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Influence of the Air Swirling Speed on the Processes of Joint Combustion of the Fuel-Air Mixture in the Active Combustion Zone of Power Plants

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Abstract: The search for new solutions in the field of energy, preventing negative impact on the environment, is one of the priority tasks for modern society. It is natural gas that has a stable position in the demand of the UES of Russia for fossil fuel. One of the promising areas is the use of biogas as a source of thermal energy for power plants. It has been established that the main difference between biogas and natural gas, which affects the density, calorific value, and speed of flame propagation, is caused by the presence of more than 30% carbon dioxide in its composition. Combined combustion of natural gas and biogas, subject to good mixing due to the tangentially swirling apparatus of the fuel-air mixture, can increase the stability of biogas combustion, reduce the maximum adiabatic temperature in the zone of active combustion of power boilers of TPPs, which in turn will lead to a decrease in the content of NO_x, CO_2 in products combustion. For the combustion of biogas at the power plants in operation at TPPs of the UES of Russia, it is important to carry out, on the basis of the theoretical data obtained on the effective combustion modes of fuels, the technical re-equipment of the burners. The paper presents a turbulence model k – ε RNG, which makes it possible to simulate the combustion of natural gas and biogas during tangential swirling of the air-fuel mixture. The qualitative characteristics of biogas, the quantitative content of NO_x, CO_2 in the combustion products, the temperature distribution in the zone of active combustion of fuel combinations - natural gas, biogas, natural gas / biogas is presented.

Keywords: Numerical simulation, modeling, biogas, co-combustion, efficiency, emission reduction

Mathematics Subject Classification: 65R20 Numerical methods for integral equations

PACS: 02.60.Cb Numerical simulation; solution of equations

1. Introduction

In Russia, as in most countries of the world, at present one of the goals of state policy is to reduce the level of threats that negatively affect the atmospheric air in populated areas [1]. One of the main sources of air pollution is the power industry, namely the flue gases from the power boilers of TPPs. In this regard, at the legislative level, low levels of maximum permissible concentrations (MPC) of pollutants in the air are established: nitrogen oxides, sulfur oxides, fly ash, benzo (a) pyrene, etc. In accordance with the forecast demand for electricity in the UES of Russia until 2024, an annual increase in electricity consumption is expected, more than 1.2% per year. According to the assessment of the needs of TPPs of the UES of Russia, it is gas that occupies a stable position in fossil fuel (more than 72% to 23-24% of coal from the total fuel demand of TPPs), while by 2024 the share of electricity generation at TPPs will increase to 64.8% [2].

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A promising direction for reducing harmful emissions into the atmosphere is the processing of organic waste and the production of biogas, which, in turn, can be burned in power boilers at TPPs. The use of biogas as fuel for power and hot water boilers is due to several reasons:

- environmental pollution and large volumes of greenhouse gas emissions.
- the need to process biodegradable household waste and purify urban and industrial wastewater, as a result of which, as a residual product, biogas is formed.
- the high cost of natural gas.

The main qualitative difference between biogas and natural gas is the presence of impurities such as ammonia, hydrogen sulfide, hydrogen, carbon dioxide. Table 1 shows the data of the Gas Institute of the National Academy of Sciences of Ukraine, determined on an Agilent 6890N chromatograph, on the composition of biogas of various origins [3].

№ п/п	Fuel source	CH_4, %	C_2H_6, %	CO_2, %	N_2, %	O_2, %	H_2S, %	Q, kJ/nm^3
1	Biogas							
2	Urban Wastewater Treatment Plant	67,75	-	31,75	0,48	0,425	-	22412
3	Distillery	69,3	-	30,2	0,2	0,3	-	24890
4	Livestock farm	69,44	-	30,36	0,09	-	0,11	24941
5	Natural gas							
6	Urengoyskoye field	98	2	0,84	1,05	-	-	36757

One of the most effective methods of burning biogas is combining it with natural gas, which in turn will preserve the heat of combustion and the heat balance of the power plant, as well as ensure a stable combustion of the flame. In addition, the organization of co-firing of natural gas and biogas will reduce the content of toxic combustion products in the exhaust flue gases of power plants. The peculiarity of biogas as a fuel is the lower, in comparison with natural gas, methane content and the presence of carbon dioxide in large quantities.

