even development of traditional energy generation sources is even more pronounced in countries with limited financial resources.

Dispatchability is determined by the ability to make operational changes in the power of the energy equipment being operated within a wide range. At the same time, it remains crucial to address the issue of minimizing greenhouse gas emissions across the entire operational range of traditional hydrocarbon energy equipment based on extracted fuels.

**Analysis of research and publications.** The elimination or significant reduction of greenhouse gas emissions in the future is considered from the perspective of transitioning to hydrogen as a fuel. Even at the stage of analyzing prospects, a number of issues arise during the development of new technology [1]. Significant financial investments are required to create hydrogen production technologies, develop the necessary logistics infrastructure, and implement energy equipment.

Currently, or in the near future, instead of fully hydrogen-based energy, the following possible measures can be identified for reducing greenhouse gas emissions:

– mixing traditional hydrocarbon fuels with hydrogen in a specific proportion, which does not require a radical change in the equipment used;

– combustion of organic material of non-extracted origin with a specified energy production volume to reduce greenhouse gas emissions, accompanied by a special technological regulation of the energy installation.

The current state of hydrogen production technology is primarily determined by the reforming of extracted organic fuel [2]. By 2021, out of 60 million tons of hydrogen produced annually, approximately 96% was obtained by processing extracted fuels (49% natural gas, 29% liquid hydrocarbons, and 18% coal), with only 4% produced by water electrolysis. This situation resulted in high indirect carbon dioxide emissions [3]. It should be noted that the energy for hydrogen production processes largely depends on the extracted organic fuel used. Thus, using hydrogen as a fuel in industry and energy will lead to higher carbon dioxide emissions than using fossil fuels for the same energy output.

In the absence of developed technology for producing carbon-neutral "green hydrogen from renewable energy sources" and "orange hydrogen from nuclear energy," the use of its "gray" variant may currently be justified in research and demonstration projects assessing the environmental impact. It should be noted that a series of studies has been conducted that allows for increasing the reliability of fuel rod operation in various modes [4, 5], which makes it possible to consider nuclear power plants as hydrogen suppliers. There are known studies on automated control models and methods for adjusting power in nuclear power plants to ensure stable control in load-following modes, which effectively allows for regulating hydrogen oxidation [6, 7].

In [8, 9], the issue of safety regarding the use of a methane-hydrogen mixture in existing boiler equipment is addressed. In [8], the moderate impact of hydrogen content up to 50% in the fuel mixture on the energy and environmental characteristics of the equipment is noted, and the possibility of safe operation at such concentrations is demonstrated. In [9], a hydrogen concentration of 50% in the fuel mixture in the pipeline is considered moderately hazardous, while a concentration up to 25% is regarded as safe. It should be noted that this refers to the safety of the mixture during normal equipment operation (without leaks). While the possibility of a gas fuel mixture leak can be minimized, it cannot be entirely ruled out. The methane-hydrogen mixture with air is more explosive than pure methane due to hydrogen's lower minimum ignition energy in air compared to methane (0,020 mJ versus 0,29 mJ – an order of magnitude difference) [1, Table 1].

There is also a thermophysical feature of the methane-hydrogen mixture that complicates its use in existing equipment. The gravimetric calorific value of hydrogen is higher compared to methane (119.9 MJ/kg versus 45.8 MJ/kg), which gives hydrogen an advantage when transported in liquid form. However, hydrogen is supplied to energy equipment in a gaseous state. In this case, the volumetric energy ratio is reversed (10.7 $MJ/m^3$ versus 33 $MJ/m^3$) [1, Table 1]. It is important to consider the potential decrease in the power output of installed energy equipment as the hydrogen concentration in the fuel mixture increases, in the absence of adjustments to the geometric dimensions of the gas supply and exhaust fuel pathways.

Boilers that use natural gas as fuel have an efficiency of 88-93%. Improving this efficiency reduces carbon dioxide emissions for a given energy production volume. In the energy balance of boilers, the largest losses occur through flue gases (5-10%). Therefore, the main area where significant efficiency improvements can be achieved is by reducing these losses. For example, in [10], a method was developed to determine the variable composition and energy content (enthalpy) of gaseous fuel during combustion. This method enables the calculation of boiler efficiency based on the current load and flue gas temperature.

