Analysis of biogas content influence on the flame properties

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Abstract. The paper has examined the combustion of gas with variable methane content where it was supplied separately from the air. The analytical one-dimensional model of the diffusion straight-flow flame in the unrestricted and restricted space and a numerical three-dimensional model of the turbulent flame in a swirled airflow were used in this research. It has been established that in furnaces (in a restricted space) the maximum flame temperature depends only on the methane content in fuel and in average lower than the temperature of clean methane combustion by 20-70°C for conventional biogas and by 100-200°C for biogas with low methane content. The methane content decrease in gas leads to the diffusion flame length decrease in comparison with clean methane combustion by approximately two-fold for conventional biogas and fourfold for combustion of biogas with low methane content. For the turbulent flame in a swirled air flow, the flame length decrease is insignificant and is around 10% for conventional biogas and around 40% for biogas with low methane content. It can be concluded that by changing the methane content in biogas the heating plant operating mode can be stable on the condition that the biogas flow rate is controlled to ensure constant heat production during its combustion and turbulent flame with a swirled airflow. Combustion, in this case, can be done using the same burner.

1. Introduction
The last two decades showed an increased demand for energy and a large number of problems around nonrenewable energy sources which triggered the interest of the researchers in alternative energy sources study [1].

Every year amount of organic waste in cities and rural communities, as well as the production of a thousand tons of sludge from wastewater and wastewater at different agricultural and food factories, is growing. Such a state of matter leads to serious economic and ecological difficulties, but at the same time, this is a great potential for biogas production [2].

Biogas is a gas that is the result of an organic matter anaerobic fermentation. Biogas can be made from solid municipal waste, wastewater, animal manure, and agricultural waste. Biogas content heavily depends on the assimilated substrate characteristics and operating conditions of its source. In general, biogas consists of CH₄, CO₂, N₂, O₂, H₂S, and H₂. Considering methane's presence in biogas, it can be used as a fuel for heat and electricity production [3].

Biogas is a natural product and is increasingly considered as a partial alternative choice against fossil fuel, which is associated, mainly, with the fact that fossil fuel accelerates global warming and depends on geopolitical issues. However, as the author [4] says the main characteristics of biogas com-
bustion such as the ignition delay period and its roles in the polluting substance production, like aromatic compounds and soot, are still unclear.

The most common and accessible way of direct biogas use is combustion due to its low cost and low maintenance expenses. This kind of utilization does not require the removal of H2S and moisture from the biogas. Usually, the conversion efficiency of biogas into heat in the dual-fuel burner is around 80-90% [5].

Raw biogas cannot be directly used in commercial burners instead of natural gas or liquefied petroleum gases (propane, butane). When biogas under pressure is supplied into the conventional burner, which is normally used for natural or liquefied gas, the air-to-fuel ratio is not sufficient for stable combustion due to CO2 high content in the biogas. Therefore, biogas combustion requires the installation of a new burner with a separate instrumentation system. Considering the fact that the biogas components have corrosive nature (H2O and H2S,) the condensate formation needs to be prevented by keeping furnace temperature higher than the dew-point temperature. This means that the furnace should be heated by natural or liquefied gas to receive a higher operating temperature [6].

A study in the paper [6] has analyzed biogas combustion, mostly flameless in the simulated combustion chamber, and demonstrated volume dependence of conventional combustion of biogas and CH4 (Figure 1, 2).

The burner design depends on the type of fuel that is being used. Different types of fuel have different rates of burning and different requirements for oxidizer which leads to changes in nozzle and burner head design.

The flame velocity plays an important role in burner designing when biogas is burned in a conventional way. In order to prevent flame blow off it is required to control the flow rate of fuel and air into the chamber. CH4 concentration in the biogas components is by one order lower than in the natural gas, therefore, conventional biogas combustion provides lower flame velocity. This means that the air and fuel flow rates should be decreased to prevent a flame blow off. The maximum flame velocity is achieved at the stoichiometric air-to-fuel ratio. Because of the non-combustible components in biogas, the flame temperature during conventional biogas combustion is lower than when natural gas is used [7].
Figure 2. Temperature change at the central line of the furnace during methane and biogas combustion [6].