The studies were carried out using the ANSYS Fluent software package, due to its wide multifunctional capabilities in modeling fuel combustion processes, with the possibility of implementing effective technical solutions, assessing temperatures in the active combustion zone, which are one of the main indicators of the intensity of the formation of toxic substances, the content of individual chemical elements and their compounds in fuel combustion products. At the same time, using well-known classical models, it is possible to analyze the simulated processes with known approximations and implement effective technical solutions. To study thermal and gas-dynamic processes, the k – ε (RNG) turbulence model is used in this work, which allows simulating the combustion process of a fuel-air mixture.

2. Computational model

To simulate the combustion process of a swirling fuel-air flow, a mathematical model is proposed consisting of [4-6], continuity equations, Navier-Stokes equation, and energy equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0; \quad (1)$$

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_i}[\tau_{ij} - \frac{2}{3}\delta_{ij}\frac{\partial u_i}{\partial x_i}] + \frac{\partial}{\partial x_j}(-\rho \overline{u_i^{'}}\overline{u_j^{'}}); \quad (2)$$

$$\frac{\partial \rho e u_i}{\partial x_i} = \rho \dot{q} + \lambda \frac{\partial^2 T}{\partial x_i^2} - \frac{\partial P u_i}{\partial x_i} + \frac{\partial u_i \tau_{ij}}{\partial x_i}; \quad (3)$$

where

$$\tau_{ij} = \tau_{ji} = \mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right); \quad (4)$$

$$-\rho\overline{u_i^{'}}\overline{u_j^{'}} = \mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\left(\rho k + \mu_\tau \frac{\partial u_i}{\partial x_i}\right)\delta_{ij}; \quad (5)$$

where u_i , u_j – velocity components; ρ – density; P – pressure; δ_{ij} – Kronecker delta Кронекера (δ_{ij} =1 if $i = j$ and δ_{ij} =0 if $i \neq j$); μ – dynamic viscosity; μ_τ – the turbulent (or vortex) viscosity; Reynolds stresses gradients $(-\rho\overline{u_i^{'}}\overline{u_j^{'}})$; e – internal energy; $\dot{q}$ – the rate of volumetric heat addition per unit mass; T – temperature; λ – thermal conductivity.

Turbulence model k-ε RNG

Equation model of turbulence k-ε RNG to describe the combustion process for an unsteady flow will take the form, according to [7,8]:

$$\frac{\partial}{\partial x_j}(\rho k u_j) = \frac{\partial}{\partial x_j}[\left(\alpha_k \mu_{eff}\right)\frac{\partial k}{\partial x_j}] + G_k + G_b - \rho\varepsilon - Y_M + S_k; \quad (6)$$

$$\frac{\partial}{\partial x_j}(\rho\varepsilon u_j) = \frac{\partial}{\partial x_j}[\left(\alpha_\varepsilon \mu_{eff}\right)\frac{\partial \varepsilon}{\partial x_j}] + C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + G_{3\varepsilon}G_b) - \rho C_{2\varepsilon}\frac{\varepsilon^2}{k} + R_\varepsilon + S_\varepsilon; \quad (7)$$

where

$$C_{1\varepsilon} = 1.42, C_{2\varepsilon} = 1.68, C_{3\varepsilon} = \tanh\left|\frac{v}{u}\right|, R_\varepsilon = \frac{c_\mu \rho \eta^3\left(1 - \frac{\eta}{\eta_0}\right)}{1 + \beta\eta^3}\frac{\varepsilon^2}{k},$$
$$C_\mu = 0.0854, \eta = S\frac{k}{\varepsilon}, \eta_0 = 4.38, \beta = 0.012,$$

where k – kinetic energy of turbulence; σ_κ, σ_ε – are the inverse effective Prandtl numbers for k and ε, respectively, in the high Reynolds number limit $(\alpha_k = \alpha_\varepsilon \approx 1.393)$; ε – dissipation rate; G_k – parameter characterizing the kinetic energy of turbulence; G_b – parameter characterizing the generation of kinetic energy in the heat flow; Y_M – parameter characterizing the contribution of fluctuating dilatation in the flow; S_k, S_ε – user-defined source terms; μ_t – turbulent viscosity; $C_{3\varepsilon}$ – the degree to the buoyancy; η – dimensionless coordinate; C_μ – empirical coefficient; S – average velocity tensor.