A change in the load of a gas boiler causes a corresponding change in the flue gas temperature. The need for cyclic power adjustments in a non-condensing boiler within the range of 40% to 100% [11] can result in flue gas temperatures varying between ~393 K and 473 K (120°C to 200°C). Maximum load corresponds to the highest temperature and the greatest

losses. The minimum temperature is determined by the need to maintain a non-condensing mode during flue gas evacuation. The presence of hydrogen in the fuel mixture, in quantities deemed safe for operational conditions, slightly increases [12, Table 4] the dew point temperature of the flue gases. This phenomenon is advantageous for condensing boilers [13] but undesirable for non-condensing energy systems [14]. In such systems, to prevent condensate formation in the flue gas channels, the flue gas temperature must be increased, which leads to a corresponding decrease in efficiency.

Theoretically, in a non-condensing flue gas removal mode, an efficiency improvement of up to ~4% could be achieved by lowering the maximum flue gas temperature to an acceptably low level. However, this approach is not feasible with the current designs of energy equipment. It should also be noted that, in a cyclic variable mode, the energy system operates at maximum load for only a small fraction of the time. Consequently, even in theory, the average efficiency improvement would amount to less than 4%.

Greenhouse gases are formed during the use of fossil fuels. They do not include alternative combustible gases obtained from the processing of secondary raw materials. Examples of such gases are the products of wood gasification or agricultural waste [15]. These gases are characterized by a lower calorific value compared to methane. Therefore, a complete replacement of methane with alternative combustible gases would lead to a decrease in the maximum capacity of the installed equipment.

The use of a fuel mixture of methane and alternative combustible gases obtained from organic raw materials or renewable sources, differentiated according to the specified load, appears to be relevant in energy steam-generating installations.

**Research Objective.** The objective of the research is to develop a method and model for reducing greenhouse gas emissions under varying load conditions of steam-generating energy equipment through the use of a regulated mixture of methane and alternative gas.

To achieve this goal, the following tasks were set:

– to develop a method and model for forming the composition of the combustible gas mixture that corresponds to the specified load of the steam-generating energy equipment;

– to determine the impact of the mixture composition on the parameters of the combustion products (flue gases);

– to determine the influence of the technical parameters of the installed energy equipment on the composition of the combustible gas mixture while ensuring the specified operating mode.

**Main Part.** The use of a low-calorific component in the combustible gas mixture can have different effects on the operation of the installed equipment. On one hand, it will lead to a change in the quantity of flue gases and, accordingly, affect the equipment's efficiency. On the other hand, reducing the calorific value of the gas fuel mixture increases the required volume to ensure the specified power output. This feature must be considered when evaluating the throughput capacity of the fuel supply system.

With varying boiler load, the adiabatic combustion temperature of the fuel remains unchanged if a constant gas composition is used and optimal excess air is provided. However, the temperature of the flue gases is variable and reaches its maximum at maximum load. The reason for this is well known. At maximum load, the greatest amount of combustion products is formed. These pass through the gas ducts at maximum speed, and, accordingly, spend the least amount of time in contact with the heat exchange surfaces. To account for this feature, one of the methods for regulating the flue gas temperature and, as a result, the boiler's efficiency, may be applied.

It can be assumed that the addition of ballast gases will influence the change in the temperature of the flue gases. For instance, as shown in [16, Table 5], at a lower load (81% versus 99%), but with a correspondingly higher air excess ratio (1.57 versus 1.2), the temperature of the flue gases was higher (167°C versus 115°C). This occurs when using fuel of constant composition. When the optimal amount of air for combustion is supplied, the

temperature ratio should be reversed. This effect can be explained by the air excess acting as a ballast gas. A similar effect can be observed when fuel is supplied as a mixture of gases, as additionally shown in [17]. Preliminary studies [18] and further development of models and methods [19] have made it possible to consider various options for using natural (methane) and blast furnace gases in different proportions by developing the fundamentals for conducting such processes [20]. In [21], the option of using natural (methane) and blast furnace gases in different proportions is discussed. The latter has a deliberately lower calorific value and contains several ballast gases. It is worth noting that at the same load (for example, 40 tons per hour of steam), when the share of blast furnace gas in the fuel mixture increases from 0.198 to 0.755, the temperature in the furnace decreases, and the temperature of the flue gases increases from 148°C to 174°C [21, Table 2].