The paper [8] demonstrates combustion temperature dependence on CO$_2$ content (Figure 3).

Figure 3. Gaseous fuel combustion with different CO$_2$ concentration [8].
There is one more way of energy use of biogas, flameless combustion, efficiency research on which is presented in the paper by the author [9]. Reactions during flameless combustion proceed at special combustion mode conditions when the volume and hot temperature of the medium are maintained higher than the temperature of self-ignition. This method does not produce any visible flame or light effect; therefore, this mode was named “flameless oxidation”. The opposite effect is present in the conventional combustion systems where reactions are concentrated in a narrow flame front. The flameless burners can burn fuel with variable composition and unsteady quality or liquid-crystal fuel. This is their major advantage [10].

Based on the author’s studies the regenerative or recuperative heat exchangers are the best equipment for heat recuperation for the flameless combustion method but it is very expensive. To keep the air temperature higher than the biogas self-ignition temperature, during the laboratory experiment [11], an additional electrical heater was used. The necessary air for combustion is supplied by the fan and the incoming air is pre-heated by the furnace exhaust gases in the heat exchanger. The air and gas flow rate regulators, gas analyzer, corresponding control valves, digital temperature meter, biogas, and CH4 capsules are the rest of the equipment that was used in this experiment.

Simple burner design is one of the prime advantages of flameless combustion systems.

The same study [6] has demonstrated the dependence of flameless combustion forming on oxygen content (Figure 4).

Figure 4. Biogas flameless combustion forming in relation to different oxygen concentration [6].

Conclusions of the chapter:
• there are several ways of biogas utilization, more specifically conventional and flameless combustion, modernization of biogas to methane purity of 95%, and its further use as a fuel;
• biogas modernization processes are very costly and most of the time economically unprofitable;
• biogas requires special combustion conditions.
2. Materials and Methods

Currently, commercial furnace designing, most of the time, is based on the approximative empirical dependences [12]. The objective of the paper is the study of gas combustion with variable methane content for use in conventional fuel facilities.

The research has used three mathematical models of gaseous fuel combustion during its separate feeding with air:

- straight-flow diffusion flame in an unrestricted space (one-dimensional analytical model);
- straight-flow diffusion flame in a restricted space (one-dimensional analytical model);
- turbulent flame in a swirled airflow (three-dimensional model).

2.1. Straight-flow diffusion flame in an unrestricted space

At diffusion combustion, a gaseous fuel, which is flowing out of the burner nozzle, involves surrounding air into the movement and creates a turbulent stream (Figure 5). The volume of the dragged in the surrounding air is limited only by the turbulent stream size. The diffusion combustion intensity depends on the speed of the gaseous fuel turbulent mixing with air. Chemical reactions of combustion in the diffusion flame proceed with a speed far much higher than the turbulent mixing and, therefore, practically doesn’t influence the diffusion flame size.

![Figure 5. Diffusion flame in an unrestricted space [13]: $L_f$ – flame length; $x_0$ – distance from a pole of a stream to a burner outlet; $x$ – coordinate; $r$ – flame diameter; $R$ – turbulent stream diameter; 1 – fuel; 2 – air; 3 – burner; 4 – an internal area of a flame; 5 – turbulent stream boundaries; 6 – flame front (area of a chemical reaction during combustion); 7 – an outer area of a flame.](image)

The calculations were done by using the one-dimensional analytical model presented in the monograph [13].

2.2. Straight-flow diffusion flame in a restricted space

The gaseous fuel combustion in a space limited by the cylindrical walls of the furnace was analyzed. In this case, the amount of air going for combustion is limited. The modeling was carried out using the excess air coefficient 1.1.