To simulate turbulent viscosity in the core, the expression is defined:

$$(\mu_\tau = \mu_{\tau 0} f(\alpha_s, \Omega, \frac{k}{\varepsilon}), \quad (8)$$

where $\mu_{\tau 0}$ – turbulent viscosity value calculated without vortex modification, Ω – the amount of a characteristic vortex is estimated; α – vortex constant, which takes on different values depending on whether the flow is vortex or only slightly vortex. This vortex modification is always valid for axisymmetric, vortex and 3D flows when the RNG model is selected. For slightly swirling flows, α_s is set to 0.07. However, for highly swirling flows, a higher value can be used α_s.

The proposed mathematical model makes it possible to simulate a non-stationary, turbulent process of mixing a fuel-air mixture, the process of combustion of biogas and natural gas. Using the proposed mathematical model, an unsteady turbulent gas flow was studied, followed by combustion in a combustion chamber. For completeness of fuel combustion and achievement of maximum efficiency during fuel combustion, the flow swirl was simulated.

3. Research results and conclusion

The combustion chamber, modeled in the "Design Modeler" module, is a cylinder with a given constant wall temperature T_wall = 617 K and having the following geometric characteristics L = 7.3 m, D = 4 m. The temperature of the combustion chamber wall is selected based on the temperature conditions of the coolant in the wall tubes of the power water-tube drum boiler. The combustion chamber has one combined burner, which allows for the simultaneous combustion of both natural gas and biogas. The burner with central gas supply contains a channel for natural gas supply S_methane = 0.00125 m^2, channel for biogas supply S_biogas = 0.078 m^2 and channel for air supply S_air = 0.234 m^2. The burner device is shown in Fig. 1.

The generation of the computational mesh was carried out using the "Meshing" module, a regular conformal mesh was applied with the refinement of mesh elements along the broaching thickness. The computational mesh in the "Meshing" module is shown in Fig. 2.