Thus, there is a potential opportunity to control the temperature of the exhaust gases at a given boiler load. To increase the temperature, it is necessary to replace some portion of the standard fuel (e.g., methane) with a gas of lower calorific value, in a corresponding greater amount. Controlling the exhaust gas temperature allows it to remain constant across the entire load range. However, this results in an increase in temperature to the value corresponding to the maximum load. Afterward, it is necessary to design a low-temperature economizer to reduce the exhaust gas temperature to levels that prevent the condensation of water vapor in the flue gas system.

The implementation of such a steam generation method and control strategy may require the use of a large volume of low-calorific fuel gas mixtures. The possible volume could be limited by the throughput capacity of the existing fuel mixture supply system and may require its modernization.

Using only the existing capacity of the fuel gas mixture delivery line presents another potential opportunity to reduce greenhouse gas emissions at loads different from the maximum. When using standard gas, such as methane, reducing the load is accompanied by a decrease in its supply. The amount of methane supplied can be further reduced by mixing in a low-calorific gas to the volume corresponding to the capacity of the fuel delivery line, while maintaining the desired load. When implementing this management method, it is essential to monitor the amount and composition of the flue gases produced. If their volume is too low, there is a risk of entering an undesirable condensation operation mode.

The presented analysis of the two control methods allows for the proposal of an operational method for a steam-generating energy unit that enables controlling (or setting) the composition of the fuel gas mixture.

**The method for calculating the composition of the gas mixture.** In the case of combustion process equilibrium, the composition and quantity of flue gases can be determined using a balance method. However, a more universal approach is to use the calculation method based on chemical kinetics equations. In this case, it is possible to determine not only the temperature of the combustion products but also their composition, if it depends on this temperature. For example, this applies to nitrogen oxides. Moreover, the calculation model can be structured in such a way that the composition of the flue gases is determined for the same volume but with different fuel mixture compositions. At the same time, the required volume of the fuel mixture will also be determined.

The calculation method is as follows:

−the volumetric fraction of the main gas is gradually reduced (from 100% to 0%) and is replaced with the corresponding amount of lower-calorific gas. For each of these ratios, knowing the compositions of the input gases, air, and the excess air coefficient α, the composition of the fuel-air mixture and its enthalpy are determined;

−using a model based on chemical kinetics equations, the composition of flue gases (in volumetric fractions) is determined for each type of fuel-air mixture, depending on the amount of lower-calorific gas;

−each individual composition of flue gases is calculated for the same volume when using any type of fuel-air mixture, for example, for 1 mole of the burning substance. A key feature of the method is the calculation of the required amount ($M_T$) of the corresponding fuel-air mixture in moles to produce 1 mole of flue gases. Since both the initial fuel-air mixture and combustion products are in the gaseous state, all mole ratios correspond to volumetric ones;

−for each fuel-air mixture composition, the amount of energy $Q_m$ transferred by 1 mole of flue gases when cooled to a specified temperature (e.g., 393 K or 120°C) is determined. This is necessary for further application regarding the ratio of standard gas to lower-calorific-value gas at different boiler loads. Additionally, the lower heating value relative to 1 mole of flue gases can be determined, and if desired, the lower heating value can be converted using the $M_T$ for 1 mole of the fuel-air mixture. The obtained results can be used to determine the ratio of standard gas to lower-calorific-value gas, as well as the required amount of the resulting mixture to ensure the boiler operates at the specified load. This ensures the consistency of the amount of flue gases.

**Model.** The input data are determined by the gross formula of the fuel mixture. The combustible gases used in energy production have a hydrocarbon composition and consist of a limited number of chemical elements. Taking into account the composition of the air used as an oxidizer, the list may include C, H, O, and N. In some cases, sulfur (S) may be present, but its presence or absence does not affect the generality of the solution. The gross formula of the fuel mixture (excluding sulfur) will have the following form:

$$C_{b_C} H_{b_H} O_{b_O} N_{b_N} , \tag{1}$$

where $b_C$, $b_H$, $b_O$ and $b_N$ are the number of atoms of the respective elements in the gross formula. Their values are determined based on the ratio of the primary and lower-calorific gases, as well as the excess oxidizer coefficient. Additionally, based on the enthalpy values of the input gases and air, the specific enthalpy of the fuel-air mixture is determined (1).