The flame can be divided into two areas: the area of streamflow which ends when all air is dragged in into a stream and the area of an afterburning (Figure 6).

At the streamflow section, around 80% of natural gas is burned. The rest of the fuel burns out in the second section of the diffusion flame as it mixes with the air that was dragged into the turbulent streamflow movement. In the second section the lower the amount of unburnt fuel in the flame the lower the combustion intensity. As a result, the flame length is longer than during combustion in the unrestricted space.

For the flame calculation, the one-dimensional analytical model was used which is presented in the monography [13].
2.3. Turbulent flame in a swirled airflow

The mathematical model of natural gas combustion and heat transfer in the vertical cylindrical furnace of the water heater included the system of differential equations of the turbulent motion, combustion, and radiative-convective heat exchange. The equations were solved using the numerical finite volume method. For every point of the calculation grid (Figure 7) were determined components of velocity in all three axes, fuel and air concentration, the temperature of the gaseous medium.

Figure 6. Diffusion flame in a restricted space [13]: \( L_f \) – flame length; \( L_s \) – streamflow length; \( x_0 \) – distance from a pole of a stream to a burner outlet; \( x \) – coordinate; \( r \) – flame diameter; \( R \) – turbulent stream diameter; \( I \) – fuel; \( 2 \) – air; \( 3 \) – burner; \( 4 \) – furnace lining; \( 5 \) – turbulent stream boundaries; \( 6 \) – flame front (area of a chemical reaction during combustion).

Figure 7. Calculation grid for numerical combustion modeling in the cylindrical furnace: 1 – fuel; 2 – air; 3 – burner; 4 – conical section of the furnace; 5 – cylindrical section of the furnace; 6 – an outlet for exhaust gases.

Semiempirical Prandtl's theory of turbulence [14] application has allowed correctly formulate the boundary conditions on the coarse grid [15] for the discretized equations, which are applied in the mathematical models for calculation of velocity, eddy viscosity, temperature, and convective heat
transfer to the walls that limit turbulent gas flow [16]. The improved differential equations of the diffusion mathematical model [17] of the radiative heat transfer in the attenuation medium have allowed to formulate the correct boundary conditions on the enclosure wall surface at the diffuse reflection of the incident radiation [18].

3. Results and Discussion

The landfill gas is characterized by a variable composition [19]. Major components of biogas are methane, nitrogen, and carbon dioxide [20]. The modeling has used biogas of two different compositions: normal with 60% of methane content and with a low methane content of 35%. The biogas combustion was compared to methane combustion.

Three variants were considered for modeling (Table 1):

a) variable composition gas combustion with the same volumetric flow rate of gas supplied into the burner;

b) combustion with a constant heating value (heat released during its combustion) using the same burner;

c) combustion with a constant heating value and burner diameter changing to ensure constant gas velocity.

Table 1. Calculation variants.

| Gas composition (by volume) | Constant gas flow rate | Constant heating value ($P = 1 \text{ MW}$) |
|-----------------------------|------------------------|-------------------------------------------|
|                             | Flow rate $Q$, m$^3$/h (m$^3$/sec) | Burner diameter $d$, mm | Exhaust velocity $v$, m/s | Burner diameter $d$, mm | Exhaust velocity $v$, m/s |
| CH$_4$ CO$_2$ N$_2$ | | |
| 100% | – | – | 100 (0.0278) | 13.3 | 200 | – | – | – | – | – |
| 60% 20% 20% | – | – | 167 (0.0463) | 13.3 | 334 | 17.2 | 200 |
| 35% 43% 22% | – | – | 286 (0.0794) | 13.3 | 572 | 22.5 | 200 |

3.1. Modeling of gas combustion in a straight-flow diffusion flame in an unrestricted space

Figure 8 and Table 2 present the results obtained during the gas combustion modeling in the unrestricted space.