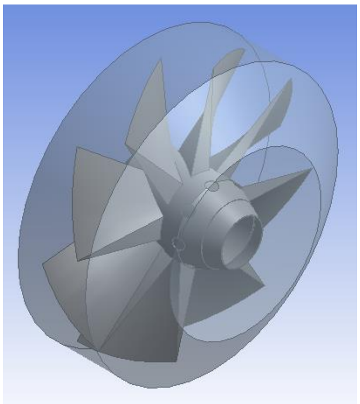


FIGURE 1. Combined burner

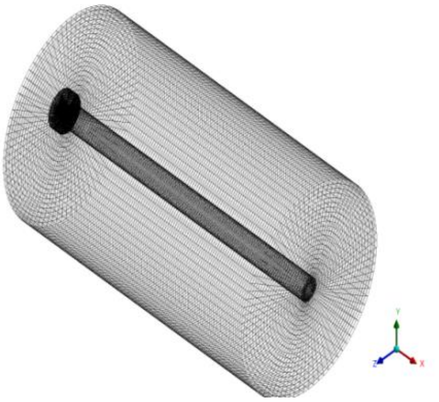
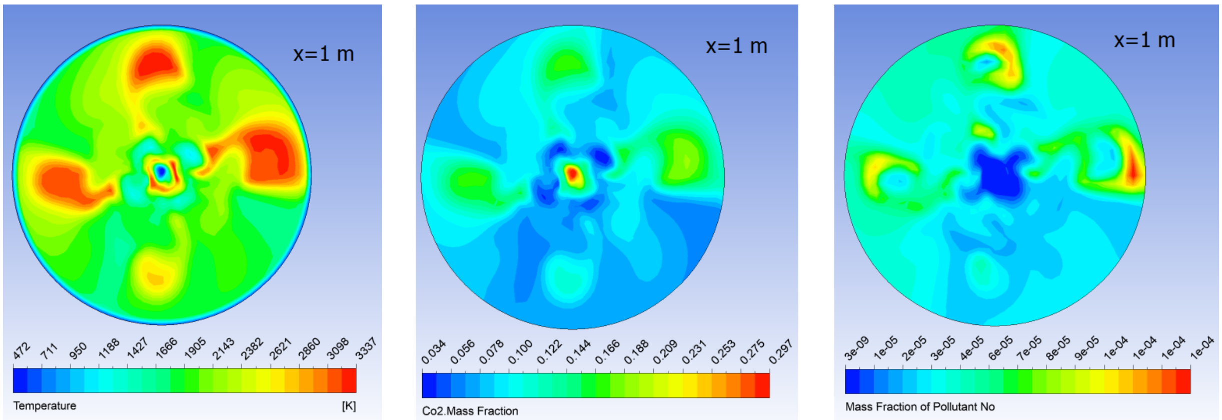


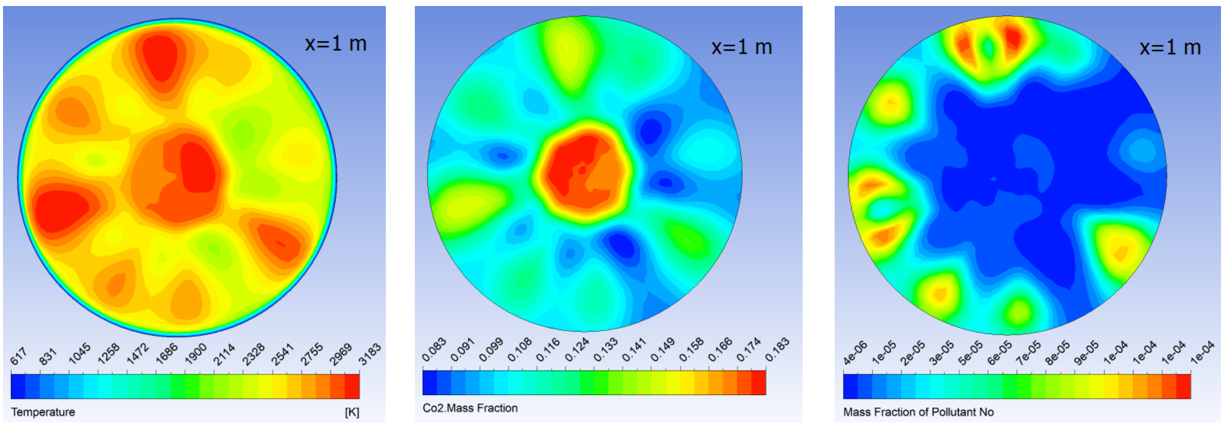
FIGURE 2. Meshing of a cylindrical combustion chamber

The study simulated joint combustion mixture of natural gas and biogas. In fig. 3,4,5,6,7 the results of combustion of fuel combinations with air swirl ω = (150, 300) rad/s, mass air flow m_air = 5.5 kg/s, air temperature at the inlet to the burner T_air = 583 K are presented.



Temperature CO_2 NO_x

(a)



Temperature CO_2 NO_x

(b)

FIGURE 3. Profile of Temperature, CO_2 and NO_x content at x = 1 m, from the embrasure of the combined burner: (a) Swirling air ω = 150 rad/s; (b) Swirling air ω = 300 rad/s.