Based on equation (1), the list of substances that may be present in the flue gases is determined. If the oxidizer excess coefficient $\alpha > 1$, and the combustion process is in equilibrium, this list may include:

$$CO_2, \; H_2O, \; O_2, \; N_2, \; NO . \tag{2}$$

In the case of $\alpha < 1$, gases that are formed due to incomplete combustion of the fuel mixture may appear: $CO$, $H_2$, $CH_4$. In this case, we will limit ourselves to the list in equation (2).

In the equilibrium process of thermodegradation of the fuel mixture, for substances from the list (2), the chemical kinetics equation can be written according to the law of mass action. Given that the substance exists in the gas phase, it is more convenient to express this equation using partial pressures rather than molar concentrations. In this case, the calculations should take into account not just one mole of the initial raw material (along with the oxidizer), but a certain number of its moles ($M_T$), which leads to a numerical equivalence between molar concentration and partial pressure. This value is determined during the calculations. For the products in (2), the law of mass action equation will have the following form:

$$\frac{P_C \cdot P_O^2}{P_{CO_2}} = K_{CO_2}(T) ; \quad \frac{P_H^2 \cdot P_O}{P_{H_2O}} = K_{H_2O}(T) ; \tag{3}$$

$$\frac{P_O^2}{P_{O_2}} = K_{O_2}(T) ; \quad \frac{P_N^2}{P_{N_2}} = K_{N_2}(T) ; \quad \frac{P_N \cdot P_O}{P_{NO}} = K_{NO}(T) . \tag{4}$$

In the denominators, there are the partial pressures of the corresponding substances. In the numerators, there are the partial pressures of the atoms of the chemical elements that make up these substances. The exponent is equal to the number of corresponding atoms in the substance's formula. $K_{CO_2}(T)$; $K_{H_2O}(T)$; $K_{O_2}(T)$; $K_{N_2}(T)$; $K_{NO}(T)$ – the equilibrium constants of the corresponding substances at the current combustion temperature (tabulated values).

Using (1), material balance equations for each chemical element included in this formula can be written for the desired model:

for [C]
$$b_C \cdot M_T = P_{CO_2} + P_C; \qquad (5)$$

for [H]
$$b_H \cdot M_T = 2 \cdot P_{H_2O} + P_H; \qquad (6)$$

for [O]
$$b_O \cdot M_T = 2 \cdot P_{CO_2} + P_{H_2O} + 2 \cdot P_{O_2} + P_{NO} + P_O; \qquad (7)$$

for [N]
$$b_N \cdot M_T = 2 \cdot P_{N_2} + P_{NO} + P_N. \qquad (8)$$

In writing these equations, it is taken into account that not one, but $M_T$ moles of the initial fuel-air mixture with the gross formula (1) are considered.

To close the system, Dalton's law is used:
$$P_\Sigma = P_{CO_2} + P_{H_2O} + P_{O_2} + P_{N_2} + P_{NO} + P_C + P_H + P_O + P_N. \qquad (9)$$

Here, $P_\Sigma$ is the pressure inside the boiler furnace.

Equations (3–9) form a closed nonlinear algebraic system for determining 10 values – 9 partial pressures and $M_T$.

$$P_{CO_2}, P_{H_2O}, P_{O_2}, P_{N_2}, P_{NO}, P_C, P_H, P_O, P_N, M_T. \qquad (10)$$

**Features of the solution.** The solution of the obtained system is complicated by the large ratio of the values of the variables included in it. This ratio can exceed 15 orders of magnitude ($>10^{15}$). To overcome this problem, all the equations in the system can be logarithmically transformed. During the solution process, the values of pressures and $M_T$, which are logarithms, are determined. In this case, the resulting values will differ by factors, not orders of magnitude.

The original system (3-9) will take the form:

$$\ln(P_C) + 2 \cdot \ln(P_O) - \ln(P_{CO_2}) = \ln(K_{CO_2}(T)); \qquad (11)$$

$$2 \cdot \ln(P_H) + \ln(P_O) - \ln(P_{H_2O}) = \ln(K_{H_2O}(T)); \qquad (12)$$

$$2 \cdot \ln(P_O) - \ln(P_{O_2}) = \ln(K_{O_2}(T)); \qquad (13)$$

$$2 \cdot \ln(P_N) - \ln(P_{N_2}) = \ln(K_{N_2}(T)); \qquad (14)$$

$$\ln(P_N) - \ln(P_O) - \ln(P_{NO}) = \ln(K_{NO}(T)); \qquad (15)$$

$$\ln(b_C) + \ln(M_T) = \ln(P_{CO_2} + P_C); \qquad (16)$$

$$\ln(b_H) + \ln(M_T) = \ln(2 \cdot P_{H_2O} + P_H); \qquad (17)$$

$$\ln(b_O) + \ln(M_T) = \ln(2 \cdot P_{CO_2} + P_{H_2O} + 2 \cdot P_{O_2} + P_{NO} + P_O); \qquad (18)$$