Conclusions of the chapter:

a) the maximum inside flame temperature depends only on the methane content and doesn’t depend on the fuel’s exhaust velocity;

b) the methane content in gas reduction leads to the flame length decrease, but for the same burner it stays constant and doesn’t depend on the fuel’s exhaust velocity.

3.2. Modeling of gas combustion in a straight-flow diffusion flame in a restricted space

Figure 9 and Table 2 present the results obtained during gas combustion modeling in the space limited by walls of the cylindrical furnace. Temperature curve behavior in Figure 9 reveals lengths of the streamflow section which ends up when all air is dragged into the flow and section of fuel afterburning.
Figure 8. Flame temperature along its length:

a) at constant flow rate $Q = 100 \text{ m}^3/\text{h}$:
- 1 – $\text{CH}_4=100\%$ ( ); 2 – $\text{CH}_4=60\%$ ( ); 3 – $\text{CH}_4=35\%$ ( );

b) at constant heating value $P = 1 \text{ MW}$, and fuel nozzle diameter $d = 13.3 \text{ mm}$:
- 4 – $\text{CH}_4=60\%$ ( ); 5 – $\text{CH}_4=35\%$ ( );

c) at constant heating value $P = 1 \text{ MW}$ and exhaust gas velocity $v = 200 \text{ m/s}$:
- 6 – $\text{CH}_4=60\%$ ( ); 7 – $\text{CH}_4=35\%$ ( ).

Figure 9. Flame temperature along its length:

a) at constant flow rate $Q = 100 \text{ m}^3/\text{h}$:
- 1 – $\text{CH}_4=100\%$ ( ); 2 – $\text{CH}_4=60\%$ ( ); 3 – $\text{CH}_4=35\%$ ( );

b) at constant heating value $P = 1 \text{ MW}$, and fuel nozzle diameter $d = 13.3 \text{ mm}$:
- 4 – $\text{CH}_4=60\%$ ( ); 5 – $\text{CH}_4=35\%$ ( );

c) at constant heating value $P = 1 \text{ MW}$ and exhaust gas velocity $v = 200 \text{ m/s}$:
- 6 – $\text{CH}_4=60\%$ ( ); 7 – $\text{CH}_4=35\%$ ( ).

Conclusions of the chapter:

a) the maximum inside flame temperature notably less dependent on the methane content in gas, for conventional biogas it is lower than the clean methane combustion temperature by 20-60°C, for biogas with low methane content – by 85-110°C;

b) the flame length gets smaller when methane content in gas decreases, but at the same time for the same burner it stays constant and doesn’t depend on the output gas velocity;

c) biogas flame length decrease value in comparison with methane flame for unrestricted and restricted space are the same.
Table 2. Flame length $L_{fl}$, m.

| Variants of modeling               | Combustion in the unrestricted space | Combustion in the restricted space |
|------------------------------------|--------------------------------------|-----------------------------------|
|                                    | $CH_4$=100% | $CH_4$=60% | $CH_4$=35% | $CH_4$=100% | $CH_4$=60% | $CH_4$=35% |
| a) constant flow rate $Q = 100$ m$^3$/h | 2.74        | 1.35        | 0.69        | 3.08        | 1.44        | 0.71        |
| b) heating value $P = 1$ MW, nozzle diameter $d = 13.3$ mm | -“-”        | 1.36        | 0.69        | -“-”        | 1.45        | 0.7        |
| c) heating value $P = 1$ MW, exhaust velocity $v = 200$ m/s | -“-”        | 1.75        | 1.16        | -“-”        | 1.87        | 1.2        |

$^a$ – of 2.74 m; $^b$ – of 3.08 m

3.3. Modeling of gas combustion in a turbulent flame in a swirled airflow

Figure 10 and Table 3 present the results of numerical modeling of methane and biogas combustion in the vertical furnace of the water heater.

Fuel and air concentration field.