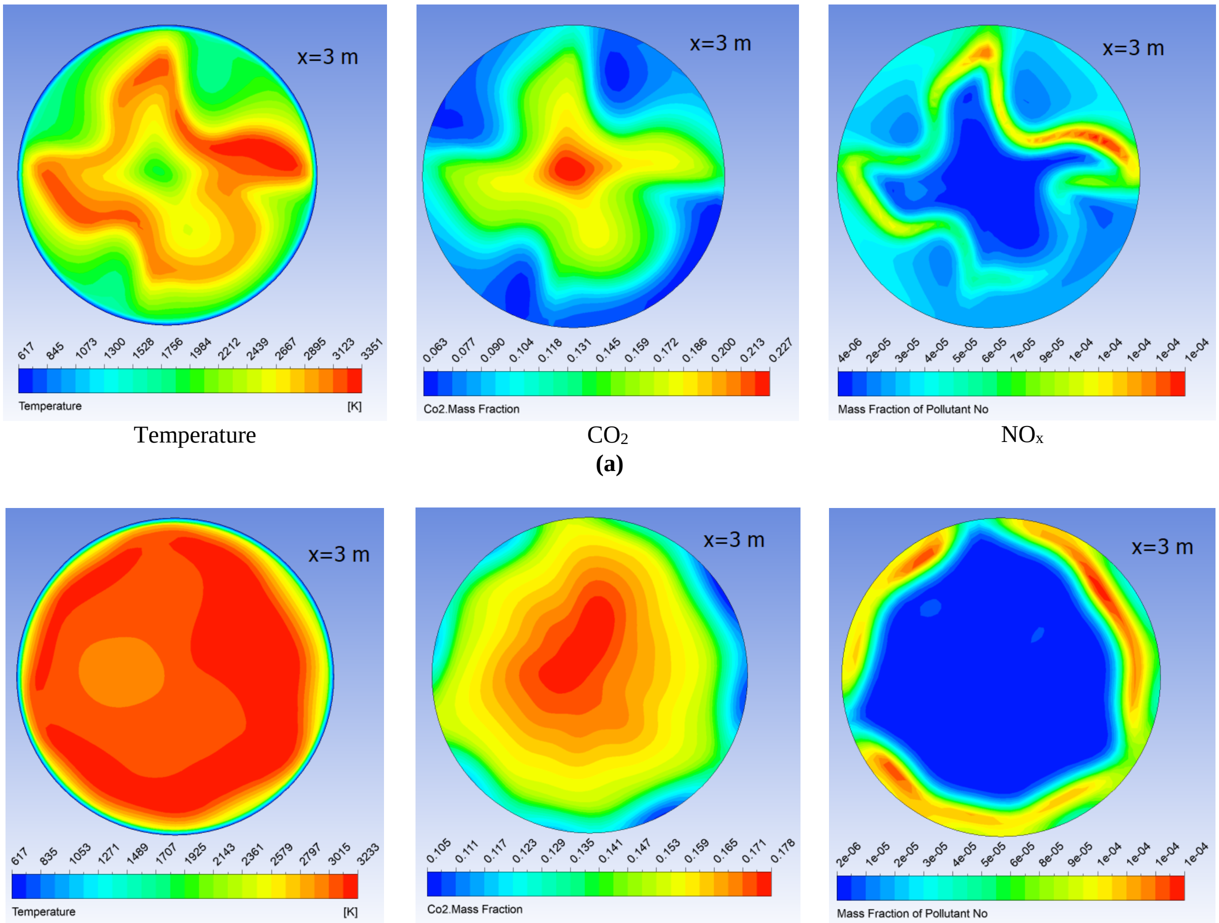


FIGURE 4. Profile of Temperature, CO_2 and NO_x content at x = 3 m, from the embrasure of the combined burner: (a) Swirling air ω = 150 rad/s; (b) Swirling air ω = 300 rad/s.

The combustion chamber air temperature is selected based on the condition of its preliminary heating in the air heater of the power boiler. Simulated fuel consumption through the burner for co-firing natural gas m_methane = 0.2 k/s, biogas m˙_biogas = 0.3 kg/s. The combustion results are presented in the vertical section of the flame at x = 1 m, x = 3 m, from the embrasure of the combined burner device.