$$\ln(b_N) + \ln(M_T) = \ln(2 \cdot P_{N_2} + P_{NO} + P_N); \qquad (19)$$

$$\ln(P_\Sigma) = \ln(P_{CO_2} + P_{H_2O} + P_{O_2} + P_{N_2} + P_{NO} + P_C + P_H + P_O + P_N). \qquad (20)$$

The combustion process in the boiler is considered isoenthalpic. The calculation method is based on determining, through an iterative process, the temperature and the corresponding composition of the reaction products (flue gases) so that their total enthalpy equals the enthalpy of the incoming fuel-air mixture.

**Results.** The solution is obtained based on the method and model presented above (3-9).

Let's represent the gross formula of the fuel-air mixture. Methane $CH_4$ is taken as the standard fuel gas, with a formation enthalpy of $I_{CH_4} = -74.85$ kJ/mol (298 K). The low-calorific gas is a mixture of gases produced during the decomposition of pine wood waste in a gasifier. In the considered case, the cooled gas mixture is at 298 K, and after the removal of condensate,

its molar composition is as follows: $CO - 0.394$, $CO_2 - 0.214$, $H_2 - 0.360$, $H_2O - 0.032$ with the gross formula:

$$C_{0.61}H_{0.78}O_{0.85} \tag{21}$$

and the enthalpy of formation $I_{mk} = -135.5 \, \text{kJ/mol}$. The absence of nitrogen is due to the fact that the gasifier, during the decomposition of wood waste, is purged not with air but with oxygen [10]. Such processes are now implemented on an industrial scale.

We define the share of low-calorific gas (21) in the mixture with methane as $\varphi \in [0,1]$. When $\varphi = 0$, the fuel gas consists of methane, and when $\varphi = 1$, it consists entirely of gas (21). In this case, the gross formula of the fuel gas mixture at different values of will have the following form:
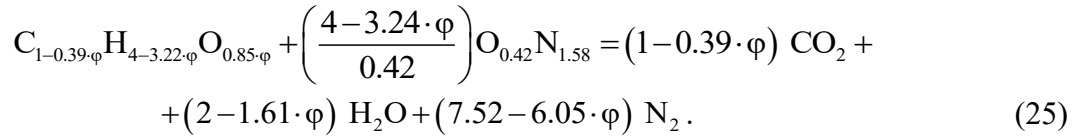
$$C_{1-0.39\cdot\varphi}H_{4-3.22\cdot\varphi}O_{0.85\cdot\varphi} . \tag{22}$$

The enthalpy of such a mixture is determined using the following equation:

$$I_{mix} = (1-\varphi)\cdot I_{CH_4} + \varphi\cdot I_{mk} . \tag{23}$$

Let the composition of air in molar fractions be: $O_2 - 0.21, N_2 - 0.79$ . In this case, the gross formula of air will be:

$$O_{0.42}N_{1.58} . \tag{24}$$

The complete oxidation reaction during the combustion of the fuel gas mixture (22), taking into account (24), can be written as:

$$C_{1-0.39\cdot\varphi}H_{4-3.22\cdot\varphi}O_{0.85\cdot\varphi} + \left(\frac{4-3.24\cdot\varphi}{0.42}\right)O_{0.42}N_{1.58} = (1-0.39\cdot\varphi)\,CO_2 +$$

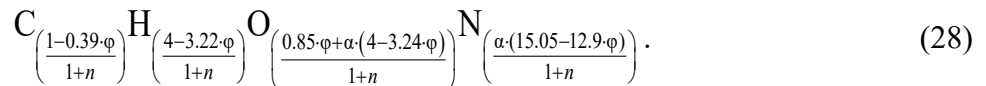$$+(2-1.61\cdot\varphi)\,H_2O + (7.52-6.05\cdot\varphi)\,N_2 . \tag{25}$$