Temperature field

| $CH_4$ | 100% | 60% | 35% | 60% | 35% | 60% | 35% |
|--------|------|-----|-----|-----|-----|-----|-----|
| $Q$, m$^3$/h | 100  | 167 | 286 | 167 | 286 |
| $P$, MW | 1.0  | 0.6 | 0.35| 1.0 |     |
| $d$, mm | 13.3 |    |    | 17.2| 22.5|
| $m$, m/s | 200 | 334 | 572 | 200 |

Figure 10. Gas, air, and temperature concentration fields for methane and biogas combustion in the vertical furnace of the water heater.
Table 3. Flame length Lfl, m, and gas temperature during methane and biogas combustion in the vertical furnace of the water heater.

| Variants of modeling          | Flame length Lfl, m; | Maximum temperature; | The mean temperature at the nozzle area: |
|-------------------------------|----------------------|-----------------------|----------------------------------------|
|                               | CH<sub>4</sub>=100%;| CH<sub>4</sub>=60%;  | CH<sub>4</sub>=35%;                    |
| a) constant flow rate Q = 100 m³/h | 1.86 m              | 1.67 m (89%)<sup>a</sup> | 1.07 m (58%)<sup>a</sup>              |
|                               | 1877°C              | 1807°C                | 1668°C                                 |
|                               | 1276°C              | 960°C                 | 983°C                                  |
| b) heating value P = 1 MW,     | -”-                 | 1.69 m (91%)<sup>a</sup>| 1.06 m (57%)<sup>a</sup>              |
| nozzle diameter d = 13.3 mm    |                     | 1840°C                | 1673°C                                 |
|                               |                     | 1401°C                | 1212°C                                 |
| c) heating value P = 1 MW,     | -”-                 | 2.17 (117%)<sup>a</sup>| 1.77 (95%)<sup>a</sup>                |
| exhaust velocity v = 200 m/s   |                     | 1817°C                | 1668°C                                 |
|                               |                     | 1414°C                | 1306°C                                 |

<sup>a</sup> – of 1.86 m.

Conclusions of the chapter:

a) the maximum temperature inside the flame for conventional biogas is lower than the temperature of clean methane combustion by 40-70°C, for biogas with low methane content – by 200°C;
b) the temperature at the furnace exit during biogas combustion can be even higher than during methane combustion because of a smaller amount of supplied air;
c) decrease of the methane content in gas leads to a much smaller decrease of the flame length than for the diffusion flame.

4. Conclusion

The paper has examined the combustion of gas with variable methane content where it was supplied separately from the air. The analytical one-dimensional model of the diffusion straight-flow flame in the unrestricted and restricted space and a numerical three-dimensional model of the turbulent flame in a swirled airflow were used in this research.

The research has examined combustion of clean methane and biogas with a methane content of 60 and 35% in three different variants: a) combustion of variable content gas with the same gas volumetric flow rate; b) combustion with a constant amount of heat, released during fuel combustion, using the same burner which required gas flow rate increase; c) combustion with a constant amount of heat, released during fuel combustion, using different diameter burners to ensure constant gas velocity.

It has been established that in furnaces (in a restricted space) the maximum flame temperature depends only on the methane content in fuel and in average lower than the temperature of clean methane combustion by 20-70°C for conventional biogas and by 100-200°C for biogas with low methane content. The methane content decrease in gas leads to the diffusion flame length decrease in comparison with clean methane combustion by approximately two-fold for conventional biogas and fourfold for combustion of biogas with low methane content. For the turbulent flame in a swirled air flow, the flame length decrease is insignificant and is around 10% for conventional biogas and around 40% for biogas with low methane content.