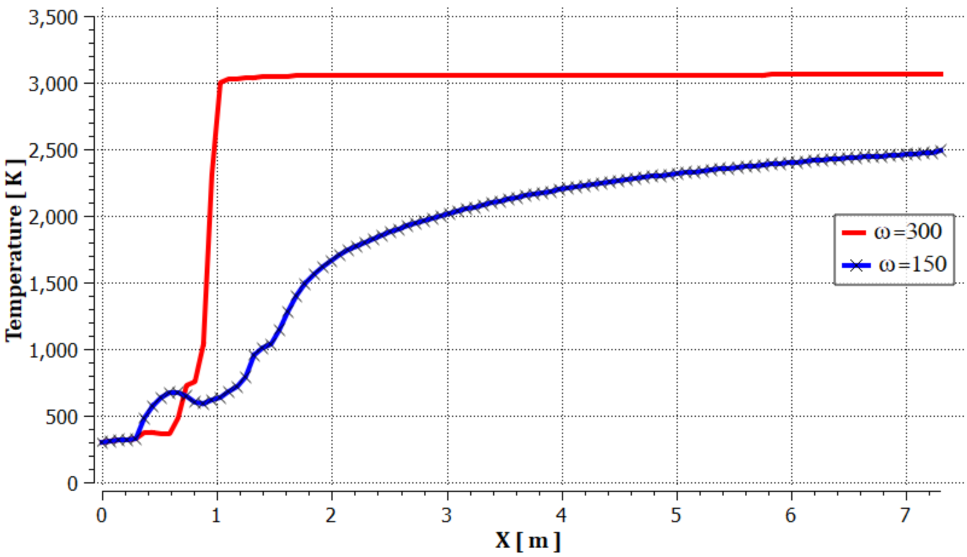
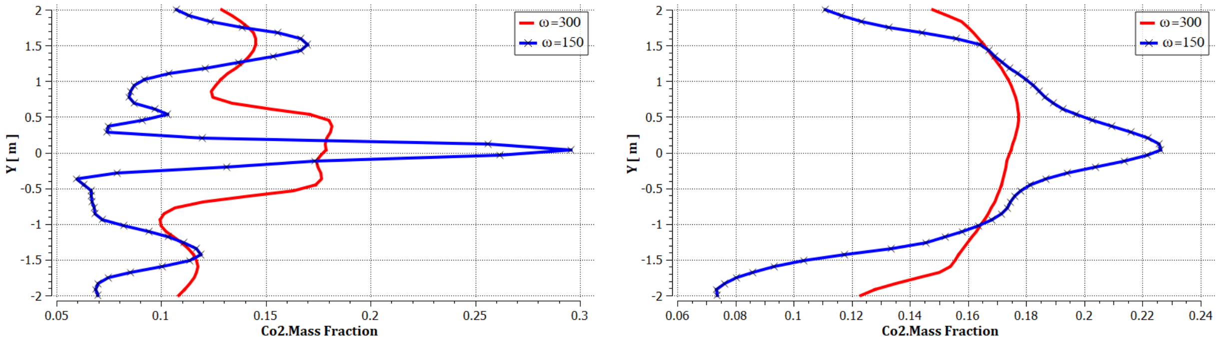


FIGURE 5. Temperature distribution along the x axis at different of air swirling.



(a) **(b)**

FIGURE 6. Profile of CO_2 content at different swirl speeds:
(a) at x = 1 m from the embrasure of the combined burner; (b) at x = 3 m from the embrasure of the combined burner.

The main problem in the combustion of biogas is the low speed of flame propagation in the combustion chamber, which leads to an increase in the length of the flame and flame separation, as a result of which the burner can go out. To stabilize the combustion process of biogas, it is necessary to increase the swirl of the fuel-air flow, and it is also desirable to use a combined method of burning biogas and natural gas in boilers. Based on the results obtained using the ANSYS Fluent software package, a significant difference in the NO_x content in combustion products can be noted, which is primarily justified by a decrease in temperature in the center of the flame, which is the main criterion for the intensity of NO_x formation in the furnace of power boilers. The decrease in temperature in the active combustion zone is explained by the presence of CO_2 in the biogas, which is a distinctive feature of this type of fuel.

Acknowledgments

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