Formula (25) is written for 1 mole of the combustible gas mixture (22). In this case, the coefficient in front of the air's gross formula on the left side of the equation is the mole stoichiometric coefficient:

$$\chi_0 = \frac{4-3.24\cdot\varphi}{0.42} . \tag{26}$$

Taking into account the oxidizer excess coefficient $\alpha$, it shows how many moles of air are used to burn 1 mole of the combustible gas mixture (22):

$$n = \alpha\cdot\chi_0 = \alpha\cdot\left(\frac{4-3.24\cdot\varphi}{0.42}\right). \tag{27}$$

Formulas (22), (24), and expression (27) allow us to determine the gross formula (1 mole) of the fuel-air mixture (1):

$$C_{\left(\frac{1-0.39\cdot\varphi}{1+n}\right)}H_{\left(\frac{4-3.22\cdot\varphi}{1+n}\right)}O_{\left(\frac{0.85\cdot\varphi+\alpha\cdot(4-3.24\cdot\varphi)}{1+n}\right)}N_{\left(\frac{\alpha\cdot(15.05-12.9\cdot\varphi)}{1+n}\right)} . \tag{28}$$

Considering air as a mixture of simple gases $(O_2 + N_2)$, we assume that its enthalpy of formation is zero. In this case, the enthalpy of formation of 1 mole of the fuel-air mixture, taking into account (23), can be determined using the relationship:

$$I = \frac{(1-\varphi)\cdot I_{CH_4} + \varphi\cdot I_{mk}}{1+n} . \tag{29}$$

The calculation results for the first regulation method with $\alpha = 1.1$ are presented in Table 1. The following abbreviations are presented in Table 1: $M_{MG}$ – the number of moles of the combustible gas mixture (taking $\varphi$ into account) required to produce 1 mole of flue gases; $M_{Air}$ – the number of moles of air needed to burn the corresponding amount of $M_{MG}$ moles of the combustible gas mixture; $T_{ad}$ – the adiabatic combustion temperature; $Q_{CP}^1$ – the amount of energy in the form of heat released when 1 mole of flue gases (combustion products) is cooled

to 473 K; m – is the volume multiplier of the corresponding gas mixture relative to the volume of methane required to produce 1 mole of flue gases.

**Table 1.**

The amount of combustible gas mixture for the same volume of flue gases

| φ | $M_{MG}$ | $M_{Air}$ | m | $Q_{CP}^1$ [kJ] 200 °C (473 K) | $T_{ad}$, [K] |
|---|---|---|---|---|---|
| 0 | 0.087 | 0.913 | 1 | 55.4 | 1965 |
| 0.1 | 0.095 | 0.912 | 1.09 | 55.8 | 1972 |
| 0.2 | 0.104 | 0.912 | 1.19 | 56.3 | 1980 |
| 0.3 | 0.115 | 0.911 | 1.32 | 56.9 | 1990 |
| 0.4 | 0.129 | 0.911 | 1.48 | 57.5 | 2000 |
| 0.5 | 0.146 | 0.910 | 1.68 | 58.5 | 2015 |
| 0.6 | 0.169 | 0.909 | 1.94 | 59.6 | 2034 |
| 0.7 | 0.200 | 0.908 | 2.30 | 61.3 | 2060 |
| 0.8 | 0.246 | 0.906 | 2.82 | 63.6 | 2095 |
| 0.9 | 0.318 | 0.902 | 3.65 | 67.4 | 2150 |
| 1 | 0.450 | 0.896 | 5.18 | 74.2 | 2243 |

Let us analyze the obtained results. It could have been assumed that an increase in the proportion of low-calorific gas (with an increase in φ) would lead to a decrease in the flue gas temperature and the energy released during their cooling. However, the calculation results demonstrate the opposite. As the proportion of low-calorific gas in the combustible gas mixture $M_{MG}$ increases, the amount of air $M_{Air}$ required for combustion remains virtually unchanged. This corresponds to a reduction in the proportion of air in the fuel-air mixture and, consequently, a decrease in the proportion of nitrogen (an inert component) in the flue gases. As a result, an increase in the proportion of low-calorific gas leads to a rise in the adiabatic combustion temperature and the amount of energy released per mole of flue gases. It is also worth noting a significant increase in the volume of the combustible gas mixture $M_{MG}$ as the proportion of the low-calorific component grows. Consequently, with the same amount of flue gases, the complete replacement of methane with low-calorific gas results in a volume increase of m = 5.18 times. In other words, on the one hand, with the current equipment configuration, it is impossible to implement the proposed control method. The existing gas supply system cannot provide the required fuel flow rate. On the other hand, modernizing the gas supply system to increase its capacity would enable a 30% increase in the output power of the existing primary equipment. Ensuring the same volumetric flow rate of the combustible gas mixture at different coefficients constitutes the second variant of the method for controlling the equipment's operation. The calculation results are presented in Table 2. Here, as in the first variant $\alpha = 1.1$.