It can be concluded that by changing the methane content in biogas the heating plant operating mode can be stable on the condition that the biogas flow rate is controlled to ensure constant heat production during its combustion and turbulent flame with a swirled airflow. Combustion, in this case, can be done using the same burner.
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References
[1] Isci A and Demirer G N 2007  Biogas production potential from cotton wastes Renew. Energy Pergamon 32 750–7
[2] Taleghani G and Kia A S 2005 Technical-economical analysis of the Saveh biogas power plant Renew. Energy Pergamon 30 441-6
[3] Hoteit A, Chandel M K, Durecu S and Delebarre A 2009 Biogas combustion in a chemical looping fluidized bed reactor Int. J. Greenh. Gas Control 3 561-7
[4] Fischer M and Jiang X 2017 Numerical studies of CO formation during biogas combustion Energy Procedia Elsevier 142 426-31
[5] Kapoor R, Ghosh P, Tyagi B, Vijay V K, Vijay V, Thakur I S, Kamayab H, Nguyen D D and Kumar A 2020 Advances in biogas valorization and utilization systems: A comprehensive review J. Clean. Prod. Elsevier Ltd 273 123052
[6] Hosseini S E, Bagheri G and Wahid MA 2014 Numerical investigation of biogas flameless combustion Energy Convers. Manag. Pergamon 81 41-50
[7] Hosseini S E and Wahid M A 2014 Development of biogas combustion in combined heat and power generation Renewable and Sustainable Energy Reviews Elsevier Ltd 40 868-75
[8] Mordant C J and Pierce W C 2014 Design and preliminary results of an atmospheric-pressure model gas turbine combustor utilizing varying CO2 doping concentration in CH4 to emulate biogas combustion Fuel. Elsevier 124 258-68
[9] Hosseini S E and Wahid M A, Abuelnuor A A A 2014 Biogas flameless combustion: A review Appl. Mech. Mater. 388 273-9
[10] Colorado A F, Herrera B A and Amell A A 2010 Performance of a Flameless combustion furnace using biogas and natural gas Bioresour. Technol. Elsevier Ltd 101 2443-9
[11] Hosseini S E and Wahid M A 2013 Biogas utilization: Experimental investigation on biogas flameless combustion in lab-scale furnace Energy Convers. Manag. Pergamon 74 426-32
[12] Kuznetsov V A and Trubaev P A 2018 Resources and Problems of the Mathematical Simulating Thermo-Technological Processes J. Phys.: Conf. Ser. 1066 012024
[13] Kuznetsov V A and Trubaev P A 2017 Matematicheskie modeli teplomassoperenosa v vy`okotemperaturny`x ustanovkax [Mathematical models of a heatmass transfer in high-temperature unit] (Belgorod: BSTU)
[14] Kuznetsov V A 2020 Semiempirical model of convective heat transfer of turbulent gases J. Eng. Phys. Thermophys. 93 543-50
[15] Kuznetsov V A 1986 Refinement of wall-turbulence hypotheses J. Eng. Phys. Thermophys. 50(6) 640-4
[16] Kuznetsov V A, Trubaev P A and Ryazancev O A 2020 An elaborated diffusion mathematic model of radiative transfer in an extinction medium IOP Conf. Ser.: Mater. Sci. Eng. 791 012015
[17] Kuznetsov V A 2017 Computational simulation of convective heat transfer of turbulent gas flows Theor. Found. Chem. Eng. 51(6) 1063-9
[18] Kuznetsov V A 2017 Mathematical model of the radiative heat exchange in the selective gases of a diffusion flame J. Eng. Phys. Thermophys. 90(2) 357-65
[19] Trubaev P A, Verevkin O V, Grishko B M, Tarasyuk P N, Shchechin I I, Suslov D Yu and Ramazanov R S 2018 Investigation of Landfill Gas Output from Municipal Solid Waste at the Polygon J. Phys.: Conf. Ser. 1066 012015
[20] Zubkova O A and Sergeeva O A 2018 Control of environmental hazards of thermal-sough products determining modern finishing polymeric building materials Chemical Bulletin 1(2) 31-37