**Table 2.**

The amount of flue gases at the same volumetric flow rate of combustible gases

| φ | $Q_{fg}$ [kJ]  120 °C (393 K) | ε | V [mole] |
|---|---|---|---|
| 0 | 57.9 | 1 | 1 |
| 0.1 | 53.6 | 0.93 | 0.92 |
| 0.2 | 49.3 | 0.85 | 0.84 |
| 0.3 | 45.1 | 0.78 | 0.76 |
| 0.4 | 40.7 | 0.7 | 0.68 |
| 0.5 | 36.4 | 0.63 | 0.6 |
| 0.6 | 32.1 | 0.55 | 0.52 |
| 0.7 | 27.8 | 0.48 | 0.44 |
| 0.8 | 23.5 | 0.41 | 0.35 |

| $\varphi$ | $Q_{fg}$ [kJ]  120 ºC (393 K) | $\varepsilon$ | V [mole] |
|---|---|---|---|
| 0.9 | 19.2 | 0.33 | 0.27 |
| 1 | 14.8 | 0.26 | 0.19 |

Here, $Q_{fg}$ represents the amount of energy in the form of heat released when the flue gases are cooled to a temperature of 393 K. These gases are formed during the combustion of a combustible gas mixture of the same volume (this volume is equal to the volume of methane consumed to produce 1 mole of flue gases); $\varepsilon$ denotes the fraction of the released heat relative to the amount generated during methane combustion; V represents the volume of flue gases produced during the combustion of a combustible gas mixture of the same volume (this volume is equal to the volume of methane consumed to produce 1 mole of flue gases).

In the case of using a gas of constant composition, the power reduction of the equipment is achieved by reducing the gas flow. In the first approximation, the power reduction and the decrease in gas flow are proportional. The change in the amount of flue gases happens in the same proportion. From Table 2, it follows that even when using a mixture of methane and low-calorific gas, the change in power $Q_{fg}$ (or $\varepsilon$) and the volume of flue gases occurs in a similar proportion. However, the amount of methane used does not correspond to this change. For example, reducing the power to ~50% ($\varepsilon=0.48$) occurs when using a mixture with $\varphi=0.7$. In this case, the mixture consists of only 30% methane, instead of 50% when pure methane is used. Thus, the volume of carbon dioxide formed from the methane component will be 20% less than when pure methane is used under the same conditions. This part of the carbon dioxide is considered greenhouse gas, as it is formed from fossil fuel.

Table 3 presents the composition of the flue gases in molar fractions for different compositions of the combustible gas mixture.

**Table 3.**

Composition of flue gases (mole fractions)

| $\varphi$ | $P_{CO_2}$ | $P_{H_2O}$ | $P_{O_2}$ | $P_{N_2}$ | $P_{NO}$ |
|---|---|---|---|---|---|
| 0 | 0.087 | 0.174 | 0.017 | 0.720 | 0.0018 |
| 0.1 | 0.091 | 0.174 | 0.016 | 0.716 | 0.0018 |
| 0.2 | 0.096 | 0.174 | 0.016 | 0.712 | 0.0019 |
| 0.3 | 0.101 | 0.174 | 0.016 | 0.706 | 0.0019 |
| 0.4 | 0.109 | 0.174 | 0.016 | 0.699 | 0.0019 |
| 0.5 | 0.118 | 0.174 | 0.016 | 0.690 | 0.0020 |
| 0.6 | 0.129 | 0.175 | 0.016 | 0.678 | 0.0021 |
| 0.7 | 0.145 | 0.175 | 0.016 | 0.661 | 0.0022 |
| 0.8 | 0.169 | 0.175 | 0.016 | 0.638 | 0.0024 |
| 0.9 | 0.206 | 0.175 | 0.016 | 0.600 | 0.0026 |
| 1 | 0.275 | 0.175 | 0.015 | 0.531 | 0.0030 |

The data in Table 3 show that the mole fraction and, accordingly, the partial pressure of water vapor are almost the same for all values of $\varphi$. Therefore, the composition of the combustible gas mixture has little effect on the dew point temperature of the flue gases. The adequacy of the obtained results is confirmed by data from [20] in the calculations of pyrolysis gas.

The characteristics of the flue gases presented in Tables 2 and 3 allow us to conclude that it is possible to control the power of the installed equipment without modernization using the second method. In this case, in the maneuvering mode, as the power decreases and the proportion of low-calorific gas increases, the amount of carbon dioxide related to the greenhouse effect decreases.

The increase in the adiabatic combustion temperature when adding low-calorific gas to the mixture leads to a slight increase in the formation of nitrogen oxides compared to using only methane (Table 3). As the calculation results show, in the power regulation range (with the increase in the share of low-calorific gas to $\varphi\sim0.7$), the increase in the mole fraction of nitrogen oxides is negligible.

**Conclusions.** As a result of the research, the following conclusions have been made:

–the most rational method for reducing greenhouse gas emissions at the current stage of technological development has been identified. The use of a mixture of methane and low-calorific gas—such as the product of wood waste gasification—helps to reduce the portion of carbon dioxide in flue gases that is related to greenhouse gases. Moreover, with the high cost of natural gas, replacing part of it with cheaper low-calorific gas can lead to a reduction in the cost of the produced energy;

–a method and model have been proposed for calculating the composition of flue gases with varying ratios of methane and low-calorific gases in their mixture. This method allows for determining both the characteristics of the combustible gas mixture based on the given parameters of the flue gases, as well as the amount of flue gases for specified fuel characteristics;

–two possible methods of controlling the power of the installed equipment by adjusting the share of low-calorific gas in the fuel mixture were considered. It was found that reducing power while attempting to maintain a constant volumetric flow rate of exhaust gases is not feasible. Moreover, in this scenario, the power (and the adiabatic combustion temperature) increases as the share of low-calorific gas rises. The reason for this effect was identified, highlighting the need for modernization of the fuel gas system and an increase in the heat exchange surface area (economizer);

–the possibility of controlling power while maintaining a constant volumetric flow rate of the combustible gas mixture was identified. As the share of low-calorific gas increases, the equipment's power decreases. It was demonstrated that it is possible to maintain exhaust gas parameters comparable to those observed when using pure methane. This feature enables the use of existing equipment without modernization for burning gas mixtures, while simultaneously reducing the share of carbon dioxide classified as a greenhouse gas.

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# МОДЕЛЬ МІНІМІЗАЦІЇ ВИКИДІВ ПАРНИКОВИХ ГАЗІВ ПРИ ЗМІННОМУ НАВАНТАЖЕННІ ПАРОВОГО КОТЛА

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Досліджено проблему зниження викидів парникових газів під час змінного навантаження парогенеруючого енергоустаткування. Мета дослідження полягає у розробці методу та моделі, які дозволяють знизити викиди парникових газів шляхом використання суміші регульованого складу метану та альтернативного низькокалорійного газу. Наукова і практична значущість роботи полягає у створенні методів регулювання, які забезпечують екологічність і енергоефективність обладнання без його модернізації. Методологія дослідження базується на математичному моделюванні процесу горіння газової суміші із застосуванням рівнянь хімічної кінетики для ізоентальпійного процесу. У межах дослідження розроблено модель, яка дозволяє з високою точністю визначати адіабатичну температуру процесу горіння паливно-повітряної суміші, а також кількісний і якісний склад продуктів згоряння, що утворюються за умов використання різного співвідношення метану та низькокалорійного газу. Виявлено залежність між витратою паливної суміші, її складом і сталістю об'єму продуктів згоряння. Показано, що підтримання постійної витрати димових газів можливе шляхом регулювання складу паливної суміші, що впливає на потужність обладнання та температуру відхідних газів. Окремо розглянуто вплив дотримання умов ізоентальпійності процесу на зміну кількості виділеної енергії, що дозволило визначити закономірності регулювання потужності обладнання без значних втрат енергоефективності. Цінність дослідження полягає у розробці нових підходів до керування енергоефективністю та скорочення викидів парникових газів. Практичне значення полягає у можливості застосування отриманих результатів для регулювання потужності обладнання без його модернізації, зберігаючи сталий обсяг продуктів згоряння.
**Ключові слова**: регулювання потужності, низькокалорійний газ, парникові гази, хімічна кінетика, ізоентальпійний процес.